HANDBOOK OF AIR CONDITIONING AND REFRIGERATION

Shan K. Wang

Second Edition

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This book is dedicated to my dear wife Joyce for her encouragement, understanding, and contributions, and to my daughter Helen and my sons Roger and David.

ABOUT THE AUTHOR

Shan K. Wang received his B.S. in mechanical engineering from Southwest Associated University in China in 1946. Two years later, he completed his M.S. degree in mechanical engineering at Harvard Graduate School of Engineering. In 1949, he obtained his M.S. in textile technology from the Massachusetts Institute of Technology.

From 1950 to 1974, Wang worked in the field of air conditioning and refrigeration in China. He was the first Technical Deputy Director of the Research Institute of Air Conditioning in Beijing from 1963 to 1966 and from 1973 to 1974. He helped to design space diffusion for the air conditioning system in the Capital and Worker's Indoor Stadium. He also designed many HVAC&R systems for industrial and commercial buildings. Wang published two air conditioning books and many papers in the 1950s and 1960s. He is one of the pioneers of air conditioning in China.

Wang joined Hong Kong Polytechnic as senior lecturer in 1975. He established the air conditioning and refrigeration laboratories and established courses in air conditioning and refrigeration at Hong Kong Polytechnic. Since 1975, he has been a consultant to Associated Consultant Engineers and led the design of the HVAC&R systems for Queen Elizabeth Indoor Stadium, Aberdeen Market Complex, Koshan Road Recreation Center, and South Sea Textile Mills in Hong Kong. From 1983 to 1987, Wang Published *Principles of Refrigeration Engineering* and *Air Conditioning* as the teaching and learning package, and presented several papers at ASHRAE meetings. The First Edition of the *Handbook of Air Conditioning and Refrigeration* was published in 1993.

Wang has been a member of ASHRAE since 1976. He has been a governor of the ASHRAE Hong Kong Chapter-At-Large since the Chapter was established in 1984. Wang retired from Hong Kong Polytechnic in June 1987 and immigrated to the United States in October 1987. Since then, he has joined the ASHRAE Southern California Chapter and devoted most of his time to writing.

PREFACE TO SECOND EDITION

Air conditioning, or HVAC&R, is an active, rapidly developing technology. It is closely related to the living standard of the people and to the outdoor environment, such as through ozone depletion and global warming. Currently, air conditioning consumes about one-sixth of the annual national energy use in the United States.

At the beginning of a new millennium, in addition to the publication of ASHRAE Standard 90.1-1999 and ASHRAE Standard 62-1999, often called the Energy standard and Indoor Air Quality standard, the second edition of *Handbook of Air Conditioning and Refrigeration* is intended to summarize the following advances, developments, and valuable experience in HVAC&R technology as they pertain to the design and effective, energy-efficient operation of HVAC&R systems:

First, to solve the primary problems that exist in HVAC&R, improve indoor air quality through minimum ventilation control by means of CO_2-based demand-controlled or mixed plenum controlled ventilation, toxic gas adsorption and chemisorption, medium- and high-efficiency filtration, and damp surface prevention along conditioned air passages. ANSI/ASHRAE Standard 52.2-1999 uses 16 minimum efficiency reporting values (MERVs) to select air filters based on particle-size composite efficiency.

Energy conservation is a key factor in mitigating the global warming effect. Electric deregulation and the use of real-time pricing instead of the time-of-use rate structure in the United States have a significant impact on the energy cost. ANSI/ASHRAE Standard 90.1-1999 has accumulated valuable HVAC&R energy-efficient experiences since the publication of Standard 90.1-1989 and during the discussions of the two public reviews.

For buildings of one or two stories when the outdoor wind speed is normal or less than normal, the space or building pressurization depends mainly on the air balance of the HVAC&R system and on the leakiness of the building. A proper space pressurization helps to provide a desirable indoor environment.

Second, there is a need for a well-designed and -maintained microprocessor-based energy management and control system for medium-size or large projects with generic controls in graphical display, monitoring, trending, totalization, scheduling, alarming, and numerous specific functional controls to perform HVAC&R operations in air, water, heating, and refrigeration systems. HVAC&R operations must be controlled because the load and outside weather vary.

The sequence of operations comprises basic HVAC&R operations and controls. In the second edition, the sequence of operations of zone temperature control of a single-zone VAV system, a VAV reheat system, a dual-duct VAV system, a fan-powered VAV system, and a four-pipe fan-coil system is analyzed. Also the sequence of operations of a plant-building loop water system control, the discharge air temperature control, and duct static pressure control in an air-handling unit are discussed.

Third, new and updated advanced technology improvements include

- Artificial intelligence, such as fuzzy logic, artificial neural networks, and expert systems, is widely used in microprocessor-based controllers.
- BACnet is an open protocol in control that enables system components from different vendors to be connected to a single control system to maximize efficiency at lowest cost.
- Computational fluid dynamics is becoming an important simulation technology in airflow, space diffusion, clean rooms, and heat-transfer developments.

- Scroll compressors are gradually replacing reciprocating compressors in packaged units and chillers because of their higher efficiency and simple construction.
- Ice storage systems with cold air distribution shift the electric power demand from on-peak hours to off-peak hours and thus significantly reduce the energy cost.
- Desiccant-based air conditioning systems replace part of the refrigeration by using evaporative cooling or other systems in supermarkets, medical operation suites, and ice rinks.
- Fault detection and diagnostics determine the reason for defects and failures and recommend a means to solve the problem. It is a key device in HVAC&R operation and maintenance.

Fourth, air conditioning is designed and operated as a system. In the second edition, HVAC&R systems are classified in three levels. At the air conditioning system level, systems are classified as individual, evaporative, space, packaged, desiccant-based, thermal storage, clean-room, and central systems. At the subsystem level, systems are classified as air, water, heating, refrigeration, and control systems. At the main component level, components such as fans, coils, compressors, boilers, evaporators, and condensers are further divided and studied. Each air conditioning system has its own system characteristics. However, each air conditioning system, subsystem, and main component can be clearly distinguished from the others, so one can thus easily, properly, and more precisely select a require system.

Fifth, computer-aided design and drafting (CADD) links the engineering design through calculations and the graphics to drafting. CADD provides the ability to develop and compare the alternative design schemes quickly and the capability to redesign or to match the changes during construction promptly. A savings of 40 percent of design time has been claimed.

Current CADD for HVAC&R can be divided into two categories: engineering design, including calculations, and graphical model drafting. Engineering design includes load calculations, energy use estimates, equipment selection, equipment schedules, and specifications. Computer-aided drafting includes software to develop duct and pipework layouts and to produce details of refrigeration plant, heating plant, and fan room with accessories.

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Shan K. Wang

PREFACE TO THE FIRST EDITION

Air conditioning, or more specifically, heating, ventilating, air ventilating, air conditioning, and refrigeration (HVAC&R), was first systematically developed by Dr. Willis H. Carrier in the early 1900s. Because it is closely connected with the comfort and health of the people, air conditioning became one of the most significant factors in national energy consumption. Most commercial buildings in the United States were air conditioned after World War II.

In 1973, the energy crisis stimulated the development of variable-air-volume systems, energy management, and other HVAC&R technology. In the 1980s, the introduction of microprocessor-based direct-digital control systems raised the technology of air conditioning and refrigeration to a higher level. Today, the standards of a successful and cost-effective new or retrofit HVAC&R projects include maintaining a healthy and comfortable indoor environment with adequate outdoor ventilation air and acceptable indoor air quality with an energy index lower than that required by the federal and local codes, often using off-air conditioning schemes to reduce energy costs.

The purpose of this book is to provide a useful, practical, and updated technical reference for the design, selection, and operation of air conditioning and refrigeration systems. It is intended to summarize the valuable experience, calculations, and design guidelines from current technical papers, engineering manuals, standards, ASHRAE handbooks, and other publications in air conditioning and refrigeration.

It is also intended to emphasize a systemwide approach, especially system operating characteristics at design load and part load. It provides a technical background for the proper selection and operation of optimum systems, subsystems, and equipment. This handbook is a logical combination of practice and theory, system and control, and experience and updated new technologies.

Of the 32 chapters in this handbook, the first 30 were written by the author, and the last two were written by Walter P. Bishop, P. E., president of Walter P. Bishop, Consulting Engineer, P. C., who has been an HVAC&R consulting engineer since 1948. Walter also provided many insightful comments for the other 30 chapters. Another contributor, Herbert P. Becker, P. E., reviewed Chaps. 1 through 6.

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Shan K. Wang

INDEX

Abbreviations, **A.**9–**A.**10
Absolute zero, **2.**5
Absorption chiller-heaters, **14.**20–**14.**22
  actual performance, **14.**22
  heating cycle, **14.**20–**14.**22
Absorption chillers, double-effect, direct-fired, **14.**6–**14.**18
  absorber and solution pumps, **14.**6–**14.**7
  air purge unit, **14.**8–**14.**9
  capacity control and part-load operation, **14.**16–**14.**17
  coefficient of performance, **14.**14
  condenser, **14.**7–**14.**8
  condensing temperature, **14.**19–**14.**20
  controls, **14.**16–**14.**18
  cooling water entering temperature, **14.**19
  cooling water temperature control, **14.**17–**14.**18
  corrosion control, **14.**20
  crystallization and controls, **14.**17
  difference between absorption and centrifugal chillers, **14.**18–**14.**19
  evaporating temperature, **14.**19
  evaporator and refrigerant pump, **14.**6
  flow of solution and refrigerant, **14.**9–**14.**11
  generators, **14.**7–**14.**8
  heat exchangers, **14.**6–**14.**7
  heat removed from absorber and condenser, **14.**19
  mass flow rate of refrigerant and solution, **14.**11–**14.**12
  monitoring and diagnostics, **14.**18
  operating characteristics and design considerations, **4.**18–**4.**20
  performance of, **14.**11–**14.**16
  rated conditions, **14.**20
  safety and interlocking controls, **14.**18
  series flow, parallel flow, and reverse parallel flow, **14.**8–**14.**9
  Standard 90.1–1999 minimum efficiency requirements, **14.**20
  system description, **14.**6–**14.**8
  thermal analysis, **14.**12–**14.**14
  throttling devices, **14.**8
Absorption heat pumps, **14.**22–**14.**24
  case study: series connected, **14.**22–**14.**24
  functions of, **14.**22
Absorption heat transformer, **14.**24–**14.**26
  coefficient of performance, **14.**26
  operating characteristics, **14.**24–**14.**25
  system description, **14.**24–**14.**25
Accuracy, **2.**6
Adiabatic process, **2.**11
Adiabatic saturation process, ideal, **2.**11
Air:
  atmospheric, **2.**1
  dry air, **2.**1–**2.**2
  mass, **3.**25
  moist air, **2.**1
  primary, **20.**4
  process, **1.**4–**1.**5
  recirculating, **20.**4
  regenerative, **1.**4–**1.**5
  secondary, **20.**4
  transfer, **20.**4
  ventilation, **4.**29
Air cleaner, electronic, **15.**69–**15.**70
Air conditioning, **1.**1–**1.**2,
  industry, **1.**15
  project development, **1.**16–**1.**17
Air conditioning processes, **20.**41–**20.**53
  adiabatic mixing, **20.**50–**20.**52
  air washer, **20.**46
  bypass mixing, **20.**52–**20.**53
  cooling and dehumidifying, **20.**47–**20.**50
  heating element humidifier, **20.**46
  humidifying, **20.**45–**20.**47
  oversaturation, **20.**46–**20.**47
  reheating, recooling and mixing, **20.**74–**20.**75
  relative humidity of air leaving coil, **20.**49–**20.**50
  sensible heat ratio, **20.**41–**20.**43
  sensible heating and cooling, **20.**44–**20.**45
  space conditioning, **20.**43–**20.**44
  steam injection humidifier, **20.**45–**20.**46

Air conditioning systems, **1.**2
air, cooling and heating systems designation, **26.**2–**26.**3
central, **1.**6
central hydronic, **1.**6
classification, basic approach, **26.**1–**26.**2
classification of, **1.**3–**1.**10, **26.**2–**26.**3
clean room, **1.**5
comfort, **1.**2–**1.**3
desiccant-based, **1.**4
evaporative-cooling, **1.**4
individual room, **1.**4
packaged, **1.**6
space, **1.**5
space conditioning, **1.**5
thermal storage, **1.**5
unitary packaged, **1.**6
Air conditioning systems, individual, **26.**8–**26.**9
advantages and disadvantages, **26.**9
basics, **26.**8–**26.**9
Air conditioning systems, packaged terminal, **26.**13–**26.**15
equipment used, **26.**13–**26.**14
heating and cooling mode operation, **26.**13–**26.**14
minimum efficiency requirements, ASHRAE/IESNA Standard 90.1–1999, **26.**14–**26.**15
system characteristics, **26.**13, **26.**15
Air conditioning systems, room, **26.**9–**26.**13
configuration, **26.**10–**26.**11
controls, **26.**12
cooling mode operation, **26.**11
energy performance and energy use intensities, **26.**11–**26.**12
equipment used in, **26.**9–**26.**10
features, **26.**12
system characteristics, **26.**12–**26.**13
Air conditioning systems, selection:
applications and building occupancies, **26.**4–**26.**5
energy efficiency, **26.**7
fire safety and smoke control, **26.**7–**26.**8
indoor air quality, **26.**5–**26.**6
initial cost, **26.**8
maintenance, **26.**8
requirements fulfilled, **26.**4
selection levels, **26.**3–**26.**4
sound problems, **26.**6–**26.**7
space limitations, **26.**8
system capacity, **26.**5
zone thermal control, **26.**6
Air conditioning systems, space conditioning, **28.**1–**28.**3
advantages and disadvantages, **28.**2–**28.**3
applications, **28.**1–**28.**2
induction systems, **28.**3
Air contaminants, indoor, **4.**27–**4.**28, **15.**61
Air duct design, principles and considerations, **17.**43–**17.**51
air leakage, **17.**48–**17.**50
critical path, **17.**48
design procedure, **17.**51–**17.**52
design velocity, **17.**45–**17.**46
duct layout, **17.**52–**17.**53
duct system characteristics, **17.**52
ductwork installation, **17.**50
fire protection, **17.**50–**17.**51
optimal air duct design, **17.**43–**17.**45
sealing requirements of ASHRAE Standard 90.1–1999, **17.**49–**17.**50
shapes and material of air ducts, **17.**50
system balancing, **17.**46–**17.**47
Air expansion refrigeration cycle, **9.**45–**9.**49
flow processes, **9.**47–**9.**48
thermodynamic principle, **9.**45–**9.**47
Air filters, **15.**64–**15.**68
classification of, **15.**65
coarse, **15.**65
filter installation, **24.**7–**24.**8
filtration mechanism, **15.**64–**15.**65
high-efficiency, **15.**66–**15.**67
low-efficiency, **15.**65–**15.**66
medium-efficiency, **15.**66–**15.**67
service life, **24.**7
ultrahigh-efficiency, HEPA and ULPA filters, **15.**68
Air filters, rating and assessments, **15.**61–**15.**62
dust-holding capacity, **15.**62
efficiency, **15.**61
pressure drop, **15.**61–**15.**62
service life, **15.**62
Air filters, test methods, **15.**62–**15.**64
composite efficiency curves, **15.**63–**15.**64
di-octylphthalate (DOP), **15.**62–**15.**63
dust spot, **15.**62
minimum efficiency reporting values (MERVs), **15.**64–**15.**65
penetration, **15.**63
removal efficiency by particle size, **15.**63
selection, **15.**71–**15.**72
test unit, **15.**64
weight arrestance, **15.**62
Air filters to remove contaminants, **24.**6–**24.**8
filter selection for IAQ, **24.**6–**24.**7
remove indoor air contaminants, **24.**6

Air filtration and industrial air cleaning, **15.**60–**15.**61
Air flow, basics, **17.**2–**17.**8
  Bernoulli equation, **17.**2
  equation of continuity, **17.**7–**17.**8
  laminar flow and turbulent flow, **17.**6–**17.**7
  pressure, **17.**3
  stack effect, **17.**5–**17.**6
  static pressure, **17.**3–**17.**4
  steady flow energy equation, **17.**2–**17.**3
  total pressure, **17.**5
  velocity distribution, **17.**3
  velocity pressure, **17.**4–**17.**5
Air flow, characteristics, **17.**8–**17.**10
  air duct, types, **17.**8
  pressure characteristics, **17.**8–**17.**10
  static regain, **17.**9
  system pressure loss,**17.**10
Air-handling units, **1.**8, **16.**1–**16.**12
  casing, **16.**4
  classification of, **16.**2–**16.**4
  coil face velocity, **16.**8–**16.**9
  coils, **16.**5
  component layout, **16.**6–**16.**8
  controls, **16.**6
  draw-through or blow-through unit, **16.**2
  exhaust section, **16.**6
  factory fabricated or field-built AHU, **16.**3
  fans, **16.**4–**16.**5
  filters, **16.**5
  functions of, **16.**1–**16.**2
  horizontal or vertical unit, **16.**2
  humidifiers, **16.**5–**16.**6
  mixing, **16.**6–**16.**7
  outdoor air intake, **16.**6
  outdoor air (makeup air) or mixing AHU, **16.**2
  selection, **16.**9–**16.**12
  single zone or multizone, **16.**2–**16.**3
  rooftop or indoor AHU, **16.**4
Air jets, **18.**5–**18.**11
  Archimedes number, **18.**11
  centerline velocities, **18.**8–**18.**9
  characteristic length, **18.**8
  confined, **18.**8–**18.**10
  confined, airflow pattern, **18.**9–**18.**10
  core zone, **18.**5
  entrainment ratio, **18.**7
  envelope, **18.**5
  free isothermal, **18.**5–**18.**7
  free nonisothermal, **18.**10–**18.**11
  main zone, **18.**6
  surface effect, **18.**8
  terminal zone, **18.**6
Air jets *(Cont.)*
  throw, **18.**7
  transition zone, **18.**6,
  velocity profile, **18.**6
Air movements, **4.**20–**4.**23
Air systems, **1.**6–**1.**8, **20.**2–**20.**4
  air conditioning rules, **20.**63
  air distribution system, **20.**3
  air economizer mode, **22.**5
  air-handling system, **20.**2
  classification, **20.**39
  constant volume systems, **20.**40–**20.**41
  cooling and heating mode, **22.**4
  mechanical ventilation system, **20.**3
  minimum outdoor air recirculating mode, **22.**5
  mixing-exhaust section, **22.**8
  occupied and unoccupied mode, **22.**5
  operating modes, **22.**4–**22.**5
  part-load operation, **22.**4–**22.**5
  purge-mode, **22.**5
  regenerative systems, **20.**3–**20.**4
  reheating, recooling, and mixing, **20.**74–**20.**75
  smoke control systems, **20.**4
  terminals, **20.**4
  ventilation systems, **20.**3
  warmup, colddown, and nighttime setback mode, **22.**5
Air temperature:
  comfort air conditioning systems, **4.**20–**4.**21
  indoor, **4.**20–**4.**23
  processing air conditioning systems, **4.**23
Air washer, **1.**11
Amplifiers, **2.**7
Annual energy use, HVAC&R systems, **1.**14
Artificial intelligence, **5.**45–**5.**53
Artificial neural networks (ANN), **5.**50–**5.**53
  learning method, **5.**52–**5.**53
  neuron, **5.**51
  neuron activation transfer **5.**51–**5.**52
  net topology, **5.**51
ASHRAE/IESNA Standard 90.1–1999,
  building envelope trade-off option, **3.**50
  compliance for building envelope, **3.**48–**3.**50
  controls, **5.**66–**5.**67
  off-hour controls, **5.**66–**5.**67
Atmospheric dust, **15.**61
Atmospheric extinction coefficient, **3.**26
Automated computer-aided drafting (AutoCAD), **1.**26

Bernoulli equation, **17.**2
Boilers, hot water, **8.**9–**8.**15
  cast-iron sectional, **8.**12
  chimney or stack, **8.**14
  combustion efficiency, **8.**13
  condensing and noncondensing , **8.**13
  electric, **8.**17
  fire-tube, **8.**10
  flow processes, **8.**10–**8.**12
  forced-draft arrangements, **8.**12
  gas and oil burners, **8.**13
  heating capacity control, **8.**14
  minimum efficiency requirements, **8.**13–**8.**14
  safety control, **8.**14–**8.**15
  Scotch Marine packaged boiler, **8.**10–**8.**12
  selection of fuel, **8.**9–**8.**10
  types of, **8.**10
Boiling point, **2.**4–**2.**5
Building:
  energy star, **25.**10
  green, **25.**8–**25.**10
  shell building, **3.**48
  speculative building, **3.**48
Building automation and control network (BACnet), **5.**41
Building automation systems, **5.**2
Building envelope, **3.**2
  ceiling, **3.**2
  energy-efficient and cost-effective measures, **3.**50–**3.**51
  exterior floor, **3.**2
  exterior wall, **3.**2
  fenestration, **3.**2
  partition wall, **3.**2
  roof, **3.**2
  skylight, **3.**2
  slab on grade, **3.**2
  Standard 90.1–1999, **3.**48–**3.**50
  wall below grade, **3.**2
  window, **3.**2
Building material:
  closed-cell, **3.**16
  open-cell, **3.**13
Building tightness, or building air leakage, **20.**5–**20.**6
  air change per hour at 50 Pa (ACH50), **20.**6
  effective leakage area, **20.**5
  exfiltration, **20.**14
  flow coefficient C_{flow}, in cfm/ft^2, **20.**6
  infiltration, **20.**14
  volume flow rate of infiltration, **20.**14
Campus-type water systems, **7.**53–**7.**58
  building entrance, **7.**56
  control of variable-speed distribution pump, **7.**56
  distribution pipes, **7.**58
  multiple-source distributed building loop, **7.**57–**7.**58
  plant-distributed building loop, **7.**56–**7.**57
  plant-distribution building loop, **7.**54–**7.**56
  pressure gradient of distribution loop, **7.**54
Carbon adsorbers, activated, **15.**70–**15.**71
  reactivation, **15.**71
Cascade systems, **9.**40–**9.**43
  advantages and disadvantages, **9.**40–**9.**41
  performance, **9.**42–**9.**43
Central plant, **1.**8–**1.**9
Central systems, **30.**2
  air and water temperature differentials, **30.**5–**30.**6
  control at part load, **30.**4
  controls in water, heating, and refrigerating systems, **30.**4
  floor-by-floor systems vs. air systems serving many floors, **30.**2–**30.**3
  influence of inlet vanes on small centrifugal fans, **30.5**–**30.7**
  separate air system, **30.**2–**30.**3
  size of air system, **30.**2
  types of VAV central systems, **30.**7
Central systems, clean-room, **30.**14–**30.**24
  airflow, **30.**14–**30.**16
  case-study: integrated-circuit fabrication, **30.**16–**30.**24
  design considerations, **30.**24
  effect of filter pressure drop difference on system performance, **30.**22–**30.**24
  energy use of components, **30.**17
  indoor requirements, **30.**16–**30.**17
  operating characteristics, **30.**18–**30.**19
  part-load operation and controls, **30.**19–**30.**20
  pressurization, **30.**16
  summer mode operation, **30.**19
  system characteristics, **30.**13
  system description, **30.**14–**30.**15, **30.**17–**30.**18
  system pressure, **30.**21
  temperature and relative humidities, **30.**16
  winter mode operation and controls, **30.**20–**30.**21
Central systems, dual-duct VAV, **30.**10–**30.**11
  system characteristics, **30.**8
  system description, **30.**10–**30.**11

Central systems, fan-powered VAV, **30.**11–**30.**13
case-study: Taipei World Trade Center, **30.**12–**30.**13
supply volume flow rate and coil load, **30.**11
system characteristics, **30.**13
system description, **30.**11
Central systems, single zone VAV, **30.**7–**30.**9
supply volume flow rate and coil load, **30.**7–**30.**8
system characteristics, **30.**8
system description, **30.**7
zone temperature control, **30.**8
Central systems, VAV cooling, VAV reheat, and perimeter-heating VAV, **30.**9–**30.**10
supply volume flow rate and coil load, **30.**9
system characteristics, **30.**8
system description, **30.**9
zone temperature control, **30.**10
Centrifugal chiller, **1.**12
air purge, **13.**24
auxiliary condenser, **13.**9–**13.**11
capacity control, **13.**19–**13.**21
capacity control by variable speed, **13.**20
capacity control using inlet vanes, **13.**20
chilled water leaving temperature control, **13.**22
comparison between inlet vanes and variable speed, **13.**21
condenser water temperature control, **13.**23
controls, **13.**22–**13.**24
difference between centrifugal compressors and fans, **13.**19
double-bundle condenser, **13.**9–**13.**10
evaporating and condensing temperatures at part-load, **13.**26–**13.**27
faults detection and diagnostics, **13.**24
functional controls and optimizing controls, **13.**22
incorporating heat recovery, **13.**9–**13.**13
operating characteristics, **13.**24–**13.**35
operating modes, **13.**9–**13.**11
part-load operation, **13.**25–**13.**27
part-load operation characteristics, **13.**25–**13.**26
performance rating conditions, **13.**8–**13.**9
refrigerant flow, **13.**7–**13.**8
required system head at part-load operation, **13.**19–**13.**20
safety controls, **13.**23–**13.**24
sequence of operations, **13.**24–**13.**25
short-cycling protection, **13.**23
surge protection, **13.**24
Centrifugal chiller *(Cont.)*
system balance at full load, **13.**25
system characteristics, **13.**12–**13.**13
system description, **13.**9
temperature lift at part-load, **13.**29–**13.**31
water-cooled, **13.**7–**13.**9
Centrifugal chiller, multiple-chiller plant, **13.**33–**13.**36
chiller staging, **13.**34
design considerations, **13.**35–**13.**36
parallel and series piping, **13.**33–**13.**34
Standard 90.1–1999 minimum efficiency requirements, **13.**35
Centrifugal compressor:
performance map,**13.**15–**13.**18
surge of, **13.**15–**13.**16
Centrifugal compressor map:
at constant speed, **13.**16–**13.**18
at variable speed, **13.**17–**13.**18
Centrifugal pumps, **7.**30–**7.**34
cavitation, **7.**33
net positive suction head (NPSH), **7.**33
net static head, **7.**32
performance curves, **7.**32–**7.**33
pump efficiency, **7.**32
pump power, **7.**32
selection, **7.**33–**7.**34
total head, **7.**30–**7.**32
volume flow, **7.**30
Centrifugal refrigeration systems, **13.**1–**13.**7
compressor, **13.**3–**13.**4
free refrigeration, **13.**31–**13.**33
free refrigeration, principle of operation, **13.**31–**13.**32
free refrigeration capacity, **13.**32–**13.**33
purge unit, **13.**5–**13.**7
refrigerants, **13.**2–**13.**3
system components, **13.**4–**13.**5
Chilled-water storage systems, stratified, **31.**18–**31.**23
basics, **31.**18–**31.**19
case-study, **31.**23–28
charging and discharging, **31.**18, **31.**26–**31.**27
charging and discharging temperature, **31.**22–**31.**23
chilled water storage system, **31.**23–**31.**25
concentric double-octagon diffusers, **31.**24–**31.**26
diffusers, **31.**20–**31.**22
figure of merit, **31.**19
inlet Reynolds number, **31.**21–**31.**22
part-load operation, **31.**27–**31.**28

Chilled-water storage systems, stratified *(Cont.)*
self-balancing, **31.**22
storage tanks, **31.**19
stratified tanks, **31.**19–**31.**20
system characteristics, **31.**10
system description, **31.**18
system performance, **31.**28
thermocline and temperature gradient, **31.**20–**31.**21
Chlorofluorocarbons (CFCs), **1.**12
Clean room, **4.**31
Clean space, **4.**31
Clearness number of sky, **3.**26
Clothing:
efficiency, **4.**6
insulation, **4.**7
CLTD/SCL/CLF method of cooling load calculation, **6.**26–**6.**32
exterior walls and roofs, **6.**26–**6.**28
fenestration, **6.**28
infiltration, **6.**31
internal loads, **6.**29–**6.**31
night shutdown mode, **6.**32
wall exposed to unconditioned space, **6.**28–**6.**29
Codes and standards, **1.**23–**1.**25
Cogeneration, **12.**25–**12.**26
using a gas turbine, **12.**28–**12.**29
Coil accessories, **15.**56–**15.**57
air stratification,**15.**58–**15.**59
air vents, **15.**56
coil cleanliness, **15.**57
coil freeze protection, **15.**58–**15.**60
condensate collection and drain system, **15.**57–**15.**58
condensate drain line, **15.**58
condensate trap, **15.**58
drain pan, **15.**58
Coil characteristics, **15.**32–**15.**39
coil construction parameters, **10.**3–**10.**4
contact conductance, **15.**37–**15.**39
direct-expansion (DX), **15.**33
fins, **15.**33–**15.**37
interference, **15.**38
steam heating, **15.**33
types of, **15.**33–**15.**34
water circuits, **15.**38–**15.**39
water cooling, **15.**33
water heating, **15.**33
Coils, DX (wet coils), **10.**2–**10.**10
(*See also* DX coils)
Coils, sensible cooling and heating (dry coils), **15.**39–**15.**48
Chilton-Colburn j-factor, **15.**41
Coils, sensible cooling and heating (dry coils) *(Cont.)*
effectiveness ε, **15.**42
fin efficiency η_f, **15.**41–**15.**42
fin surface efficiency η_s, **15.**41
fluid velocity and pressure drop, **15.**44
heat transfer in sensible cooling process, **15.**39–**15.**41
heating coils, **15.**44
JP parameter, **15.**41
number of transfer units (NTU), **15.**43
part-load operation, **15.**44
surface heat transfer coefficients, **15.**41–**15.**42
Coils, water cooling (dry-wet coils), **15.**48–**15.**52
dry-part, **15.**50
dry-wet boundary, **15.**48–**15.**49
part-load operation, **15.**50–**15.**51
selection, **15.**51–**15.**52
wet-part, **15.**50
Cold air distribution, **18.**28–**18.**30
case-study, Florida Elementary School, **18.**29
characteristics, **18.**29
vs. conventional air distribution, **18.**28
with fan-powered VAV boxes, **18.**30
high induction nozzle diffusers, **18.**28–**18.**29
performance of ceiling and slot diffusers, **18.**29–**18.**30
surface condensation, **18.**30
Commissioning, **32.**1
cost of HVAC&R commissioning, **32.**5
necessity of HVAC&R commissioning, **32.**1–**32.**2
scope of, **32.**2–**32.**3
team of HVAC&R commissioning, **32.**4
when to perform, **32.**4–**32.**5
Compound systems with flash cooler:
coefficient of performance, **9.**33, **9.**38
coil core surface area F_s, **15.**40
enthalpy of vapor mixture, **9.**32–**9.**33
flow processes, **9.**31
fraction of evaporated refrigerant in flash cooler, **9.**31–**9.**32, **9.**35–**9.**37
three-stage, **9.**35–**9.**38
two-stage, **9.**31–**9.**33
Compound system with vertical intercooler, two-stage, **9.**38–**9.**40
comparison between flash coolers and intercoolers, **9.**40
Compressibility factor, **2.**2–**2.**3
Compressors, reciprocating, **11.**5

Computational fluid dynamics (CFD), **18.**51–**18.**54
conducting CFD experiments, **18.**54
numerical methods, **18.**52–**18.**53
Reynolds-averaged Navier-Stokes equations, **18.**52
velocity vectors of the airflow in a duct section, **18.**53
Computer-aided design, **1.**25–**1.**26
Computer-aided design and drafting (CADD), **1.**25–**1.**26
Computer-aided design and interface, **17.**73
Computer-aided drafting, **1.**26
Computer-aided duct design and drafting, **17.**72–**17.**73
Computer-aided duct drafting, **17.**72
Computer-aided running processes of duct system, **19.**73
Computer-aided schedules and layering, **17.**72–**17.**73
Computer-aided piping design and drafting, **7.**58–**7.**60
computer-aided design capabilities, **7.**59–**7.**60
computer-aided drafting capabilities, **7.**58–**7.**59
input data and reports, **7.**60
pressure losses and network technique, **7.**59
pump and system operations, **7.**59
system and pipe size, **7.**59
Condensation:
in buildings, **3.**17–**3.**18
concealed condensation in building envelopes, **3.**18
visible surface, **3.**17–**3.**18
Condensation process, **10.**20–**10.**21
heat rejection factor, **10.**21–**10.**22
total heat rejection, **10.**21–**10.**22
Condensers, **10.**20–**10.**36
automatic brush cleaning for, **13.**13–**13.**15
effect of brush cleaning system,**13.**14–**13.**15
principle and operation, **13.**13–**13.**14
type of, **10.**22
Condensers, air-cooled, **10.**26–**10.**30
clearance, **10.**29
condenser temperature difference, **10.**28–**10.**29
condensing temperature, **10.**29,
construction, **10.**26–**10.**28
cooling air temperature rise, **10.**28
dirt clogging, **10.**29
heat transfer process, **10.**26–**10.**28
low ambient control, **10.**29–**10.**30
Condensers, air-cooled *(Cont.)*
oil effect, **10.**29
selections, **10.**30
subcooling, **10.**29
volume flow, **10.**28
warm air circulation, **10.**29
Condensers, evaporative, **10.**30–**10.**33
condensation process, **10.**30
cooling air, **10.**32
heat transfer, **10.**30–**10.**32
low ambient air control, **10.**33
selection and installations, **10.**33
site location, **10.**32–**10.**33
water spraying, **10.**32
Condensers, water-cooled, **10.**22–**10.**26
capacity, **10.**26
double-tube condenser, **10.**22–**10.**23
effect of oil, **10.**25
heat transfer, **10.**24–**10.**25
part-load operation, **10.**26
performance, **10.**25–**10.**26
shell-and-tube condensers, **10.**22–**10.**25
subcooling, **10.**25
types of, **10.**22
Conduit induction system, **1.**11
Constant-volume multizone system with reheat, **20.**74–**20.**78
control systems, **20.**75–**20.**76
operating parameters and calculation, **20.**76–**20.**78
reheating, recooling and mixing, **20.**74–**20.**75
system characteristics, **20.**78
Constant-volume single-zone systems, cooling mode operation, **20.**53–**20.**59
air conditioning cycle, cooling mode operation, **20.**53–**20.**54
cooling mode operation in summer, **20.**53–**20.**56
cooling mode operation in winter with space humidity control, **20.**55–57
cooling mode operation in winter without space humidity control, **20.**55–57
outdoor ventilation air and exhaust fans, **20.**58–**20.**59
part-load operation and controls, **20.**58
two-position or cycling control, **20.**58
water flow rate modulation, **20.**58
Constant-volume single-zone systems, heating mode operation, **20.**69–**20.**74
dual-thermostat, year-round zone temperature control, **20.**73–**20.**74
heating mode with space humidity control, **20.**71–**20.**73

Constant-volume single-zone systems, heating mode operation *(Cont.)*
heating mode without space humidity control, **20.**69–**20.**70
part-load operation, **20.**73
Constant-volume systems, **20.**40–**20.**41
energy per unit volume flow, **20.**41
system characteristics, **20.**40–**20.**41
Control loop, **5.**5
closed, **5.**5
open, **5.**5
Control medium, **5.**11
Control methods, **5.**7–**5.**9
comparison of, **5.**8–**5.**9
direct-digital-control (DDC), **5.**7
electric or electronic control, **5.**7–**5.**8
pneumatic control, **5.**7
Control modes, **5.**9–**5.**16
compensation control or reset, **5.**15
differential, **5.**9
floating control, **5.**11
modulation control, **5.**10
offset or deviation, **5.**13
proportional band, **5.**12
proportional control, **5.**11–**5.**13
proportional-integral-derivative (PID) control, **5.**14–**5.**15
proportional plus integral (PI) control, **5.**13–**5.**14
step-control, **5.**10–**5.**11
throttling range, **5.**12
two-position, **5.**9–**5.**10
Control systems, **5.**2
direct digital control (DDC), **1.**9
dual-thermostat year-round zone temperature control, **20.**73–**20.**74
Control valves, **5.**26–**5.**31,
actuators, **5.**26–**5.**27
equal-percentage, **5.**28
flow coefficient, **5.**31
linear, **5.**28
quick-opening, **5.**29
rangeability, **5.**29
three-way, **5.**27
two-way, **5.**27
Controlled device, **5.**5
Controlled variable, **5.**2
Controllers, **5.**21–**5.**26
direct-acting and reverse-acting, **5.**21–**5.**22
direct digital, **5.**23–**5.**26
electric and electronic, **5.**23
electric erasable programmable read-only memory (EEPROM), **5.**24
Controllers *(Cont.)*
flash erasable programmable read-only memory (flash EPROM), **5.**25
normally closed or normally open, **5.**22
pneumatic, **5.**22–**5.**23
random-access memory (RAM), **5.**24
read-only memory (ROM), **5.**23
system, **5.**23–**5.**26, **5.**38–**5.**39
unit, **5.**23–**5.**26, **5.**39
Controls:
alarming, **5.**60
discriminator, **5.**60
functional, **5.**58–**5.**61
generic, **5.**59–**5.**60
graphical displays, **5.**59
scheduling, **5.**59–**5.**60
specific, **5.**60–**5.**61
trending, **5.**59
Cooling coil load, **6.**32–**6.**34
duct heat gain, **6.**33
fan power, **6.**33
temperature of plenum air, **6.**34
ventilation load, **6.**34
Cooling coil load, components, **6.**7–**6.**8
Cooling load:
components, **6.**6–**6.**7
external, **6.**7
internal, **6.**7
Cooling load calculations:
historical development, **6.**11–**6.**12
heat balance, **6.**12–**6.**14
transfer function, **6.**14–**6.**16
Cooling media, **9.**3
Cooling towers, **10.**34–**10.**36
approach, **10.**36, **10.**41
blowdown, **10.**36
construction materials, **10.**43
counterflow forced draft, **10.**35–**10.**36
counterflow induced draft, **10.**34–**10.**35
crossflow induced draft, **10.**34–**10.**35
factors affecting performance, **10.**40
fill configuration, **10.**42–**10.**43
heat and mass transfer process, **10.**37–**10.**39
makeup, **10.**36
optimum control, **10.**43–**10.**44
outdoor wet-bulb temperature, **10.**41
part-load operation, **10.**43
performance, **10.**40–**10.**43
range, **10.**36, **10.**40
thermal analysis, **10.**36–**10.**39
tower capacity, size, **10.**37–**10.**39
tower coefficient (NTU), **10.**36–**10.**39, **10.**41
water-air ratio, **10.**41

Cooling towers *(Cont.)*
water circulating rate, **10.**40
water distribution, **10.**43
Cooling towers, operating considerations, **10.**46–**10.**48
blowdown, **10.**47
fogging, **10.**46–**10.**47
freeze protection, **10.**46
interference, **10.**46
Legionnaires' disease, **10.**47
maintenance, **10.**47–**10.**48
recirculation, **10.**46
Coordination, **1.**19
Copenhagen Amendments and Vienna Meeting, **9.**10–**9.**11
Corrosion, **7.**25

Daily range, mean, **4.**39
Dalton's law, **2.**3–**2.**4
Dampers, **5.**32–**5.**38
actuators, **5.**33
butterfly, **5.**32
characteristic ratio, **5.**35–**5.**37
gate, **5.**32
opposed-blade, **5.**33, **5.**35–**5.**37
parallel-blade, **5.**33 , **5.**35–**5.**37
sizing, **5.**37–**5.**38
split, **5.**32–33
DDC programming, **5.**53–**5.**55
evolution, **5.**53
graphical, **5.**53–**5.**54
for mechanical cooling control, **5.**55
templates, **5.**54
DDC tuning controllers, **5.**55–**5.**56
adaptive control, **5.**56
PI controllers, **5.**55
self-tuning, **5.**55
Degree days:
cooling with a base temperature of 50 °F, **4.**39
heating with a base temperature of 65 °F, **4.**39
number of, **4.**39
Degree of saturation, **2.**8
Demand-controlled ventilation (DCV), CO_2-based, **23.**5–**23.**12
application of, **23.**11–**23.**12
ASHRAE Standard 62–1999, **23.**7
base ventilation, **23.**9–**23.**10
CO_2-based DCV system, **23.**10–**23.**11
CO_2 sensor or mixed-gases sensor, **23.**7
location of CO_2 sensor, **23.**7–**23.**8
minimum outdoor air recirculation mode, **23.**6
Demand-controlled ventilation (DCV), CO_2-based *(Cont.)*
purge mode, **23.**10
substantial lag time in space CO_2 concentration dilution process, **23.**8–**23.**8
vs. time-based constant-volume control, **23.**5–**23.**6
Depletion of the ozone layer, **1.**15
Desiccant-based air conditioning systems, **29.**22–**29.**27
applications, **29.**34–**29.**35
conditions to apply, **29.**34–**29.**35
desiccant dehumidification and sensible cooling, **29.**22–**29.**24
desiccants, **29.**24–**29.**26
lithium chloride, **29.**26
molecular sieves, **29.**26–**29.**27
rotary desiccant dehumidifiers, **29.**27
silica gel, **29.**26
system characteristics, **29.**21
Desiccant-based air conditioning systems, for operating rooms, **29.**32–**29.**34
indoor environment, **29.**32–**29.**33
system description, **29.**33–**29.**34
Desiccant-based air conditioning systems, for retail store, **29.**31–32
operating characteristics, **29.**31–**29.**32
performance, **29.**32
system description, **29.**31–**29.**32
Desiccant-based air conditioning systems, for supermarket, **29.**27–**29.**31
air conditioning cycle, **29.**30–**29.**31
gas heater, **29.**30
heat-pipe heat exchanger, **29.**29–**29.**30
indirect evaporative cooler, **29.**30
loads in supermarkets, **29.**27
operating parameters in rotary desiccant dehumidifier, **29.**29
part-load operation and controls, **29.**31
refrigeration, **29.**30,
space conditioning line, **29.**28–**29.**29
system description, **29.**25, **29.**28
of the control systems, **1.**20–**1.**21
Design
documents, **1.**21–**1.**22
Design-bid, **1.**17
Design-build, **1.**17
Design intent, **32.**1
Desorption isotherm, **3.**11
Diagram:
pressure-enthalpy, **9.**17–**9.**18
temperature-entropy, **9.**18–**9.**19
Direct expansion (DX) coil, **1.**4

Discharge air temperature controls, **23.**18–**23.**23
basics, **23.**18
discharge air temperature reset, **23.**22–**23.**23
operation of air economizer, **23.**21–**23.**22
outdoor air intake, **23.**21–**23.**22
system description, **23.**19–**23.**21
Distribution of systems usage, **1.**10
Diversity factor, **1.**20
Drawings, **1.**22
air duct diagram, **1.**22
control diagrams, **1.**22
detail, **1.**22
equipment schedule, **1.**22
floor plans, **1.**22
legends, **1.**22
piping diagram, **1.**22
sections and elevations, **1.**22
Duct cleaning, **17.**74–**17.**75
Duct construction, **17.**12–**17.**18
duct hanger spacing, **17.**17
fiberglass ducts, **17.**18
flame speed and smoke developed, **17.**13
flat oval ducts, **17.**17–**17.**18
flexible ducts, **17.**18
material, **17.**12–**17.**13
maximum pressure difference, **17.**12
rectangular ducts, **17.**13
rectangular metal duct construction, **17.**15
round ducts, **17.**17
thickness of galvanized sheets, **17.**14, **17.**17
transverse joint reinforcement,**17.**16
Duct friction losses, **17.**22–**17.**31
absolute and relative roughness, **17.**22–**17.**24
circular equivalents, **17.**27–**17.**31
Colebrook formula, **17.**24
Darcey-Weisbach equation, **17.**22
duct friction chart, **17.**24–**17.**26 **17.**25–**17.**26
duct roughness, **17.**25
friction factor, **17.**22–**17.**24
Moody diagram, **17.**22–**17.**23
roughness and temperature corrections, **17.**25
Rouse limit, **17.**24
Swamee and Jain formula, **17.**24
Duct insulation, **17.**19–**17.**22
duct insulation by ASHRAE Standard 90.1–1999,**17.**19–**17.**21
temperature rise and drop, **17.**19
temperature rise curves, **17.**21–**17.**22
Duct liner, **17.**74
Duct sizing methods, **17.**53–**17.**56
constant velocity method, **19.**53–**19.**54
equal friction method, **17.**53
Duct sizing methods *(Cont.)*
static regain method, **17.**54–**17.**55
T-method, **17.**55–**17.**56
Duct static pressure and fan controls, **23.**23–**23.**26
comparison between adjustable-frequency drives and inlet vanes, **23.**24–**23.**26
duct static pressure control, **23.**23–**23.**24
sensor's location, **23.**24
set point, **23.**24
Duct systems with certain pressure losses in branch takeoffs, **17.**56–**17.**66
condensing two duct sections, **17.**59–**17.**60
cost optimization, **17.**56–**17.**59
design characteristics, **17.**56
local loss coefficients for diverging tees and wyes, **17.**60–**17.**62
return or exhaust duct systems, **17.**63
Duct systems with negligible pressure loss at branch ducts, **17.**66–**17.**72
local loss coefficients, **17.**68–**17.**69
pressure characteristics of airflow in supply ducts, **17.**66–**17.**68
rectangular supply duct with transversal slots, **17.**67
return or exhaust duct systems, **17.**71–**17.**72
supply duct systems, **17.**66
DX coils, wet coils, **10.**2–**10.**10
air-side pressure drop, **10.**8
construction and installation, **10.**3–**10.**4
DX coil effectiveness, **10.**6–**10.**7
face velocity, **10.**7–**10.**8
part-load operation, **10.**8–**10.**10
selection of DX coils, **10.**10
simultaneous heat and mass transfer, **10.**5–**10.**6
superheated region, **10.**5
two-phase region, **10.**4–**10.**5
two-region model,**10.**4–**10.**5
Dynamic losses, **17.**31–**17.**38
converging and diverging tees and wyes, **17.**34–**17.**37
elbows, **17.**31–**17.**34
entrances, exits, enlargements, and contractions, **17.**38

Earth-sun distance, **3.**25
Economizer cycle, economizers, and economizer control, **21.**8–**21.**16
air economizers, **21.**8
ANSI/ASHRAE Standard 90.1–1999 economizer control specifications, **21.**14–**21.**16

Economizer cycle, economizers, and economizer control *(Cont.)*
comparison of air and water economizers, **21.**14
comparison of enthalpy-based and temperature-based, **21.**10–**21.**12
differential enthalpy, electronic enthalpy, and fixed enthalpy, **21.**8–**21.**9
enthalpy (-based) economizer control, **21.**8–**21.**9
fixed dry-bulb and differential dry bulb, **21.**9–**21.**10
sequence of operations of a differential dry-bulb, **21.**10
sequence of operations of a differential enthalpy, **21.**9
water economizer, **21.**8,
water economizer control, **21.**12–**21.**14
Effective temperature, **4.**14
Electric heating fundamentals, **8.**15–**8.**16
electric duct heaters, **8.**17
electric furnaces and electric heaters, **8.**16–**8.**17
Electricity deregulation, **25.**14–**25.**15
California approach, **25.**15
case-study: automatic control of RTP, **25.**16–**25.**17
prior to deregulation, **25.**14
real-time pricing (RTP), **25.**15–**25.**16
Energy conservation measures, **25.**10–**25.**11
case-study-for an office, **25.**12
Energy cost budget method, ASHRAE/IESNA Standard 90.1–1999, **25.**28
Energy efficiency, **1.**13–**1.**15, **25.**1–**25.**2, **25.**5 - **25.**10
during design, construction, commissioning, and operation, **25.**2
energy audits, **25.**6
energy retrofits, **25.**6–**25.**7
energy service companies (ESCOs), **25.**7
federal mandates, **25.**5
performance contracting, **25.**7–**25.**8
reduction of unit energy rate, **25.**2–**25.**3
Energy management and control systems (EMCS), **5.**3
Energy management systems, **5.**3
Energy use (energy consumption), **1.**13–**1.**15, **25.**1–**25.**2
between HVAC&R system characteristics, **25.**12–**25.**13
building energy consumption and thermal storage systems, **31.**2
fan, motor, and drive combined efficiency, **25.**13–**25.**14
Energy use (energy consumption) *(Cont.)*
heating-cooling equipment, **25.**13
Energy use, index, **9.**55–**9.**55
energy efficiency ratio (EER), **9.**55
energy use intensities, **25.**5–**25.**6
heating season performance factor (HSPF), **9.**55
integrated part-load value (IPLV), **9.**56
kW/ton, **9.**55–**9.**56
seasonal energy efficiency ratio (SEER), **9.**56
Engineering responsibilities, **1.**18–**1.**19
Engineer's quality control, **1.**20
Environment:
cleanest, **1.**13
most precise, **1.**13
quietest, **1.**13
Environmental problems, **1.**15
Equation of state:
of an ideal gas, **2.**2
of a real gas, **2.**2
Evaporative coolers, add-on, **27.**18–**27.**24
indirect-direct cooler to a DX packaged system, **27.**18–**27.**20
tower and coil combination, **27.**22–**27.**23
tower coil and rotary wheel combination, **27.**20–**27.**22
Evaporative cooling, **27.**1
air washers, **27.**4
direct, **27.**2
direct evaporative coolers, **27.**3–**27.**4
evaporative pads, **27.**4
operating characteristics, **27.**6
rigid media, **27.**4
rotary wheel, **27.**4–**27.**6
saturation efficiency, **27.**2–**27.**4
Evaporative cooling, indirect, **27.**6–**27.**13
effectiveness, **27.**10–**27.**11
heat transfer process, **27.**7–**27.**10
operating characteristics, **27.**11–**27.**12
part-load operation and control, **27.**12–**27.**13
process, **27.**6
Evaporative cooling, indirect-direct two-stage systems, **27.**13–**27.**18
case study: Nevada's College, **27.**16–**27.**18
energy efficiency ratio and energy use intensities, **27.**16
indirect-direct two-stage evaporative cooler, **27.**13–**27.**15
system characteristics, **27.**17–**27.**18
using outdoor air as cooled and wet air, **27.**15
using return air as wet air and outdoor-return air mixture as cooled air, **27.**15–**27.**16

Evaporative cooling systems, **27.**1–**27.**2
beware of dampness, sump maintenance, and water leakage, **27.**24
design considerations, **27.**24–**27.**26
scope of applications, **27.**24
selection of summer outdoor design conditions, **27.**24–**27.**26
Evaporative heat loss, **4.**7–**4.**9
diffusion, **4.**8–**4.**9
maximum, **4.**7–**4.**8
due to regulatory sweating, **4.**7–**4.**8
respiration losses, **4.**7
from skin surface, **4.**7
Evaporators, **10.**2–**10.**20
air-cooler, **10.**2
circulating rate, **10.**20
counterflow or parallel flow, **10.**20
direct-expansion liquid cooler, **10.**18
down-feed or up-feed, **10.**20
DX coil (wet coils) **10.**2–**10.**10
flooded liquid cooler, **10.**12–1020
liquid cooler, **10.**2
liquid overfeed cooler, **10.**18–**10.**20
mechanical pump or gas pump, **10.**20
Energy, **9.**19
Expansion tank:
closed, **7.**21
diaphragm, **7.**21–23
fill pressure, **7.**21
open, **7.**20–**7.**21
water logging, **7.**24–**7.**25

Factors affecting control processes, **5.**56–**5.**58
climate change, **5.**56–**5.**57
disturbance, **5.**57
intermittent operation, **5.**57
load, **5.**56
performance of control processes, **5.**57–**5.**58
system capacity, **5.**57
thermal capacitance, **5.**58
turndown ratio, **5.**57
Fan capacity modulation, **15.**20–**15.**24
ac inverter, **15.**20–**15.**21
adjustable pitch, **15.**24
blade pitch, **15.**24
controllable pitch, **15.**24
fan speed with adjustable frequency drives, **15.**20–**15.**21
inlet cone, **15.**23–**15.**24
inlet-vanes, **15.**21–**15.**23
pulse-width-modulated inverter, **15.**21
variable-speed drives (VSDs), **15.**20–**15.**21
Fan coil, **1.**5
Fan coil systems, **28.**3–**28.**5
operating characteristics, **28.**3–**28.**5
system description, **28.**3
Fan coil systems, four-pipe, **28.**9–**28.**15
chilled water supplied to coils, **28.**11–**28.**12
dedicated ventilation system, **28.**10–**28.**11
exhaust air to balance outdoor ventilation air, **28.**12
general description, **28.**9–**28.**10
operating parameters, **28.**14–**28.**19
part-load operation, **28.**13
space recirculation systems, **28.**11
system characteristics, **28.**14–**28.**15
zone temperature control and sequence of operations, **28.**13–**28.**14
Fan coil systems, two-pipe, **28.**20–**28.**24
applications, **28.**24
changeover two-pipe systems, **28.**23–**28.**24
nonchangeover two-pipe systems, **28.**20–**28.**23
system characteristics, **28.**15
Fan coil units, **28.**5–**28.**9
coils, **28.**7
cooling and dehumidifying, **28.**8–**28.**9
fan, **28.**6–**28.**7
filters, **28.**7
sound power level, **28.**9
volume flow rate, **28.**7–**28.**8
Fan combinations, **22.**4
operating modes, **22.**4–**22.**5
Fan combinations, supply and exhaust fans, **22.**8–**22.**14
air-economizer mode, **22.**13
operating characteristics, **22.**9–**22.**10
pressure variation at the mixing box, **22.**13–**22.**14
recirculating mode and design volume flow rate, **22.**9–**22.**12
recirculating mode, 50% design flow rate, **22.**12–**22.**13
system characteristics, **22.**8–**22.**9
warmup and colddown mode, **22.**13
Fan combinations, supply and relief fans, **22.**14–**22.**18
air economizer mode and design volume flow rate, **22.**14–**22.**16
air economizer mode, 50% design flow, **22.**17
design considerations and controls, **22.**17–**22.**18
recirculating mode, **22.**14–**22.**15
warmup and cool-down mode, **22.**17
Fan combinations, supply and return fans, **22.**18–**22.**21
air economizer mode, **22.**20–**22.**21

Fan combinations, supply and return fans *(Cont.)*
comparison of three fan combination systems, **22.**21–**22.**22
controls, **22.**21
recirculating mode, **22.**18–**22.**20
Fan construction and arrangements, **15.**25–**15.**29
drive arrangements and direction of discharge, **15.**26–**15.**28
high-temperature fans, **15.**27
safety devices, **15.**28–**15.**29
sizes and class standards, **15.**25–**15.**26
spark-resistant construction, **15.**28
width and inlets, **15.**26–**15.**27
Fan-duct systems, **20.**14– **20.**17
fan laws, Buckingham π method, **20.**15–**20.**17
inlet system effect, **20.**18–**20.**19
inlet system effect loss, **20.**19
inlet system effect loss coefficient, **20.**19–**20.**20
outlet system effect, **20.**20–**20.**22
outlet system effect loss coefficient, **20.**22–**20.**23
selecting fans considering system effect losses, **20.**23–**20.**24
system effect, mechanism, **20.**17,
system operating point, **20.**15
Fan-duct systems, combination, **20.**24–**20.**31
connected in series, **20.**25–**20.**26
fan combined in parallel and connected in series with a duct system, **20.**26–**20.**27
two parallel fan-duct systems with another duct system, **20.**28–**20.**30
Fan-duct systems, modulation, **20.**31–**20.**38
blade pitch variation of axial fan, **20.**35–**20.**36
modulation curve, **20.**31–**20.**32
using dampers, **20.**33
using inlet cone, **20.**34–**20.**35
using inlet vanes, **20.**34
varying fan speed, **20.**35–**20.**36
Fan energy use, criteria of Standard 90.1–1999, **17.**10–**17.**12
for constant volume systems, **17.**10–**17.**11
for VAV systems, **17.**11–**17.**12
Fan-powered VAV box, **1.**8
Fan room, **16.**24–**16.**28
isolated, **16.**24–**16.**25
layout considerations, **16.**25–**16.**28
open, **16.**24
types of, **16.**24–**16.**25
Fan selection, **15.**29–**15.**32
case-study, **15.**32
Fan selection *(Cont.)*
comparison between various type of fans, **15.**31–**15.**32
estimated fan sound power level, **15.**30–**15.**31
Fans, fundamentals, **15.**2–**15.**7
air temperature increase through fan, **15.**5
blower, **15.**2
compression ratio, **15.**2
functions, **15.**2
influence of elevation and temperature, **15.**6–**15.**7
performance curves, **15.**5–**15.**6
power and efficiency, **15.**4–**15.**5
pressure, **15.**4
types of, **15.**2–**15.**3
volume flow rate or capacity, **15.**4
Fan stall, **15.**24–**15.**25
Fan surge, **15.**24
Fans, axial, **15.**14–**15.**20
hub ratio, **15.**14–**15.**15
number of blades, **15.**20
performance curves, **15.**17–**15.**19
power-volume flow curves, **15.**18–**15.**19
pressure-volume curves, **15.**17
propeller, **15.**15
reverse operation, **15.**20
static pressure developed, **15.**17
tip clearance, **15.**20
total efficiency-volume flow curves, **15.**18–**15.**19
tube-axial, **15.**15–**15.**16
typical vane-axial fan, **15.**19–**15.**20
types of, **15.**14–**15.**16
vane-axial, **15.**15–**15.**16
velocity triangles, **15.**16–**15.**17
Fans, centrifugal, **15.**7–**15.**4
backward-curved, **15.**8–**15.**10
blades, **15.**7
blast area, **15.**8
energy losses, **15.**9
forward-curved, **15.**11–**15.**12
impeller (fan wheel), **15.**7–**15.**8
power-volume flow curves, **15.**10–**15.**11
pressure-volume curves, **15.**9
radial-bladed, **15.**10–**15.**12
roof ventilators, **15.**14
total efficiency-volume flow curves, **15.**10
total pressure increase at fan impeller, **15.**7–**15.**8
tubular or in-line, **15.**12–**15.**13
unhoused plug/plenum,**15.**12–**15.**14
velocity triangles, **15.**8
Fans, crossflow, **15.**3–**15.**4

Fault detection and diagnostics, **5.**61–**5.**65
ANN models, **3.**64
ARX models, **5.**63–**5.**64
comparison of ARX and ANN models, **5.**65
expert systems rule-based, **5.**62–**5.**63
system and component models, **5.**64
Fenestration, **3.**29–**3.**31
Fiberglass in HVAC&R systems, **19.**17–**19.**18
problems, **19.**17–**19.**18
recommendations, **19.**18
Field experience, **1.**21
Finite difference method, **6.**34–**6.**39
cooling loads, **6.**39
interior nodes, **6.**36–**6.**37
simplify assumptions, **6.**36
space air temperature, **6.**38–**6.**39
surface nodes, **6.**37–**6.**38
Flooded liquid cooler, **10.**12–**10.**20
construction, **10.**12–**10.**14
cooling capacity, **10.**17
evaporating temperature, **10.**16
fouling factor, **10.**14–**10.**15
heat transfer, **10.**14
oil effect, **10.**17
part-load operation, **10.**17–**10.**18
performance, **10.**16–**10.**17
pool boiling and force convection model, **10.**15–**10.**16
temperature difference T_{ee} - T_{el}, **10.**16–**10.**17
Flow resistance, **17.**38–**17.**43
connected in parallel, **17.**41–**17.**42
connected in series, **17.**40–**17.**41
of duct system, **17.**42–**17.**44
of Y-connection, **17.**42–**17.**43
Flow sensors, **5.**19–**5.**20
Fouling factor, **10.**14–**10.**15
Fuzzy logic, **5.**45–**5.**47
fuzzy logic controller, **5.**47
fuzzy sets, **5.**45
membership function, **5.**45
production rules, **5.**45–**5.**47

Gas cooling, **12.**25–**12.**29
engine jacket heat recovery, **12.**28
exhaust gas heat recovery, **12.**27–**12.**28
gas-engine chiller, **12.**25–**12.**27
gas engines, **12.**27
Gaseous contaminants adsorbers and chemisorbers, **24.**8–**24.**12
activated carbon adsorbers, **24.**9
chemisorption, **24.**11
chemisorption performance, **24.**11
Gaseous contaminants adsorbers and chemisorbers *(Cont.)*
granular activated carbon (GAC) applications, **24.**10–**24.**11
granular activated carbon (GAC) performance, **24.**9–**24.**10
indoor gaseous contaminants, **24.**8–**24.**9
Gibbs-Dalton law, **2.**4
Global radiation, **3.**27–**3.**28
Global warming, **1.**15, **25.**3–**25.**5
CO_2 release, **25.**4
effect, **1.**15
Kyoto Protocol, **25.**3
mitigating measures, **25.**4–**25.**5
refrigerant emissions, **25.**4–**25.**5
total equivalent warming impact, **25.**3–**25.**4
Goal to provide an HVAC&R system, **1.**17
Green buildings, **25.**8–**25.**10
basics, **25.**8–**25.**9
case-studies, **25.**9–**25.**10
green building assessment (GBA), **25.**9
Greenhouse effect, **1.**15

Heat:
convective, **6.**2
latent, **2.**10
radiative, **6.**2
sensible, **2.**10
stored, **6.**2
Heat capacity, **3.**8
Heat of sorption, **3.**12
Heat pipe heat exchangers, **12.**23–**12.**24
Heat pump, **12.**1–**12.**3
classification of, **12.**3
cycle, **12.**2–**12.**3
Heat pump systems, air-source, **12.**5–**12.**13
capacity and selection, **12.**13
compressor, **12.**6–**12.**7
controls, **12.**13
cooling mode, **12.**9
cycling loss and degradation factor, **12.**11
defrosting, **12.**12–**12.**13
heating mode, **12.**9
indoor coil, **12.**7–**12.**8
outdoor coil, **12.**8
reversing valve, **12.**7–**12.**8
Standard 90.1–1999 minimum efficiency requirements, **12.**12
suction line accumulator,**12.**8–**12.**9
system performance, **12.**9–**12.**11
Heat pump systems, ground-coupled and surface water, **12.**17–**12.**19
Heat pump systems, groundwater, **12.**13–**12.**17

Heat pump systems, groundwater *(Cont.)*
groundwater systems, **12.**14
for hospital, **12.**14–**12.**15
for residences, **12.**15–**12.**16
Standard 90.1–1999 minimum efficiency requirements, **12.**17
Heat recovery, air-to-air, **12.**19–**12.**24
comparison between various heat exchangers, **12.**24
effectiveness, **12.**19–**12.**20
fixed-plate heat exchangers, **12.**20–**12.**21
heat pipe heat exchangers, **12.**23–**12.**24
rotary heat exchangers, **12.12.**21–**12.**23
runaround coil loops,**12.**21
types of, **12.**19
Heat recovery systems, **12.**3–**12.**5
heat balance and building load analysis, **12.**4–**12.**5
Heat rejecting systems, **10.**48–**10.**51
comparison between various systems, **10.**48–**10.**50
Standard 90.1–1999, **10.**50–**10.**51
types of, **10.**48
Heat transfer:
conductive, **3.**3–**3.**4
convective, **3.**4–**3.**5
fundamentals, **3.**2
overall, **3.**6–**3.**7
radiant, **3.**5–**3.**6
Heat transfer coefficients, **3.**8–**3.**11
forced convection, **3.**9
natural convection, **3.**10
radiant, **3.**8–**3.**9
surface, **3.**10–**3.**11, **4.**5
Heating load, **6.**39–**6.**42
basic principles, **6.**39
heat loss from products, **6.**41
infiltration, **6.**41
latent heat loss, **6.**41
night shutdown operation, **6.**41–**6.**42
pickup load and oversizing factor, **6.**42
setback, night, **6.**41–**6.**42
transmission loss, **6.**38–**6.**40
unheated spaces, **6.**40–**6.**41
Heating systems, **8.**1–**8.**2
control and operations of multizones, **8.**30–**8.**31
design considerations, **8.**30
design nomograph, **8.**30
low-pressure ducted warm air, **8.**17–**8.**22
radiant floor panel, **8.**27–**8.**31
selection of, **8.**2
system characteristics, **8.**31
thermal characteristics of floor panel, **8.**28–**8.**29
Henry's equation, **7.**23
Hot water heating systems:
design considerations, **8.**25–**8.**26
finned-tube heaters, **8.**24–**8.**25
part-load operation and control, **8.**26
two-pipe individual loop, **8.**23–**8.**24
types of, **8.**23
using finned-tube heaters, **8.**23–**8.**26
Humidifiers, **15.**72–**15.**85
humidifying load, **15.**72–**15.**73
selection and design, **15.**83–**15.**84
space relative humidity, **15.**72
types of, **15.**73
Humidifiers, atomizing and wetted element, **15.**76–**15.**78
air washers, **15.**79–**15.**82
bypass control, **15.**81
characteristics, **15.**82–**15.**83
construction of air washer, **15.**79–**15.**80
case study: White Plains ultrasonic project, **15.**77
centrifugal atomizing, **15.**77–**15.**78
functions of air washer, **15.**80
humidification process, **15.**76
oversaturation, **15.**81
performance of air washer, **15.**80–**15.**81
pneumatic atomizing, **15.**78
single-stage or multistage, **15.**81–**15.**82
ultrasonic, **15.**77
wetted element, **15.**78
Humidifiers, steam and heating element, **15.**73–**15.**76
characteristics and requirements, **15.**76
heating element, **15.**75
steam grid, **15.**73–**15.**74
steam humidifiers with separators, **15.**74–**15.**75
Humidity:
comfort air conditioning systems, **4.**23–**4.**24
process air conditioning systems, **4.**24
Humidity ratio, **2.**7
Humidity sensors, **5.**18–**5.**19
HVAC&R industry, **1.**15
h-w chart, **2.**19
Hygrometers:
capacitance, **2.**17–**2.**18
Dunmore resistance, **2.**16–**2.**17
electronic, **2.**16–**2.**17
ion-exchange resistance, **2.**16–**2.**17
mechanical, **2.**16
Hysteresis, **3.**11–**3.**12

Ice point, **2.**4–**2.**5

Ice storage systems:
comparison of various systems, **31.**17–**31.**18
types of, **31.**5
Ice storage systems, encapsulated, **31.**13–**31.**15
charging and discharging, **31.**15
chiller priority and storage priority, **31.**15
controls, **31.**14–**31.**15
encapsulated ice containers, **31.**13
location of chiller and storage tank, **31.**14
system characteristics, **31.**10
Ice storage systems, ice-harvesting, **31.**15–**31.**17
chiller operation, **31.**17
ice making or charging, **31.**16–**31.**17
system characteristics, **31.**10
system description, **31.**15–**31.**16
Ice storage systems, ice-on-coil, external melt, **31.**10–**31.**13
case-study, **31.**13
ice builders, **31.**11
ice-charging control, **31.**11
refrigeration feed, **31.**1
system characteristics, **31.**10, **31.**11–**31.**13
system description, **31.**10–**31.**11
Ice storage systems, ice-on-coil, internal melt, **31.**6–**31.**10
brine and glycol solution, **31.**6–**31.**7
case-study: operation modes, **31.**7–**31.**8
direct cooling, **31.**9
ice-burning or ice melting, **31.**9
ice-charging or ice making, **31.**8
ice storage tank, **31.**7–**31.**8
on-peak, **31.**9
system characteristics, **31.**9–**31.**10
system description, **31.**6
Indicator, **2.**6
Indoor air contaminants, **4.**27–**4.**28
bioaerosols, **4.**28
combustion products, **4.**28
nicotine, **4.**28
occupant-generated contaminants, **4.**28
radon, **4.**28
total particulates concentration, **4.**28
volatile organic compounds, **4.**28
Indoor air quality (IAQ), **4.**27
acceptable, **4.**29
basic strategies to improve, **4.**29
IAQ problems, **24.**1–**24.**2
IAQ procedure, **4.**29
ventilation rate procedure, **4.**29–**4.**31
Indoor design conditions, **4.**1–**4.**2
Infrared heaters:
electric, **8.**32–**8.**33
gas, **8.**32
Infrared heating, **8.**31–**8.**35
basics, **8.**31–**8.**32
beam radiant heaters, **8.**32
design and layout, **8.**33–**8.**35
Insufficient communication, **1.**17
Insulation material, **3.**19
moisture content, **3.**19–**3.**21
Interoperability, **5.**41
system integration, **5.**41

Knowledge-based systems (KBS), **5.**47–**5.**51
development of KBS, **5.**49
expert-systems, **5.**47–**5.**51
knowledge acquisition, **5.**49
knowledge-base, **5.**48
inference engine, **5.**48
testing, verification, and validation, **5.**49
user interface, **5.**48–**5.**49

Legal responsibility for IAQ cases, **24.**13–**24.**15
HVAC&R engineer, **24.**14–**24.**15
sick building syndrome or IAQ cases, **24.**13
who is legally responsible, **24.**13–**24.**14
Legionnaires' disease, **10.**47
Liquid absorbents, **9.**3
Lithium-bromide solution, properties of, **14.**3–**14.**6
enthalpy-concentration diagram, **14.**5–**14.**6
equilibrium chart, **14.**4
mass balance in solution, **14.**3
vapor pressure, **14.**3–**14.**4
Load:
block, **6.**9–**6.**10
coil, **6.**3
DX coil, **6.**3
heating coil, **6.**3
peak load, **6.**9–**6.**10
profile, **6.**9
refrigeration, **6.**3
space cooling, **6.**3
Load calculation method:
CLTD/SCL/CLF method, **6.**15, **6.**26–**6.**31
finite difference, **6.**34–**6.**39
TETD/TA method, **6.**15–**6.**16
transfer function (TFM), **6.**14–**6.**26
Load ratio, **5.**13

Machinery room, refrigerating, **9.**58–**9.**59
Maintenance, HVAC&R, **32.**5–**32.**6
contractors and personnel, **32.**5–**32.**6

Maintenance, HVAC&R *(Cont.)*
fault detection and diagnostics assisting predictive maintenance, **32.**6
Maintenance to guarantee IAQ, **24.**12–**24.**13
coils and ductwork, **24.**12–**24.**13
inspection, service, and access, **24.**12
monitoring of operation conditions, **24.**12
Mass-transfer coefficients, convective, **3.**15
Masterformat, **1.**23
Measurements, pressure and airflow, **17.**75–**17.**78
equal-area method, **17.**77–**17.**78
log-linear rule for round duct, **17.**77–**17.**78
log Tchebycheff rule, **17.**77**17.**78
manometer, **17.**75–**17.**77
measurements in air ducts, **17.**76–**17.**77
Pitot tube, **17.**75–**17.**77
Mechanical work, **4.**4
Metabolic rate, **4.**4
Microbial growth, eliminating, **24.**4–**24.**6
basics, **24.**4
eliminate water leaks, **24.**5
microbial growth, **24.**4–**24.**5
pressurization control, **24.**5
prevent damped surface and material, **24.**5
purge, **24.**5
ultraviolet germicidal irradiation, **24.**5–**24.**6
Moist air, **2.**1–**2.**2
calculation of the properties of, **2.**3
density, **2.**10
enthalpy, **2.**8–**2.**9
moist volume, **2.**9–**2.**10
sensible heat, **2.**10–**2.**11
Moisture content, **3.**11
Moisture migration in building materials, **3.**13–**3.**14
Moisture permeability index, **4.**8
Moisture-solid relationship, **3.**12–**3.**13
Moisture transfer, **3.**11–**3.**17
from the surface, **3.**14–**3.**15
in building envelopes, **3.**16–**3.**17
Montreal Protocol and Clean Air Act, **9.**10–**9.**11
Multistage vapor compression systems, **9.**29–**9.**31
compound systems, **9.**29–**9.**30
interstage pressure, **9.**30–**9.**31
flash cooler and intercooler, **9.**31

Network technology, **5.**43–**5.**44
Night shutdown operating mode, **6.**3–**6.**6
conditioning period, **6.**6
cool-down period, **6.**4–**6.**6
influence of stored heat, **6.**6
Night shutdown operating mode *(Cont.)*
night shutdown period, **6.**3–**6.**4
warm-up period, **6.**4–**6.**6
Noise, **4.**32
airflow, **19.**5–**19.**6
from chiller and pumps, **19.**4–**19.**5
diffusers and grilles, **19.**6
maximum duct velocities, **19.**5–**19.**6
poor fan entry and discharge, **19.**6
Noise control, recommended procedure, **19.**3–**19.**4
Noise control for typical air system, **19.**25–**19.**26
combination of supply fan noise and terminal noise, **10.**25
environment adjustment factor, **19.**26
estimated sound pressure level for space served by terminal units, **19.**25–**19.**26
plenum ceiling effect, **19.**26
Nomenclature, A.1–A.6
Greek letter symbols, A.8–A.9
subscripts, A.6–A.8

Open data communication protocol, **5.**41
application layer, **5.**42–43
ARCNET, **5.**44
BACnet, **5.**41–**5.**44
data link/physical layer, **5.**43–**5.**44
Ethernet, **5.**43–**5.**44
local area networks (LANs), **5.**43
LonTalk, **5.**44
LonTalk LAN, **5.**44
master-slave/token passing (MS/TP), **5.**44
network layer, **5.**43
network technology, **5.**43–**5.**44
point-to-point, **5.**44
proprietary network, **5.**44
Outdoor air requirements for occupants, **4.**30–**4.**31
Outdoor design conditions, **4.**38–**4.**42
Outdoor design temperature, **4.**38–**4.**42
1.0% summer wet-bulb, **4.**39
summer dry-bulb, **4.**39
summer mean coincident wet-bulb, **4.**39
winter dry-bulb, **4.**39
Overlooked commissioning, **1.**17

Packaged systems, **29.**2 -**29.**4
applications, **29.**3–**29.**4
comparison between packaged and central systems, **29.**2–**29.**3
types of, **29.**4

Packaged systems, fan-powered VAV, **29.**18–**29.**22
case-study: rooftop packaged unit, **29.**20–**29.**22
controls, **29.**20,
supply volume flow rate and coil load, **29.**19–**29.**20
system characteristics, **29.**21
system description, **29.**18–**29.**19
Packaged systems, perimeter-heating VAV, **29.**18
system characteristics, **29.**6
Packaged systems, single-zone constant-volume, **29.**4 -**29.**6
controls, **29.**5
energy use intensities, **29.**5
supply volume flow rate and coil loads, **29.**4–**29.**5
system characteristics, **29.**5–**29.**6
system description, **29.**4
Packaged systems, single-zone VAV, **29.**7–**29.**8
controls, **29.**7–**29.**8
system calculations, **29.**7
system characteristics, **29.**6
system descriptions, **29.**7
Packaged systems, VAV cooling, **29.**9–**29.**12
duct static pressure control, **29.**10–**29.**12
pressure characteristics, **29.**10
supply volume flow rate and coil load, **29.**10
system characteristics, **29.**6
system description, **29.**9–**29.**10
Packaged systems, VAV reheat, **29.**12–**29.**18
air-cooled, water-cooled, and evaporative-cooled condensers, **29.**17
air-side economizer mode, **29.**15
case-study for precision manufacturing, **29.**17–**29.**18
discharge air temperature control, **29.**15–**29.**16
evenly distributed airflow at DX coils, **29.**14–**29.**15
fan modulation, **29.**16–**29.**17
initiation of cooling stages, **29.**15–**29.**16
night setback and morning warm-up, **29.**14
reset, **29.**16
sound control, **29.**17
supply volume flow rate and coil load, **29.**12–**29.**14
system characteristics, **29.**6
system description, **29.**12–**29.**13
Packaged terminal air conditioner (PTAC), **1.**4
Packaged terminal heat pump (PTHP), **1.**4
Packaged units, **16.**12–**16.**23
controls, **16.**18–**16.**19
Packaged units *(Cont.)*
indoor air quality, **16.**18
indoor environmental control, **16.**17–**16.**18
scroll compressors and evaporative condensers, **16.**18
selection of, **16.**19–**16.**22
Standard 90.1–1999 minimum efficiency requirements, **16.**19
types of, **16.**12
Packaged units, indoor, **16.**15–**16.**16
Packaged units, rooftop, **16.**12–**16.**15
compressors, **16.**14–**16.**15
condensers, **16.**15
curb, **16.**13
DX-coils, **16.**13–**16.**14
electric heating coil, **16.**14
gas-fired furnace,**16.**14
heat pump, **16.**15
humidifiers, **16.**14
supply, return, relief, and exhaust fans, **16.**14
Packaged units, rooftop, sound control, **19.**29–19–32
basics, **19.**29
discharge side duct breakout, **19.**31
sound source on return side, **19.**31–**19.**32
sound sources and paths, **19.**30–**19.**31
structure-borne noise, **19.**32
Packaged units, split, **16.**16–**16.**17
Panel heating and cooling, **28.**33
Personal computer workstation, **5.**39–**5.**40
Plant-building-loop, **7.**43–**7.**51
balance valves, **7.**49–**7.**50
building loop, **7.**43
coil discharge air temperature control, **7.**43
common pipe thermal contamination, **7.**51
low ΔT, **7.**49
plant-loop, **7.**43
pressure differential control, **7.**45
sequence of operations, **7.**46–**7.**49
staging control, **7.**43–**7.**44
system characteristics, **7.**45–**7.**46
variable-speed pumps connected in parallel, **7.**49
water leaving chiller temperature control, **7.**43
Plant-distributed pumping, **7.**52–**7.**53
Plant-through-building loop, **7.**40–**7.**42
bypass throttling flow, **7.**40–**7.**41
distributed pumping, **7.**41
variable flow, **7.**41–**7.**42
Point or object, **5.**25
Poor indoor air quality, **1.**17
Precision, **2.**6

Pressure flow characteristics, **22.**22–**22.**24
fan characteristics, **22.**7
mixing-exhaust section and conditioned space, **22.**8
supply and relief fan combination, field survey system pressure characteristics, **22.**23–**22.**24
supply and return fan combination system, **22.**22–**22.**23
system pressure diagram, **22.**5–**22.**8
VAV systems, fixed part, **22.**5
VAV systems, variable part, **22.**5
variation of pressure in mixing box, **22.**23
Pressure sensors, **5.**19
reference pressure, **5.**19
Primary ambient-air quality standard, **4.**29
Profile angle, **3.**42
Properties of air, physical, A.15
Properties of moist air, thermodynamic, A.13–A.14
Properties of water, physical, A.15
Psychrometric chart, A.12
Pump-piping systems, **7.**34–**7.**38
connected in series, **7.**35–**7.**36
modulation of, **7.**36–**7.**37
operating point, **7.**34–**7.**35
parallel-connected, **7.**35–**7.**36
pump laws, **7.**37
system curve, **7.**34
Psychrometer, **2.**12–**2.**13
aspiration, **2.**14–**2.**15
sling, **2.**14–**2.**15
Psychrometrics, **2.**1

R-value, **3.**7
overall, **3.**7
Radiant heat loss from building, **3.**46–**3.**47
Radiated noise, **19.**18–**19.**19
break-out and break-in, **19.**18–**19.**19
break-out and break-in sound power level, **19.**19–**19.**20
duct rumble, **19.**19
Radiation, atmospheric, **3.**47
Reciprocating compression, performance, **11.**29–**11.**34
condenser, **11.**33–**11.**34
evaporator, **11.**32–**11.**33
power input, **11.**30–**11.**32
refrigeration capacity, **11.**30
Reciprocating refrigeration systems, **11.**2–**11.**42
air-cooled reciprocating chiller, **11.**2–**11.**3
air-cooled reciprocating DX cooler, **11.**2
Reciprocating refrigeration systems *(Cont.)*
balance of capacities of selected components, **11.**35–**11.**36
capacity control, **11.**24–**11.**26
compressor components, **11.**5–**11.**8
crankcase heater, **11.**7–**11.**8
cylinder block and piston, **11.**7
cylinder unloader, **11.**24
filter dryer and strainer, **11.**10–**11.**11
frost control, **11.**27
hot-gas bypass control, **11.**26
liquid overfeed,**11.**3–**11.**4
liquid receiver, **11.**8
liquid-suction heat exchanger, **11.**8–**11.**10
low-pressure and high-pressure controls, **11.**26–**11.**27
low-temperature control, **11.**27
minimum performance, ASHRAE/IESNA Standard 90.1–1999, **11.**41–**11.**42
motor overload control, **11.**29
multistage, **11.**4
oil lubrication, **11.**7
oil-pressure failure control, **11.**27–**11.**29
on/off control, **11.**24
pressure relief valves, **11.**11–**11.**12
real cycle of a single-stage, **11.**4–**11.**5
reciprocating compressors, **11.**5
refrigerant charge valve, **11.**12
safety controls, **11.**26–**11.**29
service valves, **11.**11–**11.**12
solenoid valves, **11.**11
speed modulation control, **11.**24–**11.**26
suction and discharge valves, **11.**7
system balance, **11.**34–**11.**36
Reciprocating refrigeration systems, air-cooled direct-expansion, **11.**36–**11.**42
compressor short cycling, **11.**40
defrosting, **11.**40–**11.**41
liquid slugging, **11.**40
main problems, **11.**40–**11.**42
oil returns, **11.**40
operating balance, **11.** 36–**11.**37
part-load operation using an unloader, **11.**38–**11.**39
pressure characteristics, **11.**37–**11.**38
proper refrigerant charge, **11.**41–**11.**42
pump-down control, **11.**39–**11.**40
Refrigerant flow control devices, **10.**51–**10.**58
advantages of electric expansion valves, **10.**56
analog valves, **10.**55–**10.**56
capacity superheat curve, **10.**52
capillary tubes, **10.**57–**10.**58
cross charge, **10.**53–**10.**54
electric expansion valves, **10.**55–**10.**56

Refrigerant flow control devices *(Cont.)*
external equalizer, **10.**52–**10.**53
float valves, high-side, **10.**56
float valves, low-side, **10.**56–**10.**57
hunting of thermostatic expansion valve, **10.10.**54–**10.**55
limited liquid charge, **10.**53–**10.**54
liquid charge, **10.**53–**10.**54
operating characteristics, **10.**51–**10.**52
pulse-width-modulated valve, **10.**55–**10.**56
step motor valve, **10.**55
straight charge, **10.**53–**10.**54
thermostatic expansion valves, **10.**51–**10.**53
Refrigerant piping for reciprocating refrigeration system, **11.**12–**11.**23
copper tubing, **11.**12–**11.**13
discharge line, **11.**20–**11.**21
discharge line sizing, **11.**20–**11.**21
double riser, **11.**16–**11.**17
liquid line, **11.**21–**11.**23
liquid line sizing, **11.**22–**11.**23
maximum pressure drop, **11.**17
minimum refrigeration load for oil entrainment up hot-gas riser, **11.**20
minimum refrigeration load for oil entrainment up suction riser, **11.**19
oil trap and piping pitch, **11.**15–**11.**16
parallel connections, **11.**23
piping design, **11.**13
pressure drop of valves, and fittings **11.**15–**11.**16
size of copper tubing, **11.**14
sizing procedure, **11.**14–**11.**15
suction line, **11.**15–**11.**20
suction line sizing, **11.**18–**11.**19
suction line sizing chart, **11.**17–**11.**18
Refrigerants, **9.**3
azeotropic, **9.**3
blends, **9.**3
CFCs replacements, **9.**13
classification, **9.**13–**9.**16
concentration shift, **11.**46–**11.**47
conversions and replacements, **9.**11
glide, **9.**3–**9.**4, **11.**46–**11.**47
global warming potentials, **9.**7–**9.**10
chlorofluorocarbons (CFCs) and halons, **9.**16
hydrochlorofluorocarbons (HCFCs), **9.**15–**9.**16
hydrofluorocarbons (HFCs), **9.**13–**9.**14
inorganic compounds, **9.**16
near azeotropic, **9.**3
numbering of, **9.**4
ozone depletion potentials, **9.**7–**9.**10
phase-out of CFC's and halons , **9.**10
Refrigerants *(Cont.)*
recovery, recycle, and reclaiming, **9.**11–**9.**13
reducing leakage and preventing deliberate venting, **9.**11–**9.**13
restrict production of HCFCs, **9.**10–**9.**11
storage of, **9.**59
use of, **9.**7
zeotropic, **9.**3
Refrigerants, properties, **9.5**–**9.**7
effectiveness of refrigeration cycle, **9.**5
evaporating and condensing pressure, **9.**6
inertness, **9.**6
leakage detection, **9.**6–**9.**7
oil miscibility, **9.**6
physical properties, **9.**6
refrigeration capacity, **9.**6
safety requirements, **9.**5
thermal conductivity, **9.**6
Refrigerants safety, **9.**56
Refrigerating machinery room, **9.**58–59
storage of refrigerants, **9.**59
Refrigeration, **9.**2
unit of, **9.**17
Refrigeration compressors, **9.**51–**9.**56
direct-drive, belt drive, and gear drive, **9.**53
energy use index, **9.**55–**9.**56
hermetic, semihermetic, and open, **9.**53
isentropic, and polytropic analysis, **9.**54–**9.**55
motor, mechanical, and compression efficiency, **9.**54
performance, **9.**53–**9.**56
positive displacement and nonpositive displacement, **9.**51–**9.**53
volumetric efficiency, **9.**53–**9.**54
Refrigeration cycles, **9.**17
air expansion , **9.**45–**9.**49
Carnot, **9.**19–**9.**21
coefficient of performance, **9.**21–**9.**22
cycle performance, **9.**22–**9.**24
determination of enthalpy by polynomials, **9.**24–**9.**25
ideal vapor compression, single stage, **9.**22–**9.**26
performance, **9.**19–**9.**21
Refrigeration effect, refrigerating load, refrigerating capacity, **9.**25–**9.**26
Refrigeration processes, **9.**16–**9.**17
Refrigeration systems, **9.**2
absorption, **9.**2, **14.**1–**14.**3
air or gas expansion, **9.**2
cascade, **9.**40–**9.**43
centrifugal, **13.**1–**13.**7
classifications, **9.**49–**9.**51
compound, **9.**31–**9.**40

Refrigeration systems *(Cont.)*
developments, recent ,9.51
high-probability systems, application rules, **9.**56–**9.**57
low-probability systems, application rules, **9.**57–**9.**58
multistage vapor compression, **9.**29–**9.**31
reciprocating , **11.**2–**11.**42
vapor compression, **9.**2
Refrigeration systems, absorption, **14.**1–**14.**3
applications, **14.**3
cost analysis, **14.**2–**14.**3
historical development, **14.**2
types of, **14.**1–**14.**2
Refrigeration systems, rotary, **11.**42–**11.**43
main components, **11.**43
rotary compressor, **11.**42–**11.**43
system performance, **11.**43
Refrigeration systems, screw, **11.**55
air-cooled screw chillers, **11.**55
ASHRAE/IESNA Standard 90.1–1999 minimum performance, **11.**54–**11.**55
capacity control, **11.**53–**11.**52
controls, **11.**53
economizer, **11.**54
electric expansion valves, **11.**55
location of installation, **11.**55
oil cooling, **11.**51, **11.**53
performance of twin-screw compressor, **11.**52–**11.**53
screw compressors, **11.**50–**11.**52
system performance, **11.**55
types of, **11.**50
variable volume ratio, **11.**54
Refrigeration systems, scroll, **11.**43–1150
capacity control and part-load performance, **11.**47–**11.**48
chillers, **11.**48–**11.**49
circulating concentration shift, **11.**46–**11.**47
compressor performance, **11.**46
concentration shift, **11.**46–**11.**47
heat exchanger flow configuration, **11.**47
radial and axial compliance, **11.**44–**11.**45
scroll compressors, **11.**44–**11.**45
system characteristics, **11.**48
temperature glide, **11.**46–**11.**47
types of, **11.**43–**11.**44
Relative humidity, **2.**7–**2.**8
Residuals, **5.**61
normalized, **5.**62
Resistance temperature detectors (RTDs), **2.**6, **5.**18
Retrofit, remodeling, and replacement, **1.**19
Return and exhaust inlets, **18.**17–**18.**20
exhaust inlets, **18.**19
light troffer diffuser, **18.**19–**18.**20
return grilles, **18.**18–**18.**19
return slots, **18.**18–**18.**19
troffer diffuser slot, **18.**18–**18.**19
Return and exhaust systems, **22.**2–**22.**3
ANSI/ASHRAE Standard 90.1–1999 dampers specifications, **22.**3
enclosed parking garage ventilation, **22.**3
exhaust hoods, **22.**3
low-level return systems, **22.**2–**22.**3
return ceiling plenum, **22.**2
types of, **22.**2
Room, **6.**2
Room air conditioner, **1.**4
Room heat pump, **1.**4
Room sound power level and room sound pressure level, relationship, **19.**23–**19.**24
array of ceiling diffusers, **19.**24
single or multiple sound sources, **19.**23–**19.**24

Safety factor, **1.**20
Semiheated space, **3.**49
Sensible heat exchange, **4.**5
Sensing element, **5.**16
Sensitivity, **2.**6
Sensors, **2.**6, **5.**16–**5.**17
air, **5.**16–**5.**18
air quality (VOC), **5.**20
CO_2, **5.**20
drift, **5.**16
intelligent network, **5.**21
occupancy, **5.**20–**5.**21
resistance temperature detectors (RTD), **5.**18
temperature sensors, **5.**18
wireless zone, **5.**21
Sequence of operations, **5.**5–**5.**6
Set point, **5.**5
Shading coefficients, **3.**36
Shading devices, **3.**40–**3.**43
draperies, **3.**41
external, **3.**42–**3.**43
indoor, **3.**40–**3.**42
overhang, **3.**42
roller shades, **3.**41–**3.**42
side fin, **3.**42
venetian blinds, **3.**40–**3.**41
Shading from adjacent buildings, **3.**43–**3.**44
Sick building, **4.**27
Sick building syndrome, **1.**17, **4.**27
Silencers, **19.**12–**19.**17
characteristics, **19.**14–**19.**15

Silencers *(Cont.)*
dissipative, **19.**14
free area ratio, **19.**15
insertion loss, **19.**15
locations of, **19.**15–**19.**16
packless, **19.**14
pressure drop of, **19.**15
reflection-dissipative, **19.**14
selection of, **19.**17
self noise of, **19.**15
sound-attenuating plenum, **19.**13–**19.**14
types of, **19.**13–**19.**14
Silencers, active, **19.**14
frequency limits, **19.**16
operating characteristics, **19.**16
performance, **19.**17
system characteristics, **19.**16–**19.**17
Simulation, energy software DOE-2.1E, **25.**25–**25.**28
energy efficiency measures, **25.**27
energy simulation software, **25.**25–**25.**26
loads, **25.**25
plant, **25.**27–**25.**28
systems, **25.**26–**25.**27
Simulation, system, **25.**17–**25.**19
dynamic simulation, **25.**18
energy simulation, **25.**17
performance equations, **25.**17–**25.**18
physical modeling, **25.**18
sequential, **25.**19
simultaneous, **25.**19
steady-state, **25.**18–**25.**19
Simulation of a centrifugal chiller, **25.**19–**25.**25
centrifugal compressor model, **25.**23–**25.**25
condenser model, **25.**22
cooling tower model, **25.**23
evaporator model, **25.**20–**25.**21
operating parameter, **25.**20
simulation methodology, **25.**20
system model, **25.**19–**25.**20
Skin wetness, **4.**9
Smoke control and fire safety, **22.**24–**22.**38
ANSI/NFPA 92A and 92B, **22.**28
automatic sprinkler on fire protection, **22.**27–**22.**28
effective area and flow rates, **22.**27
fire safety in buildings **22.**24–**22.**25
smoke control in atria, **22.**28
smoke management in atria, malls, and large areas, **22.**28
smoke movement in buildings, **22.**25–**22.**27
zone smoke control, **22.**31–**22.**32
zone smoke control, design considerations, **22.**32
Software, load calculations and energy analysis, **6.**42–**6.**49
building load analysis and system thermodynamics (BLAST), **6.**42
TRACE-600, **6.**42–**6.**49
Sol-air temperature, **3.**47
Solar angles, **3.**22–**3.**25
altitude angle, **3.**23–**3.**24
angle of incidence, **3.**23–**3.**25
hour angle, **3.**22–**3.**24
latitude angle, **3.**22–**3.**24
relationships, **3.**23–**3.**24
solar azimuth angle, **3.**23–**3.**24
solar declination angle, **3.**22–**3.**24
surface-solar azimuth angle, **3.**23–**3.**24
Solar constant, **3.**25
Solar heat gain coefficient (SHGC), **3.**33
Solar heat gain factors, **3.**37
Solar intensity, **3.**24–**3.**25
direct normal radiation, **3.**26
Solar radiation, **3.**25–**3.**29
apparent, **3.**26
diffuse radiation, **3.**26
direct radiation, **3.**26
extraterrestrial intensity of, **3.**25
for a clear sky, **3.**26–**3.**283.28–**3.**29
reflection of, **3.**28
Sorption isotherm, **3.**11–**3.**12
Sound, **4.**32
airborne, **4.**32
octave bands, **4.**33
power, **4.**32
power level, **4.**32–**4.**33
pressure level, **4.**32–**4.**33
Sound attenuation, along duct-borne path, **19.**6–**19.**12
duct-borne crosstalk,**19.**11
in ducts, **19.**6–**19.**9
at elbows and branch takeoffs, **19.**9–**19.**10
end reflection loss, **19.**10–**19.**11
inner-lined round ducts, **19.**7
lined flexible ducts, **19.**8–**19.**9
lined rectangular ducts, **19.**8
unlined rectangular sheet-metal ducts, **19.**7
unlined round ducts, **19.**7
Sound control, **19.**1–**19.**2
control at design stage, **19.**3
Sound control criteria, **4.**34
A-weighted sound level, **4.**34
noise criteria (NC), **4.**34
room criteria (RC), **4.**34
Sound paths, **19.**2–**19.**3
airborne, **19.**2
duct-borne, **19.**2

Smoke paths *(Cont.)*
radiated sound, **19.**2
structure-borne, **19.**2–**19.**3
Space, **6.**2
Space air diffusion, mixing flow, design procedure, **18.**31–**18.**34
choose an optimum throw/characteristic length ratio, **18.**33
design characteristics of slot diffusers in perimeter zone, **18.**33–**18.**34
drop of cold air jet, **18.**34
final layout, **18.**34
select the type of supply outlet, **18.**31–**18.**32
sound level, **18.**34
total pressure loss of supply outlet, **18.**34
volume flow rate per outlet or unit length, **18.**32–**18.**33
Space air diffusion, principles, **18.**2–**18.**5
age of air, **18.**4–**18.**5
air change effectiveness, **18.**4
air diffusion performance index (ADPI), **18.**3–**18.**4
draft, **18.**2
draft temperature, effective, **18.**2–**18.**3
nominal air change effectiveness, **18.**5
nominal time constant, **18.**5
space air velocity vs. space air temperature, **18.**3
space diffusion effectiveness factor, **18.**4
turbulence intensity, **18.**2–**18.**3
ventilation effectiveness, **18.**4
Space airflow pattern, displacement flow, **18.**38–**18.**43
ceiling plenum, **18.**41
supply air velocity, **18.**41
unidirectional flow, **18.**38–**19.**39
unidirectional flow for clean rooms, **18.**39–**18.**40
ventilating ceiling, **18.**40–**18.**41
Space airflow pattern, mixing flow, **18.**20–**18.**28
airflow pattern, **18.**20
principles and characteristics, **18.**21
reverse air streams in the occupied zone, **18.**21
stratified mixing flow, **18.**25–**18.**28
stratified mixing flow using nozzles, **18.**27–**18.**28
types and locations of return and exhaust inlets, **18.**21
types and locations of supply outlets, **18.**21
using ceiling diffusers, **18.**23–**18.**24
using high-side outlets, **18.**21–**18.**23
using sill or floor outlets, **18.**24–**18.**25
using slot diffusers, **18.**24
Space airflow pattern, projecting flow, **18.**44–**18.**48
applications of desktop task conditioning systems, **18.**48
benefits of, **18.**44
desktop task conditioning systems, **18.**46–**18.**48
distance between target zone and supply outlet, **18.**44
horizontal vs. vertical jet, **18.**44–**18.**46
industrial spot cooling systems, **18.**44–**18.**46
performance of desktop task conditioning systems, **18.**47–**18.**48
recommendations in spot cooling design, **18.**46
target velocities, **18.**46
thermal sensation, **18.**46
Space airflow pattern, stratified displacement flow, **18.**42–**18.**43
comparison of stratified displacement flow and mixing flow, **18.**43
operating characteristics, **18.**42–**18.**43
two-zone stratified model,**18.**42
Space airflow pattern, upward flow underfloor air distribution, **18.**48–51
applications, **18.**51
consistent access plenum temperature, **18.**50
design considerations, **18.**50–**18.**51
floor plenum master zone air temperature control, **18.**50
heat unneutralized, **18.**50
thermal storage of floor plenum, **18.**49
upward flow from floor plenum, **18.**48–**18.**49
Space heat extraction rate, **6.**3
Space heat gain, **6.**3
Space pressurization and return/relief volume flow controls, **23.**16–**23.**18
characteristics of space pressure control, **23.**16–**23.**17
VAV systems return/relief fan volume flow control, **23.**17–**23.**18
volume flow of air leakage and effective leakage area, **23.**17
Space pressurization control, **24.**13
Space pressurization or building pressurization, **4.**37–**4.**38, **20.**7–**20.**14
airflow balance, **20.**11–**20.**13
air systems and mechanical ventilation systems, **20.**11
characteristics, **20.**7
by differential flow, **20.**11–**20.**13
differentials, **4.**37–**4.**38
neutral pressure level, **20.**7–**20.**9

Space pressurization or building pressurization *(Cont.)*
stack effect, **20.**7–**20.**8
stack effect for high-rise buildings, **20.**9–**20.**10
wind effect, **20.**10–**20.**11
Specifications, **1.**22–**1.**23
Stairwell pressurization, **22.**29–**22.**34
bottom single injection or bottom and top injection, **22.**34–**22.**35
characteristics, **22.**29–**22.**30
overpressure relief and feedback control, **22.**30–**22.**31
pressure drop coefficient, **22.**34
stair and shaft vents, **22.**31
system pressure loss, **22.**33–**22.**35
volume flow rate, **22.**32–**22.**33
Standard 90.1–1999 for building envelope, **3.**48–50
Standard 90.1–1999, simplified approach option for small and medium HVAC&R systems, **29.**8–**29.**9
Steam point, **2.**4–**2.**5
Subcooling, **9.**26
Superheating, **9.**26–**9.**27
Supply air condition, determination, **20.**62–**20.**66
air conditioning rules, **20.**63
graphical method, **20.**63–**20.**64
influence of sensible heat ratio, **20.**64–**20.**66
Supply outlets, **18.**11–**18.**17
ceiling diffusers, **18.**12–**18.**14
gang-operated turning vanes, **18.**17
grilles, **18.**11–**18.**12
induction, **18.**14
nozzle diffusers, **18.**16–**18.**17
nozzles, **18.**16–**18.**17
plenum box, **18.**14–**18.**15
registers, **18.**11–**18.**12
slot diffusers, **18.**14–**18.**16
split dampers, **18.**17
Supply volume flow rate, **20.**59–**20.**62
based on space cooling vs. heating load, **20.**59–60
rated volume flow of supply and return fans, **20.**61–**20.**62
requirements other than cooling load, **20.**60–**20.**62
temperature difference vs. enthalpy difference, **20.**60
volumetric vs. mass flow rate, **20.**60
System pressure diagram, **22.**5–**22.**8
duct static pressure control, **22.**5–**22.**7
Supply volume flow rate *(Cont.)*
fan characteristics, **22.**7–**22.**8

Temperature, **2.**4
dew point, **2.**11
globe, **4.**9
mean radiant, **4.**9–**4.**12
mean surface temperature of clothing, **4.**5
measurements, **2.**6
operative, **4.**5
Temperature scales, **2.**4–**2.**5
absolute scale, **2.**5
Celsius, **2.**4–**2.**5
Fahrenheit, **2.**4–**2.**5
Kelvin, **2.**4–**2.**5
Rankine, **2.**4–**2.**5
thermodynamic, **2.**5
Testing, adjusting, and balancing (TAB), **32.**2–**32.**4
Thermal comfort, **4.**15–**4.**20
ASHRAE comfort zones, **4.**.17–**4.**18
comfort-discomfort diagrams, **4.**17–**4.**20
factors affecting, **4.**14–**4.**15
Fanger's comfort chart, **4.**15–**4.**17
Fanger's comfort equation, **4.**15–**4.**17
heart rate (HR), **4.**19–**4.**20
predicted mean vote (PMV), **4.**15–**4.**17
thermal sensational scale, **4.**16
Thermal insulation, **3.**18–**3.**22
economic thickness, **3.**21
Thermal interaction:
between human body and indoor environment, **4.**2
steady-state thermal equilibrium, **4.**3
transient energy balance, **4.**3
two-node model, **4.**2
Thermal resistance, **3.**4
of airspaces, **3.**21–**3.**22
convective, **3.**5
Thermal resistance ratio, **3.**19–**3.**21
Thermal storage systems, **31.**1–**31.**5
benefits and drawbacks, **31.**2–**31.**3
full storage or load shift, **31.**3–**31.**5
ice-storage and chilled water storage, **31.**5
impact of electric deregulation, **31.**2
partial storage or load leveling, **31.**3–**31.**5
system description, **31.**1–**31.**2
Thermistors, **2.**6
Thermodynamic wet bulb temperature, **2.**12
Thermometer, globe, **4.**9
Total shortwave irradiance, **3.**34, **3.**37
TRACE 600 input, **6.**42–**6.**49
external loads, **6.**45

TRACE 600 input *(Cont.)*
internal loads, **6.**46–**6.**47
job, **6.**44–**6.**45
load methodology, **6.**43–**6.**44
schedules, **6.**45–**6.**46
structure and basics, **6.**42–**6.**43
TRACE 600, minimum input, run, and outputs, **6.**47
Transducers, **5.**21
Transfer function, method, **6.**14–**6.**26
ceiling, floors, and interior partition walls, **6.**16–**6.**17
conversion of heat gain to cooling load, **6.**24–**6.**25
electric motors, **6.**21–**6.**23
equipment and appliances, **6.**21–**6.**23
exterior wall and roofs, **6.**16
heat extraction rate, **6.**25
heat loss to surroundings, **6.**25–**6.**26
heat to space, **6.**20
infiltration, **6.**24
lighting, **6.**18–**6.**21
space air temperature, **6.**25
window glass, **6.**17
Transmission losses, **19.**19–**19.**23
for selecting building structures, **19.**23
TL_{in} for flat oval ducts, **19.**22
TL_{in} for rectangular ducts, **19.**22
TL_{in} for round ducts, **19.**22
TL_{out} for flat oval ducts, **19.**21
TL_{out} for rectangular ducts, **19.**21
TL_{out} for round ducts, **19.**20–**19.**21
Transmitters, **5.**21
Triple point, **2.**4–**2.**5
T-w chart, **2.**19

Unit conversion, Inch-Pound (I-P) units to SI units, A.15–A.17
Updated technology, **1.**17

Valves, **7.**16–**7.**17
balancing, **7.**17
check, **7.**16–**7.**17
connections and ratings, **7.**17–**7.**18
gate, **7.**16
globe, **7.**16–**7.**17
materials, **7.**18
pressure relief, **7.**17
Vapor retarders, **3.**17, **3.**18
Variable-air-volume (VAV) systems, **1.**11, **21.**2–**21.**56
comparison between various VAV systems, **21.**56
Variable-air-volume (VAV) systems *(Cont.)*
dew point control, **23.**27–**23.**28
diagnostics, **23.**28
functional controls, **23.**26–**23.**28
interaction between controls, **23.**29–**23.**30
nighttime setback and warmup or cooldown control, **23.**26–**23.**27
override, **23.**29–**23.**30
recommendations for VAV controls, **23.**28–**23.**29
sequence control, **23.**29
specific controls, **23.**2
steam humidifier control, **23.**27
types of, **21.**2–**21.**3
VAV systems, dual duct, **21.**33–**21.**44
case-study, **21.**42–**21.**44
discharge air temperature control, **21.**40–**21.**41
mixing mode operation, **21.**38
mixing VAV box, **21.**36–**21.**38
number of supply fans, **21.**36
part-load operation, **21.**43–**21.**44
system description, **21.**33–36
winter heating and winter cooling mode operation, **21.**43
zone control and sequence of operations, **21.**38–**21.**40
zone supply flow rate, **21.**41–**21.**42
VAV systems, fan-powered, **21.**44–**21.**56
design considerations, **21.**55–**21.**56
fan energy use, **21.**54–**21.**55
fan-powered VAV box, **21.**48–**21.**50
parallel fan-powered VAV box, **21.**48–**21.**50
parallel fan-powered VAV box, fan characteristics, **21.**50–**21.**51
series fan-powered VAV box, **21.**48–**21.**49
supply volume flow rate,**21.**53–**21.**54
system description, **21.**44–**21.**47
zone control and sequence of operations, **21.**52–**21.**53
VAV systems, single-zone, **21.**2–**21.**18
air conditioning cycle and system calculations, **21.**4, **21.**16–**21.**17
system description, **21.**3–**21.**5
year-round operation of, **21.**5–**21.**8
zone temperature control - sequence of operations, **21.**17–**21.**18
VAV systems, VAV cooling, VAV reheat, and perimeter heating VAV systems, **21.**18–**21.**33
air skin VAV system, **21.**21
ANSI/ASHRAE Standard 90.1–1999 specifications, **21.**20

VAV systems, VAV cooling, VAV reheat, and perimeter heating VAV systems *(Cont.)*
minimum ventilation, discharge air temperature, and duct static pressure controls , **23.**30–**23.**34
perimeter heating VAV systems, **21.**20–**21.**21
reheating VAV box, **21.**23
sequence of operations, primary considerations, **23.**30–**23.**35
stability of zone control, **21.**26–**21.**27
VAV box, **21.**21–**21.**23
VAV box, pressure dependent and pressure independent, **21.**23
VAV box, sound level, **21.**23–**21.**25
VAV cooling systems, **21.**18–**21.**19
VAV reheat system, case-study, **21.**27–**21.**33
VAV reheat system, cooling mode part-load operation, **21.**29–**21.**30
VAV reheat system, dead-band mode, **21.**25–**21.**26
VAV reheat system, winter cooling mode in interior zone, **21.**32–**21.**33
VAV reheat system, winter reheating in perimeter zone, **21.**30–**21.**32
VAV reheat system, volume flow rate and coil load, **21.**28–**21.**29
VAV reheat systems, **21.**19
VAV reheat zone temperature control sequence of operations, **21.**25–**21.**26
Ventilation, **24.**2
air economizer, **24.**2–**24.**3
minimum outdoor air damper and economizer damper, **24.**3
minimum ventilation control, **24.**3
outdoor air requirement, **24.**2
purge operation, **24.**2–**24.**3
time of operation, **24.**2
Ventilation control, minimum, **23.**2–**23.**5
ASHRAE Standard 62–1999, **23.**3 -**23.**4
basic approach, **23.**2
conference rooms, **23.**16
direct measurement of minimum outdoor air intake, **23.**15
fan tracking systems, **23.**15–**23.**16
high-occupancy areas, **23.**5
indoor air quality procedure, **23.**3–**23.**4
outdoor air injection fan, **23.**14–**23.**15
recirculation of unused outdoor air, **23.**4–**23.**5
types of, **23.**2–**23.**3
ventilation rate procedure, **23.**3
Ventilation control, minimum, mixed-plenum pressure, **23.**12–**23.**14
applications, **23.**14
Ventilation control, minimum, mixed-plenum pressure *(Cont.)*
monitoring plenum pressure, **23.**12–**23.**13
monitoring pressure drop of louver and damper, **23.**13–**23.**14
supply and return fans, **23.**13
Volume flow control, **5.**33–**5.**35
branch flow control, **5.**33–**5.**34
bypass control, **5.**35–**5.**34
mixed-air control, **5.**33–**5.**34

Warm air furnace, **8.**3–**8.**9
annual fuel utilization efficiency (AFUE), **8.**7
circulating fan, **8.**3–**8.**4
condensing or noncondensing, **8.**7
control and operation, **8.**8–**8.**9
gas burners, **8.**3
gas-fired, **8.**3
heat exchangers, **8.**3
ignition, **8.**3
minimum efficiency, **8.**8
power vent or natural vent, **8.**7
steady state efficiency (SSE), **8.**7
thermal efficiency, **8.**6–**8.**7
types of, **8.**3
venting arrangements, **8.**4
Warm air heating system, low-pressure ducted, **8.**17–**8.**23
duct efficiency, **8.**20
duct leakage, **8.**20–**8.**21
location of furnace, **8.**20
part-load operation and control, **8.**21–**8.**22
supply and return duct, **8.**18
supply duct and return plenum, **8.**18
system efficiency, **8.**20
thermal stratification, **8.**21
Water:
chilled, **1.**8
column (WC), **4.**38
condenser, **1.**8
valves, **5.**26
vapor, **2.**1–**2.**2
Water heat gain factor, **10.**21
Water impurities, **7.**25–**7.**26
Water piping, **7.**7–**7.**16
dimensions, copper, **7.**10–**7.**11
dimensions, steel, **7.**8–**7.**9
expansion and contraction, **7.**14–**7.**15
fittings, **7.**18–**7.**19
insulation, **7.**7.15–**7.**16
material, **7.**7
supports, **7.**14–**7.**15

Water piping *(Cont.)*
system accessories, **7.**19
Water-source heat pumps, **1.**5, **28.**26–**28.**27
control, **28.**30–**28.**3
energy performance by ASHRAE Standard 90.1–1999, **28.**27
Water-source heat pump systems, **28.**24–**28.**33
air system and maintenance, **28.**29
case-study, **28.**31–**28.**32
close-circuit evaporative water cooler, **28.**27–**28.**29
controls, **28.**30–**28.**31
design considerations, **28.**32–**28.**33
loop temperatures, **28.**25–**28.**26
operating characteristics, **28.**24–**28.**25
safety controls, **28.**30
storage tank, **28.**29
system characteristics, **28.**15
system description, **28.**24–**28.**35
water heater, **28.**29
water-loop temperature control, **28.**30
Water systems, **1.**8, **7.**2
accessories, **7.**18–**7.**19
air in, **7.**23–**7.**24
campus type, **7.**53–**7.**58
chilled, **7.**2
chiller plant, **7.**39
closed, **7.**2
condenser or cooling, **7.**2
dual-temperature, **7.**2
evaporative-cooled, **7.**2
friction chart, copper pipes, **7.**6
friction chart, plastic pipes, **7.**7
friction chart, steel pipes **7.**6
hot, **7.**2
maximum allowable pressures, **7.**12–**7.**13
once through, **7.**4
open, **7.**2
Water systems *(Cont.)*
oxidation, **7.**24–25
pressure drop, **7.5**–**7.**7
pressurization control, **7.**19–**7.**20
pump location, **7.**23
temperature difference, **7.**4–**7.**5
types of, **7.**40
variable flow, **7.**40
volume flow, **7.**4–**7.**5
volume flow, chilled water, **7.**38–**7.**39
water velocity, **7.**5
waterlogging, **7.**24–25
wire-to-water efficiency, **7.**37–**7.**38
Water treatments, **7.**27–**7.**28
chemical feeding, **7.**27
microbiological control, **7.**26
scale and corrosion control, **7.**26
Wet bulb:
constant, **2.**13
depression, **2.**13
temperature, **2.**12–**2.**14
Window glass:
clear plate, **3.**29
double-strength sheet glass, **3.**36
glass temperature, **3.**35
heat gain for double-glazing, **3.**34–**3.**36
heat gain for single-glazing, **3.**32–**3.**34
insulating, **3.**29
low-emissivity (low-*E*), **3.**29–**3.**30
optical properties, **3.**30–**3.**31
reflective coated, **3.**29
spectral transmittance, **3.**31
tinted heat-absorbing, **3.**29
type of, **3.**29–**3.**30
U-values, **3.**33

Zone, **6.**2

CONTENTS

Preface to Second Edition xi
Preface to First Edition xiii

Chapter 1. Introduction 1.1

Chapter 2. Psychrometrics 2.1

Chapter 3. Heat and Moisture Transfer through Building Envelope 3.1

Chapter 4. Indoor and Outdoor Design Conditions 4.1

Chapter 5. Energy Management and Control Systems 5.1

Chapter 6. Load Calculations 6.1

Chapter 7. Water Systems 7.1

Chapter 8. Heating Systems, Furnaces, and Boilers 8.1

Chapter 9. Refrigerants, Refrigeration Cycles, and Refrigeration Systems 9.1

Chapter 10. Refrigeration Systems: Components 10.1

Chapter 11. Refrigeration Systems: Reciprocating, Rotary, Scroll, and Screw 11.1

Chapter 12. Heat Pumps, Heat Recovery, Gas Cooling, and Cogeneration Systems 12.1

Chapter 13. Refrigeration Systems: Centrifugal 13.1

Chapter 14. Refrigeration Systems: Absorption 14.1

Chapter 15. Air Systems: Components—Fans, Coils, Filters, and Humidifiers 15.1

Chapter 16. Air Systems: Equipment—Air-Handling Units and Packaged Units 16.1

Chapter 17. Air Systems: Air Duct Design 17.1

Chapter 18. Air Systems: Space Air Diffusion 18.1

Chapter 19. Sound Control 19.1

Chapter 20. Air Systems: Basics and Constant-Volume Systems 20.1

Chapter 21. Air Systems: Variable-Air-Volume Systems 21.1

Chapter 22. Air Systems: VAV Systems—Fan Combination, System Pressure, and Smoke Control 22.1

Chapter 23. Air Systems: Minimum Ventilation and VAV System Controls 23.1

Chapter 24. Improving Indoor Air Quality 24.1

Chapter 25. Energy Management and Global Warming 25.1

Chapter 26. Air Conditioning Systems: System Classification, Selection, and Individual Systems 26.1

Chapter 27. Air Conditioning Systems: Evaporative Cooling Systems and Evaporative Coolers 27.1

Chapter 28. Air Conditioning Systems: Space Conditioning Systems 28.1

Chapter 29. Air Conditioning Systems: Packaged Systems and Desiccant-Based Systems 29.1

Chapter 30. Air Conditioning Systems: Central Systems and Clean-Room Systems 30.1

Chapter 31. Air Conditioning Systems: Thermal Storage Systems 31.1

Chapter 32. Commissioning and Maintenance 32.1

Appendix A. Nomenclature and Abbreviations A.1

Appendix B. Psychrometric Chart, Tables of Properties, and I-P Units to SI Units Conversion B.1

Index follows Appendix B

CHAPTER 1
INTRODUCTION

1.1 AIR CONDITIONING 1.1
1.2 COMFORT AND PROCESSING AIR CONDITIONING SYSTEMS 1.2
Air Conditioning Systems 1.2
Comfort Air Conditioning Systems 1.2
Process Air Conditioning Systems 1.3
1.3 CLASSIFICATION OF AIR CONDITIONING SYSTEMS ACCORDING TO CONSTRUCTION AND OPERATING CHARACTERISTICS 1.3
Individual Room Air Conditioning Systems 1.4
Evaporative-Cooling Air Conditioning Systems 1.4
Desiccant-Based Air Conditioning Systems 1.4
Thermal Storage Air Conditioning Systems 1.5
Clean-Room Air Conditioning Systems 1.5
Space Conditioning Air Conditioning Systems 1.5
Unitary Packaged Air Conditioning Systems 1.6
1.4 CENTRAL HYDRONIC AIR CONDITIONING SYSTEMS 1.6
Air System 1.6
Water System 1.8
Central Plant 1.8
Control System 1.9
Air, Water, Refrigeration, and Heating Systems 1.10
1.5 DISTRIBUTION OF SYSTEMS USAGE 1.10
1.6 HISTORICAL DEVELOPMENT 1.11
Central Air Conditioning Systems 1.11
Unitary Packaged Systems 1.12
Refrigeration Systems 1.12
1.7 POTENTIALS AND CHALLENGES 1.13
Providing a Healthy and Comfortable Indoor Environment 1.13
The Cleanest, Quietest, and Most Precise and Humid Processing Environment 1.13
Energy Use and Energy Efficiency 1.13
Environmental Problems—CFCs and Global Warming 1.15
Air Conditioning or HVAC&R Industry 1.15
1.8 AIR CONDITIONING PROJECT DEVELOPMENT 1.16
Basic Steps in Development 1.16
Design-Bid and Design-Build 1.17
The Goal—An Environmentally Friendlier, Energy-Efficient, and Cost-Effective HVAC&R System 1.17
Major HVAC&R Problems 1.17
1.9 DESIGN FOR AIR CONDITIONING SYSTEM 1.18
Engineering Responsibilities 1.18
Coordination between Air Conditioning and Other Trades, Teamwork 1.19
Retrofit, Remodeling, and Replacement 1.19
Engineer's Quality Control 1.20
Design of the Control System 1.20
Field Experience 1.21
New Design Technologies 1.21
1.10 DESIGN DOCUMENTS 1.21
Drawings 1.22
Specifications 1.22
1.11 CODES AND STANDARDS 1.23
1.12 COMPUTER-AIDED DESIGN AND DRAFTING (CADD) 1.25
Features of CADD 1.25
Computer-Aided Design 1.25
Computer-Aided Drafting (CAD) 1.26
Software Requirements 1.26
REFERENCES 1.26

1.1 AIR CONDITIONING

Air conditioning is a combined process that performs many functions simultaneously. It conditions the air, transports it, and introduces it to the conditioned space. It provides heating and cooling from its central plant or rooftop units. It also controls and maintains the temperature, humidity, air movement, air cleanliness, sound level, and pressure differential in a space within predetermined

limits for the comfort and health of the occupants of the conditioned space or for the purpose of product processing.

The term *HVAC&R* is an abbreviation of *heating, ventilating, air conditioning, and refrigerating*. The combination of processes in this commonly adopted term is equivalent to the current definition of air conditioning. Because all these individual component processes were developed prior to the more complete concept of air conditioning, the term *HVAC&R* is often used by the industry.

1.2 COMFORT AND PROCESSING AIR CONDITIONING SYSTEMS

Air Conditioning Systems

An air conditioning, or HVAC&R, system is composed of components and equipment arranged in sequence to condition the air, to transport it to the conditioned space, and to control the indoor environmental parameters of a specific space within required limits.

Most air conditioning systems perform the following functions:

1. Provide the cooling and heating energy required
2. Condition the supply air, that is, heat or cool, humidify or dehumidify, clean and purify, and attenuate any objectionable noise produced by the HVAC&R equipment
3. Distribute the conditioned air, containing sufficient outdoor air, to the conditioned space
4. Control and maintain the indoor environmental parameters – such as temperature, humidity, cleanliness, air movement, sound level, and pressure differential between the conditioned space and surroundings — within predetermined limits

Parameters such as the size and the occupancy of the conditioned space, the indoor environmental parameters to be controlled, the quality and the effectiveness of control, and the cost involved determine the various types and arrangements of components used to provide appropriate characteristics.

Air conditioning systems can be classified according to their applications as (1) comfort air conditioning systems and (2) process air conditioning systems.

Comfort Air Conditioning Systems

Comfort air conditioning systems provide occupants with a comfortable and healthy indoor environment in which to carry out their activities. The various sectors of the economy using comfort air conditioning systems are as follows:

1. The commercial sector includes office buildings, supermarkets, department stores, shopping centers, restaurants, and others. Many high-rise office buildings, including such structures as the World Trade Center in New York City and the Sears Tower in Chicago, use complicated air conditioning systems to satisfy multiple-tenant requirements. In light commercial buildings, the air conditioning system serves the conditioned space of only a single-zone or comparatively smaller area. For shopping malls and restaurants, air conditioning is necessary to attract customers.

2. The institutional sector includes such applications as schools, colleges, universities, libraries, museums, indoor stadiums, cinemas, theaters, concert halls, and recreation centers. For example, one of the large indoor stadiums, the Superdome in New Orleans, Louisiana, can seat 78,000 people.

3. The residential and lodging sector consists of hotels, motels, apartment houses, and private homes. Many systems serving the lodging industry and apartment houses are operated continuously, on a 24-hour, 7-day-a-week schedule, since they can be occupied at any time.

4. The health care sector encompasses hospitals, nursing homes, and convalescent care facilities. Special air filters are generally used in hospitals to remove bacteria and particulates of submicrometer

size from areas such as operating rooms, nurseries, and intensive care units. The relative humidity in a general clinical area is often maintained at a minimum of 30 percent in winter.

5. The transportation sector includes aircraft, automobiles, railroad cars, buses, and cruising ships. Passengers increasingly demand ease and environmental comfort, especially for long-distance travel. Modern airplanes flying at high altitudes may require a pressure differential of about 5 psi between the cabin and the outside atmosphere. According to the *Commercial Buildings Characteristics* (1994), in 1992 in the United States, among 4,806,000 commercial buildings having 67.876 billion ft^2 (6.31 billion m^2) of floor area, 84.0 percent were cooled, and 91.3 percent were heated.

Process Air Conditioning Systems

Process air conditioning systems provide needed indoor environmental control for manufacturing, product storage, or other research and development processes. The following areas are examples of process air conditioning systems:

1. In textile mills, natural fibers and manufactured fibers are hygroscopic. Proper control of humidity increases the strength of the yarn and fabric during processing. For many textile manufacturing processes, too high a value for the space relative humidity can cause problems in the spinning process. On the other hand, a lower relative humidity may induce static electricity that is harmful for the production processes.

2. Many electronic products require clean rooms for manufacturing such things as integrated circuits, since their quality is adversely affected by airborne particles. Relative-humidity control is also needed to prevent corrosion and condensation and to eliminate static electricity. Temperature control maintains materials and instruments at stable condition and is also required for workers who wear dust-free garments. For example, a class 100 clean room in an electronic factory requires a temperature of 72 ± 2°F (22.2 ± 1.1°C), a relative humidity at 45 ± 5 percent, and a count of dust particles of 0.5-μm (1.97×10^{-5} in.) diameter or larger not to exceed 100 particles/ft^3 (3531 particles/m^3).

3. Precision manufacturers always need precise temperature control during production of precision instruments, tools, and equipment. Bausch and Lomb successfully constructed a constant-temperature control room of 68 ± 0.1°F (20 ± 0.56°C) to produce light grating products in the 1950s.

4. Pharmaceutical products require temperature, humidity, and air cleanliness control. For instance, liver extracts require a temperature of 75°F (23.9°C) and a relative humidity of 35 percent. If the temperature exceeds 80°F (26.7°C), the extracts tend to deteriorate. High-efficiency air filters must be installed for most of the areas in pharmaceutical factories to prevent contamination.

5. Modern refrigerated warehouses not only store commodities in coolers at temperatures of 27 to 32°F (− 2.8 to 0°C) and frozen foods at − 10 to − 20°F (− 23 to − 29°C), but also provide relative-humidity control for perishable foods between 90 and 100 percent. Refrigerated storage is used to prevent deterioration. Temperature control can be performed by refrigeration systems only, but the simultaneous control of both temperature and relative humidity in the space can only be performed by process air conditioning systems.

1.3 CLASSIFICATION OF AIR CONDITIONING SYSTEMS ACCORDING TO CONSTRUCTION AND OPERATING CHARACTERISTICS

Air conditioning systems can also be classified according to their construction and operating characteristics as follows.

Individual Room Air Conditioning Systems

Individual room, or simply individual air conditioning systems employ a single, self-contained room air conditioner, a packaged terminal, a separated indoor-outdoor split unit, or a heat pump. A heat pump extracts heat from a heat source and rejects heat to air or water at a higher temperature for heating. Unlike other systems, these systems normally use a totally independent unit or units in each room. Individual air conditioning systems can be classified into two categories:

- Room air conditioner (window-mounted)
- Packaged terminal air conditioner (PTAC), installed in a sleeve through the outside wall

The major components in a factory-assembled and ready-for-use room air conditioner include the following: An evaporator fan pressurizes and supplies the conditioned air to the space. In tube-and-fin coil, the refrigerant evaporates, expands directly inside the tubes, and absorbs the heat energy from the ambient air during the cooling season; it is called a direct expansion (DX) coil. When the hot refrigerant releases heat energy to the conditioned space during the heating season, it acts as a heat pump. An air filter removes airborne particulates. A compressor compresses the refrigerant from a lower evaporating pressure to a higher condensing pressure. A condenser liquefies refrigerant from hot gas to liquid and rejects heat through a coil and a condenser fan. A temperature control system senses the space air temperature (sensor) and starts or stops the compressor to control its cooling and heating capacity through a thermostat (refer to Chap. 26).

The difference between a room air conditioner and a room heat pump, and a packaged terminal air conditioner and a packaged terminal heat pump, is that a four-way reversing valve is added to all room heat pumps. Sometimes room air conditioners are separated into two split units: an outdoor condensing unit with compressor and condenser, and an indoor air handler in order to have the air handler in a more advantageous location and to reduce the compressor noise indoors.

Individual air conditioning systems are characterized by the use of a DX coil for a single room. This is the simplest and most direct way of cooling the air. Most of the individual systems do not employ connecting ductwork. Outdoor air is introduced through an opening or through a small air damper. Individual systems are usually used only for the perimeter zone of the building.

Evaporative-Cooling Air Conditioning Systems

Evaporative-cooling air conditioning systems use the cooling effect of the evaporation of liquid water to cool an airstream directly or indirectly. It could be a factory-assembled packaged unit or a field-built system. When an evaporative cooler provides only a portion of the cooling effect, then it becomes a component of a central hydronic or a packaged unit system.

An evaporative-cooling system consists of an intake chamber, filter(s), supply fan, direct-contact or indirect-contact heat exchanger, exhaust fan, water sprays, recirculating water pump, and water sump. Evaporative-cooling systems are characterized by low energy use compared with refrigeration cooling. They produce cool and humid air and are widely used in southwest arid areas in the United States (refer to Chap. 27).

Desiccant-Based Air Conditioning Systems

A desiccant-based air conditioning system is a system in which latent cooling is performed by desiccant dehumidification and sensible cooling by evaporative cooling or refrigeration. Thus, a considerable part of expensive vapor compression refrigeration is replaced by inexpensive evaporative cooling. A desiccant-based air conditioning system is usually a hybrid system of dehumidification, evaporative cooling, refrigeration, and regeneration of desiccant (refer to Chap. 29).

There are two airstreams in a desiccant-based air conditioning system: a process airstream and a regenerative airstream. Process air can be all outdoor air or a mixture of outdoor and recirculating

air. Process air is also conditioned air supplied directly to the conditioned space or enclosed manufacturing process, or to the air-handling unit (AHU), packaged unit (PU), or terminal for further treatment. Regenerative airstream is a high-temperature airstream used to reactivate the desiccant.

A desiccant-based air conditioned system consists of the following components: rotary desiccant dehumidifiers, heat pipe heat exchangers, direct or indirect evaporative coolers, DX coils and vapor compression unit or water cooling coils and chillers, fans, pumps, filters, controls, ducts, and piping.

Thermal Storage Air Conditioning Systems

In a thermal storage air conditioning system or simply thermal storage system, the electricity-driven refrigeration compressors are operated during off-peak hours. Stored chilled water or stored ice in tanks is used to provide cooling in buildings during peak hours when high electric demand charges and electric energy rates are in effect. A thermal storage system reduces high electric demand for HVAC&R and partially or fully shifts the high electric energy rates from peak hours to off-peak hours.

A thermal storage air conditioning system is always a central air conditioning system using chilled water as the cooling medium. In addition to the air, water, and refrigeration control systems, there are chilled-water tanks or ice storage tanks, storage circulating pumps, and controls (refer to Chap. 31).

Clean-Room Air Conditioning Systems

Clean-room or clean-space air conditioning systems serve spaces where there is a need for critical control of particulates, temperature, relative humidity, ventilation, noise, vibration, and space pressurization. In a clean-space air conditioning system, the quality of indoor environmental control directly affects the quality of the products produced in the clean space.

A clean-space air conditioning system consists of a recirculating air unit and a makeup air unit—both include dampers, prefilters, coils, fans, high-efficiency particulate air (HEPA) filters, ductwork, piping work, pumps, refrigeration systems, and related controls except for a humidifier in the makeup unit (refer to Chap. 30).

Space Conditioning Air Conditioning Systems

Space conditioning air conditioning systems are also called *space air conditioning systems*. They have cooling, dehumidification, heating, and filtration performed predominately by fan coils, water-source heat pumps, or other devices within or above the conditioned space, or very near it. A fan coil consists of a small fan and a coil. A water-source heat pump usually consists of a fan, a finned coil to condition the air, and a water coil to reject heat to a water loop during cooling, or to extract heat from the same water loop during heating. Single or multiple fan coils are always used to serve a single conditioned room. Usually, a small console water-source heat pump is used for each control zone in the perimeter zone of a building, and a large water-source heat pump may serve several rooms with ducts in the core of the building (interior zone, refer to Chap. 28).

Space air conditioning systems normally have only short supply ducts within the conditioned space, and there are no return ducts except the large core water-source heat pumps. The pressure drop required for the recirculation of conditioned space air is often equal to or less than 0.6 in. water column (WC) (150 Pa). Most of the energy needed to transport return and recirculating air is saved in a space air conditioning system, compared to a unitary packaged or a central hydronic air conditioning system. Space air conditioning systems are usually employed with a dedicated (separate) outdoor ventilation air system to provide outdoor air for the occupants in the conditioned space.

Space air conditioning systems often have comparatively higher noise level and need more periodic maintenance inside the conditioned space.

Unitary Packaged Air Conditioning Systems

Unitary packaged air conditioning systems can be called, in brief, packaged air conditioning systems or packaged systems. These systems employ either a single, self-contained packaged unit or two split units. A single packaged unit contains fans, filters, DX coils, compressors, condensers, and other accessories. In the split system, the indoor air handler comprises controls and the air system, containing mainly fans, filters, and DX coils; and the outdoor condensing unit is the refrigeration system, composed of compressors and condensers. Rooftop packaged systems are most widely used (refer to Chap. 29).

Packaged air conditioning systems can be used to serve either a single room or multiple rooms. A supply duct is often installed for the distribution of conditioned air, and a DX coil is used to cool it. Other components can be added to these systems for operation of a heat pump system; i.e., a centralized system is used to reject heat during the cooling season and to condense heat for heating during the heating season. Sometimes perimeter baseboard heaters or unit heaters are added as a part of a unitary packaged system to provide heating required in the perimeter zone.

Packaged air conditioning systems that employ large unitary packaged units are central systems by nature because of the centralized air distributing ductwork or centralized heat rejection systems. Packaged air conditioning systems are characterized by the use of integrated, factory-assembled, and ready-to-use packaged units as the primary equipment as well as DX coils for cooling, compared to chilled water in central hydronic air conditioning systems. Modern large rooftop packaged units have many complicated components and controls which can perform similar functions to the central hydronic systems in many applications.

1.4 CENTRAL HYDRONIC AIR CONDITIONING SYSTEMS

Central hydronic air conditioning systems are also called central air conditioning systems. In a central hydronic air conditioning system, air is cooled or heated by coils filled with chilled or hot water distributed from a central cooling or heating plant. It is mostly applied to large-area buildings with many zones of conditioned space or to separate buildings.

Water has a far greater heat capacity than air. The following is a comparison of these two media for carrying heat energy at 68°F (20°C):

	Air	Water
Specific heat, Btu/lb · °F	0.243	1.0
Density, at 68°F, lb/ft^3	0.075	62.4
Heat capacity of fluid at 68°F, Btu/ft^3 · °F	0.018	62.4

The heat capacity per cubic foot (meter) of water is 3466 times greater than that of air. Transporting heating and cooling energy from a central plant to remote air-handling units in fan rooms is far more efficient using water than conditioned air in a large air conditioning project. However, an additional water system lowers the evaporating temperature of the refrigerating system and makes a small- or medium-size project more complicated and expensive.

A central hydronic system of a high-rise office building, the NBC Tower in Chicago, is illustrated in Fig. 1.1. A central hydronic air conditioning system consists of an air system, a water system, a central heating/cooling plant, and a control system.

Air System

An air system is sometimes called the *air-handling system*. The function of an air system is to condition, to transport, to distribute the conditioned, recirculating, outdoor, and exhaust air, and to control the indoor environment according to requirements. The major components of an air system

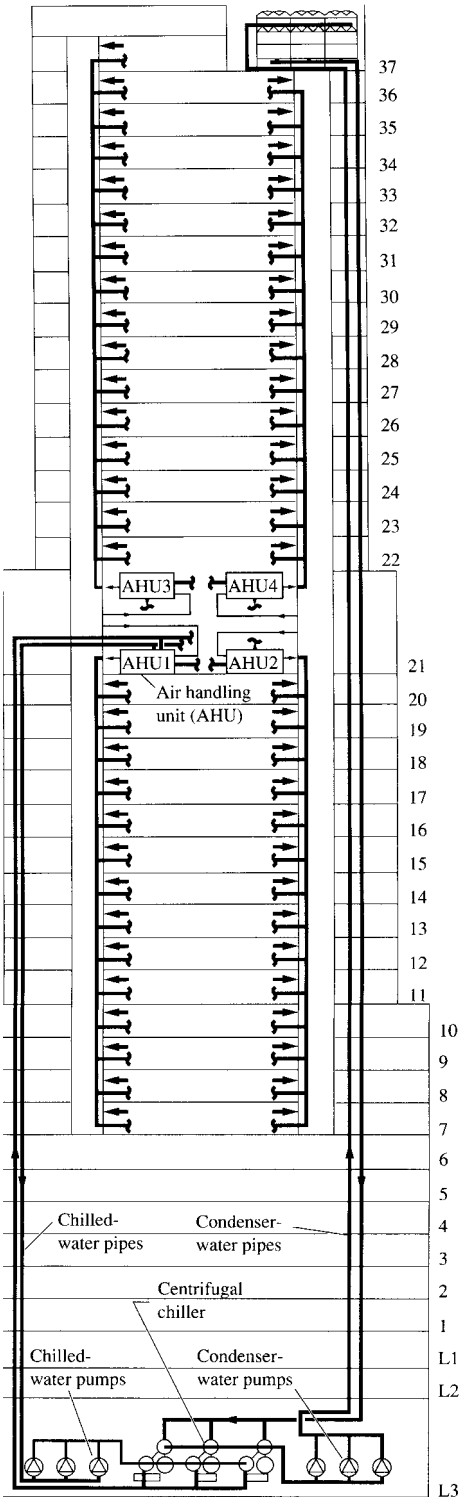

FIGURE 1.1 Schematic diagram of the central hydronic air conditioning system in NBC Tower.

are the air-handling units, supply/return ductwork, fan-powered boxes, space diffusion devices, and exhaust systems.

An air-handling unit (AHU) usually consists of supply fan(s), filter(s), a cooling coil, a heating coil, a mixing box, and other accessories. It is the primary equipment of the air system. An AHU conditions the outdoor/recirculating air, supplies the conditioned air to the conditioned space, and extracts the returned air from the space through ductwork and space diffusion devices.

A fan-powered variable-air-volume (VAV) box, often abbreviated as fan-powered box, employs a small fan with or without a heating coil. It draws the return air from the ceiling plenum, mixes it with the conditioned air from the air-handling unit, and supplies the mixture to the conditioned space.

Space diffusion devices include slot diffusers mounted in the suspended ceiling; their purpose is to distribute the conditioned air evenly over the entire space according to requirements. The return air enters the ceiling plenum through many scattered return slots.

Exhaust systems have exhaust fan(s) and ductwork to exhaust air from the lavatories, mechanical rooms, and electrical rooms.

The NBC Tower in Chicago is a 37-story high-rise office complex constructed in the late 1980s. It has a total air conditioned area of about 900,000 ft^2 (83,600 m^2). Of this, 256,840 ft^2 (23,870 m^2) is used by NBC studios and other departments, and 626,670 ft^2 (58,240 m^2) is rental offices located on upper floors. Special air conditioning systems are employed for NBC studios and departments at the lower level.

For the rental office floors, four air-handling units are located on the 21st floor. Outdoor air either is mixed with the recirculating air or enters directly into the air-handling unit as shown in Fig. 1.2. The mixture is filtrated at the filter and is then cooled and dehumidified at the cooling coil during cooling season. After that, the conditioned air is supplied to the typical floor through the supply fan, the riser, and the supply duct; and to the conditioned space through the fan-powered box and slot diffusers.

Water System

The water system includes chilled and hot water systems, chilled and hot water pumps, condenser water system, and condenser water pumps. The purpose of the water system is (1) to transport chilled water and hot water from the central plant to the air-handling units, fan-coil units, and fan-powered boxes and (2) to transport the condenser water from the cooling tower, well water, or other sources to the condenser inside the central plant.

In Figs. 1.1 and 1.2, the chilled water is cooled in three centrifugal chillers and then is distributed to the cooling coils of various air-handling units located on the 21st floor. The temperature of the chilled water leaving the coil increases after absorbing heat from the airstream flowing over the coil. Chilled water is then returned to the centrifugal chillers for recooling through the chilled water pumps.

After the condenser water has been cooled in the cooling tower, it flows back to the condenser of the centrifugal chillers on lower level 3. The temperature of the condenser water again rises owing to the absorption of the condensing heat from the refrigerant in the condenser. After that, the condenser water is pumped to the cooling towers by the condenser water pumps.

Central Plant

The refrigeration system in a central plant is usually in the form of a chiller package. Chiller packages cool the chilled water and act as a cold source in the central hydronic system. The boiler plant, consisting of boilers and accessories, is the heat source of the heating system. Either hot water is heated or steam is generated in the boilers.

In the NBC Tower, the refrigeration system has three centrifugal chillers located in lower level 3 of the basement. Three cooling towers are on the roof of the building. Chilled water cools from 58 to 42°F (14.4 to 5.6°C) in the evaporator when the refrigerant is evaporated. The refrigerant is then

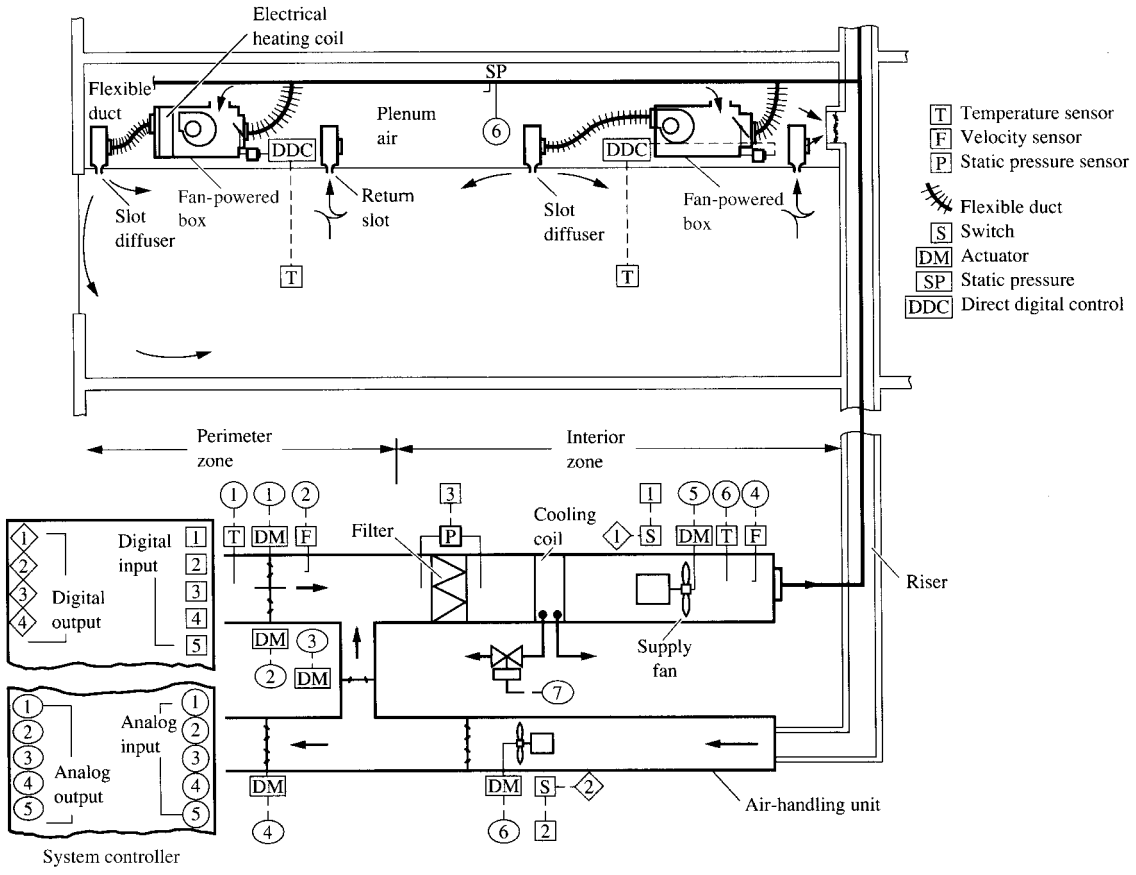

FIGURE 1.2 Schematic drawing of air system for a typical floor of offices in the NBC Tower.

compressed to the condensing pressure in the centrifugal compressor and is condensed in liquid form in the condenser, ready for evaporation in the evaporator.

There is no boiler in the central plant of the NBC Tower. To compensate heat loss in the perimeter zone, heat energy is provided by the warm plenum air and the electric heating coils in the fan-powered boxes.

Control System

Modern air conditioning control systems for the air and water systems and for the central plant consist of electronic sensors, microprocessor-operated and -controlled modules that can analyze and perform calculations from both digital and analog input signals, i.e., in the form of a continuous variable. Control systems using digital signals compatible with the microprocessor are called *direct digital control* (DDC) systems. Outputs from the control modules often actuate dampers, valves, and relays by means of pneumatic actuators in large buildings and by means of electric actuators for small projects.

In the NBC Tower, the HVAC&R system is monitored and controlled by a microprocessor-based DDC system. The DDC controllers regulate the air-handling units and the terminals. Both communicate with the central operating station through interface modules. In case of emergency, the

fire protection system detects alarm conditions. The central operating station gives emergency directions to the occupants, operates the HVAC&R system in a smoke control mode, and actuates the sprinkler water system.

Air, Water, Refrigeration, and Heating Systems

Air, water, refrigeration, heating, and control systems are the subsystems of an air conditioning or HVAC&R system. Air systems are often called secondary systems. Heating and refrigeration systems are sometimes called primary systems.

Central hydronic and space conditioning air conditioning systems both have air, water, refrigeration, heating, and control systems. The water system in a space conditioning system may be a chilled/hot water system. It also could be a centralized water system to absorb heat from the condenser during cooling, or provide heat for the evaporator during heating.

For a unitary packaged system, it consists of mainly air, refrigeration, and control systems. The heating system is usually one of the components in the air system. Sometimes a separate baseboard hot water heating system is employed in the perimeter zone.

A evaporative-cooling system always has an air system, a water system, and a control system. A separate heating system is often employed for winter heating.

In an individual room air conditioning system, air and refrigeration systems are installed in indoor and outdoor compartments with their own control systems. The heating system is often a component of the supply air chamber in the room air conditioner. It can be also a centralized hot water heating system in a PTAC system.

Air conditioning or HVAC&R systems are therefore often first described and analyzed through their subsystems and main components: such as air, water, heating, cooling/refrigeration, and control systems. Air conditioning system classification, system operating characteristics, and system selection must take into account the whole system.

Among the air, water, and refrigeration systems, the air system conditions the air, controls and maintains the required indoor environment, and has direct contact with the occupants and the manufacturing processes. These are the reasons why the operating characteristics of an air conditioning system are esssentially represented by its air system.

1.5 DISTRIBUTION OF SYSTEMS USAGE

According to surveys conducted in 1995 by the Department of Energy/Energy Information Administration (DOE/EIA) of the United States, for a total floor space of 58,772 million ft^2 (5462 million m^2) in commercial buildings in 1995 and for a total of 96.6 million homes in 1993 (among these, 74.1 million homes were air conditioned), the floor space, in million square feet, and the number of homes using various types of air conditioning systems are as follows:

	Million ft^2	Percent of floor space	Million homes	Percent of homes
Individual room systems	12,494	22	33.1	45
Evaporative-cooling systems	2,451	4		
Space conditioning systems (estimated)		(8)		
Unitary packaged systems (including air-source heat pump as well as desiccant-based systems)	26,628	48	41.0	55
Central hydronic systems (including thermal storage and clean-room systems)	13,586	24		
Others	949	2		

Much of the floor space may be included in more than one air conditioning system. Given the possibility that the floor space may be counted repeatedly, the original data listed in the DOE/EIA publication were modified. The 8 percent of the space system includes part in central hydronic systems and part in unitary packaged systems. Among the air conditioned homes in 1993, the unitary packaged system is the predominate air conditioning system in U.S. homes.

1.6 HISTORICAL DEVELOPMENT

The historical development of air conditioning can be summarized briefly.

Central Air Conditioning Systems

As part of a heating system using fans and coils, the first rudimentary ice system in the United States, designed by McKin, Mead, and White, was installed in New York City's Madison Square Garden in 1880. The system delivered air at openings under the seats. In the 1890s, a leading consulting engineer in New York City, Alfred R. Wolf, used ice at the outside air intake of the heating and ventilating system in Carnegie Hall. Another central ice system in the 1890s was installed in the Auditorium Hotel in Chicago by Buffalo Forge Company of Buffalo, New York. Early central heating and ventilating systems used steam-engine-driven fans. The mixture of outdoor air and return air was discharged into a chamber. In the top part of the chamber, pipe coils heat the mixture with steam. In the bottom part is a bypass passage with damper to mix conditioned air and bypass air according to the requirements.

Air conditioning was first systematically developed by Willis H. Carrier, who is recognized as the father of air conditioning. In 1902, Carrier discovered the relationship between temperature and humidity and how to control them. In 1904, he developed the air washer, a chamber installed with several banks of water sprays for air humidification and cleaning. His method of temperature and humidity regulation, achieved by controlling the dew point of supply air, is still used in many industrial applications, such as lithographic printing plants and textile mills.

Perhaps the first air-conditioned office was the Larkin Administration Building, designed by Frank L. Wright and completed in 1906. Ducts handled air that was drawn in and exhausted at roof level. Wright specified a refrigeration plant which distributed 10°C cooling water to air-cooling coils in air-handling systems.

The U.S. Capitol was air-conditioned by 1929. Conditioned air was supplied from overhead diffusers to maintain a temperature of 75°F (23.9°C) and a relative humidity of 40 percent during summer, and 80°F (26.7°C) and 50 percent during winter. The volume of supply air was controlled by a pressure regulator to prevent cold drafts in the occupied zone.

Perhaps the first fully air conditioned office building was the Milan Building in San Antonio, Texas, which was designed by George Willis in 1928. This air conditioning system consisted of one centralized plant to serve the lower floors and many small units to serve the top office floors.

In 1937, Carrier developed the conduit induction system for multiroom buildings, in which recirculation of space air is induced through a heating/cooling coil by a high-velocity discharging airstream. This system supplies only a limited amount of outdoor air for the occupants.

The variable-air-volume (VAV) systems reduce the volume flow rate of supply air at reduced loads instead of varying the supply air temperature as in constant-volume systems. These systems were introduced in the early 1950s and gained wide acceptance after the energy crisis of 1973 as a result of their lower energy consumption in comparison with constant-volume systems. With many variations, VAV systems are in common use for new high-rise office buildings in the United States today.

Because of the rapid development of space technology after the 1960s, air conditioning systems for clean rooms were developed into sophisticated arrangements with extremely effective air filters. Central air conditioning systems always will provide a more precisely controlled, healthy, and safe indoor environment for high-rise buildings, large commercial complexes, and precision-manufacturing areas.

Unitary Packaged Systems

The first room cooler developed by Frigidaire was installed about in 1928 or 1929, and the "Atmospheric Cabinet" developed by the Carrier Engineering Company was first installed in May 1931. The first self-contained room air conditioner was developed by General Electric in 1930. It was a console-type unit with a hermetically sealed motor-compressor (an arrangement in which the motor and compressor are encased together to reduce the leakage of refrigerant) and water-cooled condenser, using sulfur dioxide as the refrigerant. Thirty of this type of room air conditioner, were built and sold in 1931.

Early room air conditioners were rather bulky and heavy. They also required a drainage connection for the municipal water used for condensing. During the postwar period the air-cooled model was developed. It used outdoor air to absorb condensing heat, and the size and weight were greatly reduced. Annual sales of room air conditioners have exceeded 100,000 units since 1950.

Self-contained unitary packages for commercial applications, initially called *store coolers*, were introduced by the Airtemp Division of Chrysler Corporation in 1936. The early models had a refrigeration capacity of 3 to 5 tons and used a water-cooled condenser. Air-cooled unitary packages gained wide acceptance in the 1950s, and many were split systems incorporating an indoor air handler and an outdoor condensing unit.

Packaged units have been developed since the 1950s, from indoor to rooftops, from constant-volume to variable-air-volume, and from few to many functions. Currently, packaged units enjoy better performance and efficiency with better control of capacity to match the space load. Computerized direct digital control, one of the important reasons for this improvement, places unitary packaged systems in a better position to compete with central hydronic systems.

Refrigeration Systems

In 1844, Dr. John Gorrie designed the first commercial reciprocating refrigerating machine in the United States. The hermetically sealed motor-compressor was first developed by General Electric Company for domestic refrigerators and sold in 1924.

Carrier invented the first open-type gear-driven factory-assembled, packaged centrifugal chiller in 1922 in which the compressor was manufactured in Germany; and the hermetic centrifugal chiller, with a hermetically sealed motor-compressor assembly, in 1934. The direct-driven hermetic centrifugal chiller was introduced in 1938 by The Trane Company. Up to 1937, the capacity of centrifugal chillers had increased to 700 tons.

During the 1930s, one of the outstanding developments in refrigeration was the discovery by Midgely and Hene of the nontoxic, nonflammable, fluorinated hydrocarbon refrigerant family called Freon in 1931. Refrigerant-11 and refrigerant-12, the chlorofluorocarbons (CFCs), became widely adopted commercial products in reciprocating and centrifugal compressors. Now, new refrigerants have been developed by chemical manufacturers such as DuPont to replace CFCs, so as to prevent the depletion of the ozone layer.

The first aqueous-ammonia absorption refrigeration system was invented in 1815 in Europe. In 1940, Servel introduced a unit using water as refrigerant and lithium bromide as the absorbing solution. The capacities of these units ranged from 15 to 35 tons (52 to 123 kW). Not until 1945 did Carrier introduce the first large commercial lithium bromide absorption chillers. These units were developed with 100 to 700 tons (352 to 2460 kW) of capacity, using low-pressure steam as the heat source.

Positive-displacement screw compressors have been developed in the United States since the 1950s and scroll compressors since the 1970s because of their higher efficiency and smoother rotary motion than reciprocating compressors. Now, the scroll compressors gradually replace the reciprocating compressors in small and medium-size refrigeration systems. Another trend is the development of more energy-efficient centrifugal and absorption chillers for energy conservation. The energy consumption per ton of refrigeration of a new centrifugal chiller dropped from 0.80 kW/ton (4.4 COP_{ref}) in the late 1970s to 0.50 kW/ton (7.0 COP_{ref}) in the 1990s. A series of rotary motion refrigeration

compressors with small, medium, to large capacity and using scroll, screw, or centrifugal compressors will be manufactured from now on.

1.7 POTENTIALS AND CHALLENGES

Air conditioning or HVAC&R is an industry of many potentials and challenges, including the following.

Providing a Healthy and Comfortable Indoor Environment

Nowadays, people in the United States spend most of their time indoors. A healthy and comfortable indoor environment provided by air conditioning is a necessity for people staying indoors, no matter how hot or cold and dry or humid the outside climate might be. According to the American Housing Survey conducted by the U.S. Census Bureau in 1991, of 92.3 million homes, 66 million, or 71 percent of the total, were air conditioned, and 81.9 million, or nearly 89 percent of the total, were heated. According to the Energy Information Administration (EIA), in 1992, for a total floor area of 67.8 billion ft^2 (6.3 billion m^2) of commercial buildings, 84 percent were cooled and 91 percent were heated.

The Cleanest, Quietest, and Most Precise and Humid Processing Environment

A class 1 clean room to manufacture integrated circuits in a semiconductor factory may be the cleanest processing environment that is provided in the semiconductor industry in the 1990s. The dust particle count does not exceed 1 particle/ft^3 (35 particles/m^3) of a size of 0.5 μm and larger, with no particle exceeding 5 μm. A constant-temperature room of 68°F ± 0.1°F (20 ± 0.56°C) is always surrounded by other constant-temperature rooms or by a buffer area of lower tolerance to maintain the fluctuation of its sensed temperature within ± 0.05°F (0.028°C) during working hours.

A recording studio in a television broadcasting station often needs a noise criteria (NC) curve (refer to Chap. 4) of NC 15 to 20, in which a sound level less than that of a buzzing insect can be heard. A refrigerated warehouse that stores vegetables such as cabbage, carrots, and celery needs a temperature of 32°F (0°C) and a relative humidity of 98 to 100 percent to prevent deterioration and loss of water.

Air conditioning or HVAC&R systems will provide not only the cleanest, precisest, quietest, and most humid environment with fluctuations of the controlled variable within required limits, but also at optimum energy use and first cost.

Energy Use and Energy Efficiency

Based on the data published in the *Annual Energy Review* in 1993, the total energy use in 1992 in the United States was 82.14 quad Btu or 10^{15} Btu (86.66 EJ, or 10^{18} J). The United States alone consumed about one-fourth of the world's total production. Of the total energy use of 82.14 quad Btu in the United States, the residential/commercial sector consumed about 36 percent of the total, the industrial sector consumed another 36 percent, and transportation consumed the rest—28 percent. Petroleum, natural gas, and coal were the three main sources, providing more than 85 percent of the energy supply in 1992 in the United States.

According to DOE/EIA energy consumption survey 0321 for residential buildings (1993) and survey 0318 for commercial buildings (1995), the average annual energy use of HVAC&R systems

in the residential/commercial sector was about 45 percent of the total building energy consumption. Also assuming that the annual energy use of HVAC&R systems was about 1 percent of the total in both industrial and transportation sectors, then the estimate of annual energy use of the HVAC&R systems in 1992 in the United States was about 17 percent of the total national energy use, or one-sixth of the total national energy use.

The world energy resources of petroleum, natural gas, and coal are limited. The population of the United States in 1992 was only about one-twentieth of the world's total population; however, we consumed nearly one-fifth of the world's total energy produced. Energy use must be reduced.

After the energy crisis in 1973, the U.S. Congress enacted the Energy Policy and Conservation Act of 1975 and the National Energy Policy Act of 1992. The enactment of energy efficiency legislation by federal and state governments and the establishment of the Department of Energy (DOE) in 1977 had a definite impact on the implementation of energy efficiency in United States.

In 1975, the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) published Standard 90-75, *Energy Conservation in New Building Design*. This standard was revised and cosponsored by the Illuminating Engineering Society of North America as ASHRAE/IES Standard 90.1-1989, *Energy Efficient Design of New Buildings Except New Low-Rise Residential Buildings*, in 1989; and it was revised again as current ASHRAE/IESNA Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, in 1999. Many other organizations also offered valuable contributions for energy conservation. All these events started a new era in which energy efficiency has become one of the important goals of HVAC&R system design and operation.

Because of all these efforts, the increase in net annual energy use for residential and commercial buildings, i.e., the total energy use minus the electrical system energy loss in the power plant, from 1972 to 1992 in the United States was only 1.3 percent, as shown in Fig. 1.3. However, during the same period, the increase in floor area of commercial buildings in the United States was about 83

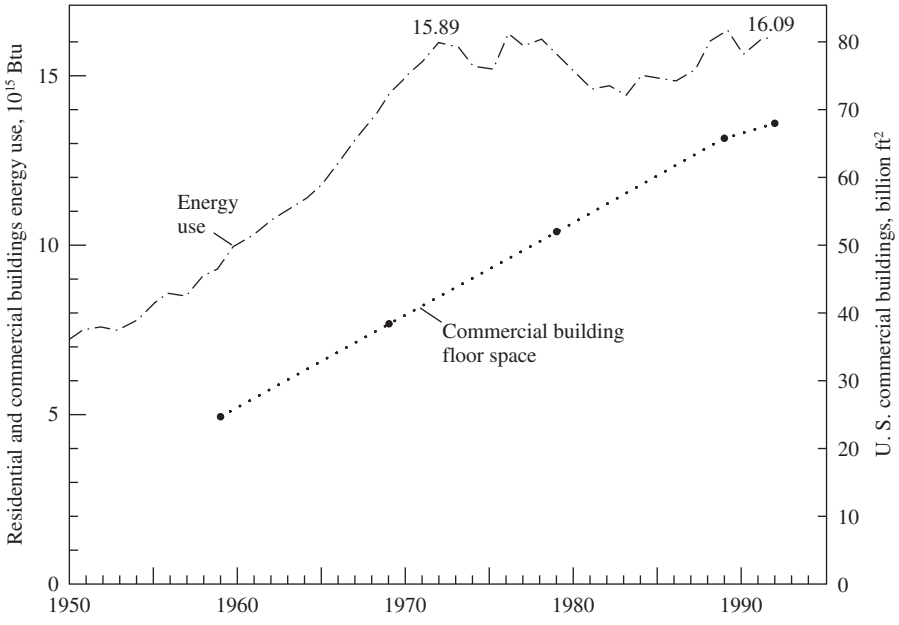

FIGURE 1.3 Residential and commercial building energy use and commercial building floor space increase between 1992 and 1950 in the United States.

percent. Energy efficiency will be a challenge to everyone in the HVAC&R industry in this generation as well as in many, many generations to come.

Environmental Problems—CFCs and Global Warming

The surface of the earth is surrounded by a layer of air, called the atmosphere. The lower atmosphere is called the homosphere, and the upper atmosphere is called the stratosphere.

In the mid-1980s, chlorofluorocarbons (CFCs) were widely used as refrigerants in mechanical refrigeration systems, to produce thermal insulation foam, and to produce aerosol propellants for many household consumer products. CFC-11 (CCl_3F) and CFC-12 (CCl_2F_2) are commonly used CFCs. They are very stable. Halons are also halogenated hydrocarbons. If CFCs and halons leak or are discharged from a refrigeration system during operation or repair to the lower atmosphere, they will migrate to the upper stratosphere and decompose under the action of ultraviolet rays throughout their decades or centuries of life. The free chlorine atoms will react with oxygen atoms of the ozone layer in the upper stratosphere and will cause a depletion of this layer. The theory of the depletion of the ozone layer was proposed in the early and middle-1970s. The ozone layer filters out harmful ultraviolet rays, which may cause skin cancer and are a serious threat to human beings. Furthermore, changes in the ozone layer may significantly influence weather patterns. Since 1996, actions have been taken to ban the production of CFCs and halons, before it is too late.

A cloudless homosphere is mainly transparent to short-wave solar radiation but is quite opaque to long-wave infrared rays emitted from the surface of the earth. Carbon dioxide (CO_2) has the greatest blocking effect of all; water vapor and synthetic CFCs also play important roles in blocking the direct escape of infrared energy. The phenomenon of transparency to incoming solar radiation and blanketing of outgoing infrared rays is called the *greenhouse effect*. The increase of the CO_2, water vapor, CFCs, and other gases, often called greenhouse gases (GHGs), eventually will result in a rise in air temperature near the earth's surface. This is known as the global warming effect.

Over the past 100 years, global warming has caused an increase of 1°F. For the same period, there was a 25 percent increase of CO_2. During the 1980s, the release of CO_2 to the atmosphere was responsible for about 50 percent of the increase in global warming that was attributable to human activity, and the release of CFCs had a 20 percent share. Some scientists have predicted an accelerated global warming in the coming 50 years because of the increase in the world's annual energy use. Further increases in global temperatures may lead to lower rainfalls, drop in soil moisture, more extensive forest fires, more flood, etc.

CFC production in developed countries has been banned since January 1, 1996. Carbon dioxide is the product of many combustion processes. Alternative refrigerants to replace CFCs must also have a low global warming potential. Designers and operators of the HVAC&R systems can reduce the production of CO_2 through energy efficiency and the replacement of coal, petroleum, and natural gas power by hydroelectric, solar, and nuclear energy, etc. More studies and research are needed to clarify the theory and actual effect of global warming.

Air Conditioning or HVAC&R Industry

The air conditioning or HVAC&R industry in the United States is an expanding and progressing industry. In 1995, the installed value of nonresidential air conditioning hit $20 billion. According to American Refrigeration Institute (ARI) and *Heating/Piping/Air Conditioning* data, from 1985 to 1995, the annual rate of increase of installed value of air conditioning systems is 8.7 percent. Because of the replacement of the old chillers using CFCs, in 1994, the installed value of retrofit, remodeling, and replacement accounted for up to two-thirds of all HVAC&R expenditures. This trend may continue in the beginning of the new century.

Based on data from ARI, in 1995, shipment of packaged units reached a record figure of 5 million products, of which heat pumps comprised a one-fifth share. Centrifugal and screw chiller

shipments were 7500 units, absorption chillers 502 units, and reciprocating chillers, at a significantly smaller capacity, were 14,000 units.

1.8 AIR CONDITIONING PROJECT DEVELOPMENT

Basic Steps in Development

The basic steps in the development and use of a large air conditioning system are the design, construction, commissioning, operation, energy efficiency upgrading, and maintenance. Figure 1.4 is a diagram which outlines the relationship between these steps and the parties involved. The owner sets the criteria and the requirements. Design professionals in mechanical engineering consulting

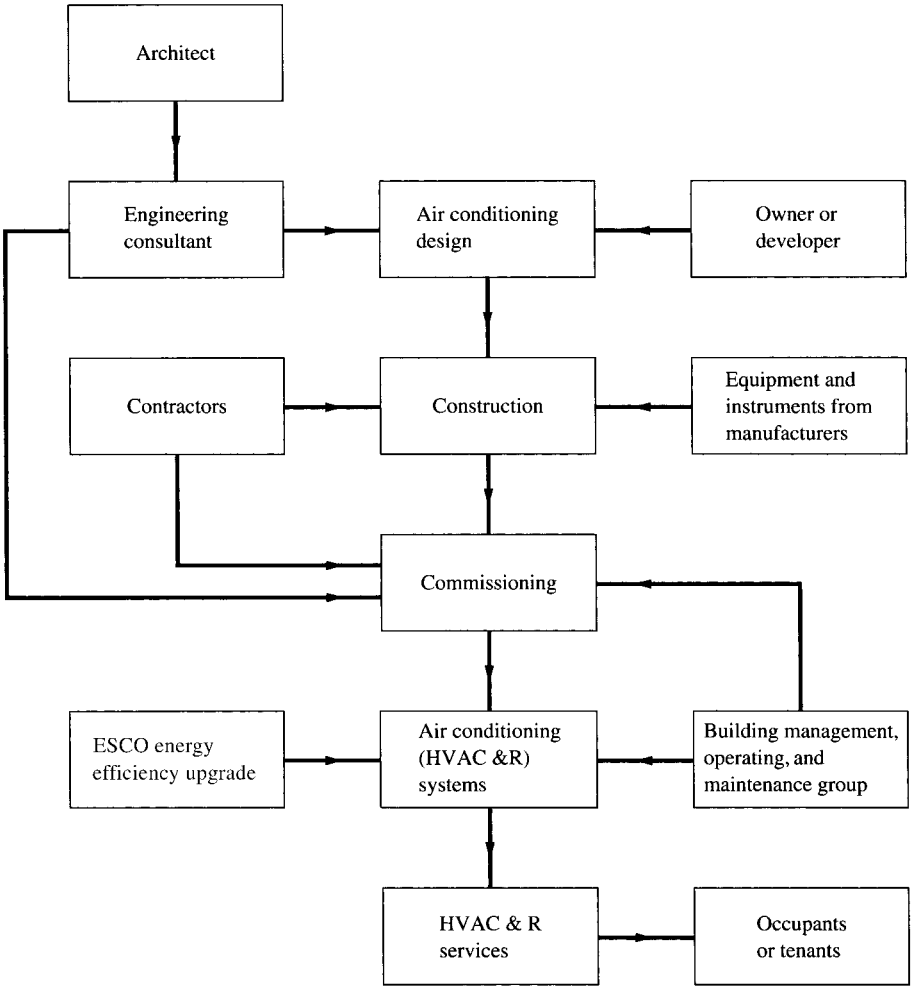

FIGURE 1.4 Steps in the development and use of air conditioning systems in buildings.

firms design the air conditioning system and prepare the design documents. Manufacturers supply the equipment, instruments, and materials. Contractors install and construct the air conditioning system. After construction, the air conditioning system is commissioned by a team, and then it is handed over to the operation and maintenance group of the property management for daily operation. Following a certain period of operation, an energy service company (ESCO) may often be required to upgrade the energy efficiency of the HVAC&R system (energy retrofit).

Design-Bid and Design-Build

There are two types of project development: design-bid and design-build. A design-bid project separates the design and installation responsibilities, whereas in a design-build project, engineering is done by the installing contractor. Some reasons for a design-build are that the project is too small to retain an engineering consultant, or that there is insufficient time to go through the normal design-bid procedures.

According to Bengard (1999), the main advantages of design-build include established firm price early, single-source responsibility, accelerated project delivery, and performance guarantees. The market has experienced nearly 300 percent domestic growth since 1986.

The Goal—An Environmentally Friendlier, Energy-Efficient, and Cost-Effective HVAC&R System

The goal is to provide an HVAC&R system which is environmentally friendlier, energy-efficient, and cost-effective as follows:

- Effectively control indoor environmental parameters, usually to keep temperature and humidity within required limits.
- Provide an adequate amount of outdoor ventilation air and an acceptable indoor air quality.
- Use energy-efficient equipment and HVAC&R systems.
- Minimize ozone depletion and the global warming effect.
- Select cost-effective components and systems.
- Ensure proper maintenance, easy after-hour access, and necessary fire protection and smoke control systems.

Major HVAC&R Problems

In Coad's paper (1985), a study by Wagner-Hohn-Ingles Inc. revealed that many large new buildings constructed in the early 1980s in the United States suffer from complaints and major defects. High on the list of these problems is the HVAC&R system. The major problems are these:

1. *Poor indoor air quality (IAQ)—sick building syndrome*. Poor indoor air quality causes the sick building syndrome. The National Institute of Occupational Safety and Health (NIOSH) in 1990 reported that between 1971 and 1988, 529 field investigations found that lack of outdoor air, improper use, and poor operation and maintenance of HVAC&R systems were responsible for more than one-half of sick building syndrome incidents. Field investigations found that 20 to 30 percent of the buildings had poor-air-quality problems. Sick building syndrome is covered in detail in Chap. 4.

2. *Updated technology*. In recent years, there has been a rapid change in the technology of air conditioning. Various types of VAV systems, air and water economizer, heat recovery, thermal storage, desiccant dehumidification, variable-speed drives, and DDC devices have become more effective and more advanced for energy efficiency. Many HVAC&R designers and operators are not

properly equipped to apply and use these systems. Unfortunately, these sophisticated systems are managed, constructed, and operated under the same budget and schedule constraints as the less sophisticated systems. After years of operation, most HVAC&R equipment and controls need to be upgraded for energy efficiency.

3. *Insufficient communication between design professionals, construction groups, and operators.* Effective operation requires a knowledgeable operator to make adjustments if necessary. The operator will operate the system at his or her level of understanding. If adequate operating and maintenance documents are not provided by the designer and the contractor for the operator, the HVAC&R system may not be operated according to the designer's intentions.

4. *Overlooked commissioning*. Commissioning means testing and balancing all systems, functional testing and adjusting of components and the integrated system, and adjusting and tuning the direct digital controls. An air conditioning system is different from the manufacturing products having models and prototypes. All the defects and errors of the prototypes can be checked and corrected during their individual tests, but the more complicated HVAC&R system, as constructed and installed, is the end product. Therefore, proper commissioning, which permits the system to perform as specified in the design documents, is extremely important. Unfortunately, the specifications seldom clearly designate competent technicians for the responsibility of commissioning the entire integrated system. Commissioning is covered in detail in Chap. 32.

5. *Reluctant to try innovative approaches*. In addition to Coad's paper, a survey was conducted by the American Consulting Engineers Council (ACEC) in 1995. For the 985 engineering firms that responded out of about 5500 firms total, there were 522 legal claims filed against them, 49 percent of suits resolved without payment. Among the 522 legal claims, 9 percent were filed against mechanical engineering firms. Fifty-two percent of 1995 legal claims were filed by owners, and 13 percent were filed by contractors or subcontractors.

Because of the legal claims, insurance companies discourage innovation, and engineering consulting firms are reluctant to try innovative techniques. Twenty-one percent of these firms said they were very much reluctant, 61 percent said they were somewhat affected, and 25 percent said they were a little affected. Support and encouragement must be given to engineering firms for carefully developed innovative approaches to projects. In addition to the ASHRAE Technology Awards, more HVAC&R innovative awards should be established in large cities in the United States to promote innovative approaches.

1.9 DESIGN FOR AIR CONDITIONING SYSTEM

System design determines the basic characteristics. After an air conditioning system is constructed according to the design, it is difficult and expensive to change the design concept.

Engineering Responsibilities

The normal procedure in a design-bid project includes the following steps and requirements:

1. Initiation of a construction project by owner or developer
2. Selection of design team
3. Setting of the design criteria and indoor environmental parameters
4. Selection of conceptual alternatives for systems and subsystems; preparation of schematic layouts of HVAC&R
5. Preparation of contract documents, working drawings, specifications, materials and construction methods, commissioning guidelines
6. Competitive bidding by contractors
7. Evaluation of bids; negotiations and modification of contract documents

8. Advice on awarding of contract
9. Review of shop drawings and commissioning schedule, operating and maintenance manuals
10. Monitoring, supervision, and inspection of construction
11. Supervision of commissioning: testing and balancing; functional performance tests
12. Modification of drawings to the as-built condition and the finalization of the operation and maintenance manual
13. Acceptance

Construction work starts at contract award following the bidding and negotiating and ends at the acceptance of the project after commissioning.

It is necessary for the designer to select among the available alternatives for optimum comfort, economics, energy conservation, noise, safety, flexibility, reliability, convenience, and maintainability. Experience, education, and judgment all enter into the selection process. If both a complicated system and a simple system yield the same performance, the simple system is preferred for its reliability, operator convenience, and lower cost.

Coordination between Air Conditioning and Other Trades, Teamwork

Air conditioning, plumbing, and fire protection systems are mechanical systems in a building. Both mechanical and electrical systems provide building services for the occupants and goods inside the building. Coordination between the mechanical engineer for HVAC&R and the architect, as well as between mechanical and structural or electrical engineers, or teamwork, becomes important.

Factors requiring input from both the architect and the mechanical engineer include the following:

1. Shape and the orientation of the building
2. Thermal characteristics of the building envelope, especially the type and size of the windows and the construction of external walls and roofs
3. Location of the ductwork and piping to avoid interference with each other, or with other trades
4. Layout of the diffusers and supply and return grilles
5. Minimum clearance provided between the structural members and the suspended ceiling for the installation of ductwork and piping
6. Location and size of the rooms for central plant, fan rooms, duct and pipe shafts

If the architect makes a decision that is thermally unsound, the HVAC&R engineer must offset the additional loads by increasing the HVAC&R system capacity. Lack of such coordination results in greater energy consumption.

HVAC&R and other building services must coordinate the following:

1. Utilization of daylight and the type of artificial light to be installed
2. The layout of diffusers, grilles, return inlets, and light troffers in the suspended ceiling
3. Integration with fire alarm and smoke control systems
4. Electric power and plumbing requirements for the HVAC&R equipment and lighting for equipment rooms
5. Coordination of the layout of the ductwork, piping, electric cables, etc.

Retrofit, Remodeling, and Replacement

Retrofit of an HVAC&R system must be tailored to the existing building and integrated with existing systems. Each retrofit project has a motivation which may be related to environment, safety and health, indoor environment, energy conservation, change of use, etc. Because of ozone depletion,

the production of CFCs ceased in the United States on January 1, 1996. All refrigeration systems using CFC refrigerants will be replaced by or converted to non-CFC refrigerant systems. For the sake of safety, smoke control systems and stairwell pressurization are items to be considered. For the sake of the occupants' health, an increase in the amount of outdoor air may be the right choice. If energy conservation is the first priority, the following items are among the many that may be considered: efficiency of lighting; energy consumption of the fans, pumps, chillers, and boilers; insulation of the building envelope; and the remodeling or replacement of fenestration. For improving the thermal comfort of the occupant, an increase of the supply volume flow rate, an increase of cooling and heating capacity, and the installation of appropriate controls may be considered.

Engineer's Quality Control

In 1988, the American Consulting Engineers Council chose "People, Professionalism, Profits: A Focus on Quality" as the theme of its annual convention in New York City. Quality does not mean perfection. In HVAC&R system design, quality means functionally effective, for health, comfort, safety, energy conservation, and cost. Quality also implies the best job, using accepted professional practices and talent. Better quality always means fewer complaints and less litigation. Of course, it also requires additional time and higher design cost. This fact must be recognized by the owner and developer.

The use of safety factors allows for the unpredictable in design, installation, and operation. For example, a safety factor of 1.1 for the calculated cooling load and of 1.1 to 1.15 for the calculated total pressure drop in ductwork is often used to take into account unexpected inferiority in fabrication and installation. An HVAC&R system should not be overdesigned by using a greater safety factor than is actually required. The initial cost of an overdesigned HVAC&R system is always higher, and it is energy-inefficient.

When an HVAC&R system design is under pressure to reduce the initial cost, some avenues to be considered are as follows:

1. Select an optimum safety factor.
2. Minimize redundancy, such as standby units.
3. Conduct a detailed economic analysis for the selection of a better alternative.
4. Calculate the space load, the capacity of the system, and the equipment requirements more precisely.
5. Adopt optimum diversity factors based on actual experience data observed from similar buildings.

A *diversity factor* is defined as the ratio of the simultaneous maximum load of a system to the sum of the individual maximum loads of the subdivisions of a system. It is also called the *simultaneous-use factor*. For example, in load calculations for a coil, block load is used rather than the sum of zone maxima for sizing coils for AHUs installed with modulating control for the chilled water flow rate.

The need for a higher-quality design requires that engineers have a better understanding of the basic principles, practical aspects, and updated technology of HVAC&R systems in order to avoid overdesign or underdesign and to produce a satisfactory product.

Design of the Control System

Controlling and maintaining the indoor environmental parameters within predetermined limits depends mainly on adequate equipment capacity and the quality of the control system. Energy can be saved when the systems are operated at part load with the equipment's capacity following the system load accurately by means of capacity control.

Because of the recent rapid change of HVAC&R controls from conventional systems to energy management systems, to DDC with microprocessor intelligence, and then to open-protocol BACnet, many designers have not kept pace. In 1982, Haines did a survey on HVAC&R control system design and found that many designers preferred to prepare a conceptual design and a sequence of operations and then to ask the representative of the control manufacturer to design the control system. Only one-third of the designers designed the control system themselves and asked the representative of the control manufacturer to comment on it.

HVAC&R system control is a decisive factor in system performance. Many of the troubles with HVAC&R arise from inadequate controls and/or their improper use. The designer should keep pace with the development of new control technology. He or she should be able to prepare the sequence of operations and select the best-fit control sequences for the controllers from a variety of the manufacturers that offer equipment in the HVAC&R field. The designer may not be a specialist in the details of construction or of wiring diagrams of controllers or DDC modules, but he or she should be quite clear about the function and sequence of the desired operation, as well as the criteria for the sensors, controllers, DDC modules, and controlled devices.

If the HVAC&R system designer does not perform these duties personally, preparation of a systems operation and maintenance manual with clear instructions would be difficult. It would also be difficult for the operator to understand the designer's intention and to operate the HVAC&R system satisfactorily.

Field Experience

It is helpful for the designer to visit similar projects that have been operated for more than 2 years and talk with the operator before initiating the design process. Such practice has many advantages.

1. The designer can investigate the actual performance and effectiveness of the air conditioning and control systems that he or she intends to design.
2. According to the actual operating records, the designer can judge whether the system is overdesigned, underdesigned, or the exact right choice.
3. Any complaints or problems that can be corrected may be identified.
4. The results of energy conservation measures can be evaluated from actual performance instead of theoretical calculations.
5. The designer can accumulate valuable practical experience from the visit, even from the deficiencies.

New Design Technologies

Computer-aided design and drafting (CADD) and knowledge-based system (KBS) or expert system assisted design are prospective new design technologies that have been used in HVAC&R design. CADD is covered in detail in Sec. 1.12, and KBS is covered in Chap. 5.

1.10 DESIGN DOCUMENTS

In the United States, the construction (installation) of air conditioning or HVAC&R systems is usually performed according to a contract between the owner or developer and the installer, the contractor. The owner specifies the work or job that is committed to be accomplished within a time period. The contractor shall furnish and install equipment, ductwork, piping, instruments, and the related material of the HVAC&R system for a given compensation. The construction of an air conditioning or HVAC&R system is usually a part of the construction of a building.

Both drawings and specifications are legal documents. They are legal because the designer conveys the requirements of the owner or developer to the contractor through these documents. Drawings and specifications define the work to be done by the contractor. They should be clear and should precisely and completely show the work to be accomplished. Drawings and specifications are complements to each other. Things to be defined should be shown or specified only once, either in drawing or in specification. Repetition often causes ambiguity and error.

Drawings

The layout of an HVAC&R system and the locations and dimensions of its equipment, instruments, ducts, pipes, etc., are best shown and illustrated by drawings. HVAC&R drawings consist of mainly the following:

- *Floor plans*. System layout including plant room, fan rooms, mechanical room, ductwork, and pipelines is always illustrated on floor plans. Each floor has at least one floor plan. HVAC&R floor plans are always drawn over the same floor plan of the architectural drawing.
- *Detail drawings*. These drawings show the details of a certain section of an HVAC&R system, or the detail of the installation of certain equipment, or the connection between equipment and ductwork or pipeline. Standard details are often used to save time.
- *Sections and elevations*. Sectional drawings are helpful to show the inner part of a section of a system, a piece of equipment, or a device. They are especially useful for places such as the plant room, fan room, and mechanical room where lots of equipment, ductwork, and pipelines are found. Elevations often show clearly the relationship between the HVAC&R components and the building structure.
- *Piping diagram*. This diagram shows the piping layout of the water system(s) and the flow of water from the central plant to the HVAC&R equipment on each floor.
- *Air duct diagram*. This diagram illustrates the air duct layout as well as the airflow from the air-handling unit or packaged unit to the conditioned spaces on each floor through space diffusion devices.
- *Control diagrams*. These diagrams show the zone level control systems, each type of functional control system for air-handling units or packaged units, water systems, outdoor air ventilation systems, sequencing of compressors, network communication, etc.
- *Equipment schedule*. This schedule provides the quatity and performance characteristics of the equipment or device in tabulated form as drawings. Drawings are always available to the installer at any time but the specifications are not. This is why the equipment schedule should appear on drawings instead of inside the specification.
- *Legends*. Symbols and abbreviations are often defined in a legend. ASHRAE has proposed a set of symbols in the *Fundamentals* handbook.

Sometimes, three-dimensional isometric drawings are necessay for piping and air duct diagrams. For a floor plan, a scale of 1/8 in. = 1 ft (1:100) is often used. The size of the drawings should be selected according to the size of the project. Drawing sheet sizes of 24 × 36 in., 30 × 42 in., and 36 × 48 in. (610 × 915 mm, 762 × 1067 mm, and 915 × 1219 mm) are widely adopted for large projects. After completion, every drawing should be carefully checked for errors and omissions.

Specifications

Detailed descriptions of equipment, instruments, ductwork, and pipelines, as well as performances, operating characteristics, and control sequences are better defined in specifications. Specifications usually consist of the legal contract between the owner and the contractor, installer, or vendor, and

the technical specifications that specify in detail the equipment and material to be used and how they are installed.

The Construction Specifications Institute (CSI) has developed a format, called the *Masterformat*, for specifications. This *Masterformat* is widely adopted by most HVAC&R construction projects. *Masterformat* promotes standardization and thereby facilitates the retrieval of information and improves construction communication. The 1988 edition of *Masterformat* consists of 16 divisions: 01000 General Requirements, 02000 Site Work, 03000 Concrete, . . ., 14000 Conveying Systems, 15000 Mechanical, and 16000 Electrical. In mechanical, it is subdivided into the following major sections:

15050 Basic Mechanical Materials and Methods
15250 Mechanical Insulation
15300 Fire Protection
15400 Plumbing
15500 Heating, Ventilating, and Air Conditioning
15550 Heat Generation
15650 Refrigeration
15750 Heat Transfer
15850 Air Handling
15880 Air Distribution
15950 Controls
15990 Testing, Adjusting, and Balancing

Each of the above-mentioned sections may contain: general considerations, equipment and material, and field installation.

According to whether the wanted vendor is specified or not, specifications can be classified into two categories: (1) performance specification, which depends only on the performance criteria, and (2) or-equal specification, which specifies the wanted vendor.

Here are some recommendations for writing an HVAC&R specification:

- The required indoor environmental parameters to be maintained in the conditioned space during summer and winter outdoor design conditions, such as temperature, humidity, outdoor ventilation air rates, air cleanliness, sound level, and space pressure, shall be clearly specified in the general consideration of section 15500.
- Use simple, direct, and clear language without repetition.
- All the terms must be well defined and written in a consistent manner.
- Don't write specifications or refer to other works without having personal knowledge of the content or even understanding its meaning.
- All the specifications must be tailored to fit the designed HVAC&R system. Never list an item that is not listed in the specified HVAC&R system, such as the return ducts in a fan-coil system.

1.11 CODES AND STANDARDS

Codes are generally mandatory state or city laws or regulations that force the designer to create the design without violating human safety and welfare. State and city codes concerning structural integrity, electrical safety, fire protection, and prevention of explosion of pressure vessels must be followed.

Standards describe consistent methods of testing, specify confirmed design guidelines, and recommend standard practices. Conformance to standards is usually voluntary. However, if design

criteria or system performance is not covered in local codes, ASHRAE Standards become the vital document to assess the indoor air quality in a lawsuit against designer or contractor.

Since the energy crisis of 1973, the U. S. federal government and Congress have published legislation which profoundly affects the design and operation of HVAC&R systems. As mentioned before, in 1975, the Energy Policy and Conservation Act and in 1978 the National Energy Conservation Policy Act were enacted for energy conservation. Also in 1992, the Clean Air Act Amendments phase out the use of CFCs and recycling of refrigerants, to protect the ozone layer as well as to provide an acceptable indoor air quality.

Among the HVAC&R related standards published by various institutions, the following by ASHRAE and some other institutions have greatly influenced the design and operation of air conditioning systems:

- ASHRAE/IES Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, which is often called the Energy Standard. This standard is under continuous maintenance by a standing standard committee (SSPC) for regular publication of addenda or revisions.
- ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality*, which is often called the Indoor Air Quality Standard. This standard is under continuous maintenance.
- ANSI/ASHRAE Standard 55-1992, *Thermal Environmental Conditions for Human Occupancy*
- ASHRAE Standard 15-1992, *Safety Code for Mechanical Refrigeration.*
- ANSI/ASHRAE Standard 34-1997, *Number Designation and Safety Classification of Refrigerants*. This standard is under continuous maintenance.
- ANSI/ASHRAE Standard 135-1995, *BACnet: A Data Communication Protocol for Building Automation and Control Networks.*
- Air Conditioning and Refrigeration Institute, *Standards for Unitary Equipment.*
- National Fire Protection Association NFPA 90A, *Standard for the Installation of Air Conditioning and Ventilating Systems.*
- Sheet Metal and Air Conditioning Contractors' National Association, *HVAC Duct Construction Standards—Metal and Flexible.*

These standards are covered in detail in later chapters.

Recently, many HVAC&R consulting firms seek compliance with the International Organization for Standardization (ISO) 9000 for quality control. ISO is a specialized international agency for standardization, at present comprising the national standards institutions of about 90 countries. The American National Standards Institue (ANSI) is a member that represents the United States.

The ISO 9000 series has now become the quality mangement standard worldwide. It set the guidelines for management participation, design control, review of specifications, supplier monitoring, and services provided. The goal of ISO 9000 is to guarantee that products or the services provided by a manufacturer or an engineering consulting firm are appropriate to the specifications and are within the tolerances agreed upon. The ISO 9000 series consists of five quality system standards:

ISO 9000	*Guidelines of the Selection and the Application of Quality Management and Quality Assurance Standards*
ISO 9001	*Modeling of Quality Assurance in Design Development, Manufacturing, Installation and Servicing*
ISO 9002	*Modeling of Quality Assurance in Production and Installation*
ISO 9003	*Modeling of Quality Assurance in Final Inspection and Testing*
ISO 9004	*Guidelines for the Implementation of Quality Management and to Provide Quality Systems*

There are no legal requirements to be registered to ISO 9000 standards. Meeting ISO 9000 standards does not mean that the firm provides better service or products than those of nonregistered companies.

Many companies are assembling quality assurance programs which show that they are accepting compliance to ISO 9000 standards instead of registration. Such an arrangement will identify the ISO 9000 standard that can be best applied to your operation and also support your customer's ISO 9000 quality audit.

1.12 COMPUTER-AIDED DESIGN AND DRAFTING (CADD)

Because personal computers (PCs) provide a low-cost tool for computations and graphics and owing to the availability of lots of design computer software during the 1980s, computer-aided design and drafting (CADD) for air conditioning systems has grown quickly in recent years. Today, computer software is often used to develop design documents, drawings, and specifications in common use in engineering consulting firms. According to the 1994 CADD Application and User Survey of design firms in *Engineering Systems* (June 1994), ". . . firms with high productivity reported that they perform 95 percent of projects on CADD."

In addition to the CADD, there is software available for computer-aided facilities management (CAFM) for operation and maintenance.

Features of CADD

Compared with conventional design and drafting, CADD of air conditioning systems has the following features:

- Different trade designers such as those in architecture, HVAC&R, plumbing, fire protection, and electrical engineers can merge their work in plot output files to produce composite drawings in different layers. Mechanical system design can integrate with equipment selection programs.
- Graphical model construction of air conditioning systems and subsystems in two-dimensional (2D) or three-dimensional (3D) presentation uses architectural and structural models as backgrounds.
- CADD links the engineering design models based on calculation and the graphical model based on drafting.
- It provides the ability to develop and compare the alternate design schemes quickly as well as the capability to redesign or to match the changes during construction promptly.
- CADD establishes database libraries for design and graphical models.
- It develops design documents including as-built drawings and equipment schedules.

A saving of design time up to 40 percent has been claimed by some engineering consulting firms.

Computer-Aided Design

Current CADD for HVAC&R systems can be classified into two categories: (1) engineering design and simulation, and (2) drafting graphical model construction. The software for engineering design can again be subdivided into the following groups:

- *Load calculations, energy simulation, and psychrometric analysis*. Software for space load calculations could be a part of the energy simulation software. However, load calculations are often mainly used to determine the peak or block load at design conditions. These loads are the inputs for equipment selection. However, the software for energy simulation is used mainly to determine the optimum selection from different alternatives. Psychrometric analysis is sometimes a useful tool for load calculation and energy simulation.

- *Equipment selection*. All large HVAC&R equipment manufacturers offer software, often called an electronic catalog, to help customers select their product according to the capacity, size, configuration, and performance.
- *Equipment schedule and specifications*. Software generates equipment schedules or CSI masterformat for specifications.
- *Price list*. Software reports the price of base unit, accessories, and materials.

Computer-Aided Drafting (CAD)

- Software to reproduce architectural drawings is the foundation of CADD. Graphical models of items such as ducts, pipes, equipment, and system components in HVAC&R systems are developed against backgrounds of architecture and structural models. Automated computer-aided drafting (AutoCAD) is the leading PC-based drafting software used by most design firms to assist the designer during design with automated drawings, databases, and layering schemes.
- Software applications to develop duct and piping work layouts are the two primary CADs used in air conditioning system design. They can interface with architectural, electrical, and plumbing drawings through AutoCAD. CAD for duct and piping work can also have hydraulic modeling capacities. Tags and an HVAC&R equipment schedule including components and accessories can be produced as well. CAD for ducts and piping work is covered in detail in corresponding chapters.
- Software is also available to produce details of the refrigerating plant, heating plant, and fan room with accessories, duct, and piping layout. Many manufacturers also supply libraries of files for AutoCAD users to input the details of their products into CAD drawings.
- HVAC&R CAD for ducts and piping work layout and details often uses graphical model construction. The elements of the graphical model are linked to its associated information and stored in the databases. Model construction starts by locating an end device such as a diffuser or terminal. Add the supply and return pipes from different layers. Software for graphical model construction has the features of copying, repeating, moving, rotating, and mirror-imaging which enable the designer to construct the graphical models quickly. After supply and return main ducts, equipment, and components are added, this completes the supply and return duct systems of this floor. Duct and piping layouts can also be accomplished by drawing a single-line layout on the CAD, and the software will convert to two-dimensional or three-dimensional drawings.

Software Requirements

- A CADD software tailored to a specific HVAC&R design or drafting purpose is needed.
- Built-in error checking and error finding are necessary features.
- Transferability of drawings and data among different CADD systems is important. ASHRAE Handbook 1995, *HVAC Applications*, recommended that software interface to the most prominent formats: the Initial Graphics Exchange Specification (IGES), the Standard Exchange Format (SEF), and the Data Exchange Format (DXF).
- While using the many available computer software programs for loads, energy, and hydraulic property calculations, one should verify and calibrate the results against field-measured values in order to improve the effectiveness of the computer software when the conditions and affecting factors may be varied.

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CHAPTER 2
PSYCHROMETRICS

2.1 PSYCHROMETRICS 2.1
Moist Air 2.1
Equation of State of an Ideal Gas 2.2
Equation of State of a Real Gas 2.2
Calculation of the Properties of Moist Air 2.3
2.2 DALTON'S LAW AND THE GIBBS-DALTON LAW 2.3
2.3 AIR TEMPERATURE 2.4
Temperature and Temperature Scales 2.4
Thermodynamic Temperature Scale 2.5
Temperature Measurements 2.6
2.4 HUMIDITY 2.7
Humidity Ratio 2.7
Relative Humidity 2.7
Degree of Saturation 2.8
2.5 PROPERTIES OF MOIST AIR 2.8
Enthalpy 2.8
Moist Volume 2.9
Density 2.10
Sensible Heat and Latent Heat 2.10
Specific Heat of Moist Air at Constant Pressure 2.10
Dew-Point Temperature 2.11
2.6 THERMODYNAMIC WET-BULB TEMPERATURE AND THE WET-BULB TEMPERATURE 2.11
Ideal Adiabatic Saturation Process 2.11
Thermodynamic Wet-Bulb Temperature 2.12
Heat Balance of an Ideal Adiabatic Saturation Process 2.12
Psychrometer 2.12
Wet-Bulb Temperature 2.12
Relationship between Wet-Bulb Temperature and Thermodynamic Wet-Bulb Temperature 2.14
2.7 SLING AND ASPIRATION PSYCHROMETERS 2.14
2.8 HUMIDITY MEASUREMENTS 2.16
Mechanical Hygrometers 2.16
Electronic Hygrometers 2.16
Comparison of Various Methods 2.19
2.9 PSYCHROMETRIC CHARTS 2.19
2.10 DETERMINATION OF THERMODYNAMIC PROPERTIES ON PSYCHROMETRIC CHARTS 2.22
Computer-Aided Psychrometric Calculation and Analysis 2.25
REFERENCES 2.25

2.1 PSYCHROMETRICS

Psychrometrics is the study of the thermodynamic properties of moist air. It is used extensively to illustrate and analyze the characteristics of various air conditioning processes and cycles.

Moist Air

The surface of the earth is surrounded by a layer of air called the atmosphere, or atmospheric air. From the point of view of psychrometrics, the lower atmosphere, or homosphere, is a mixture of dry air (including various contaminants) and water vapor, often known as moist air.

The composition of dry air is comparatively stable. It varies slightly according to geographic location and from time to time. The approximate composition of dry air by volume percent is the following:

Nitrogen	78.08
Oxygen	20.95
Argon	0.93
Carbon dioxide	0.03
Other gases such as neon, sulfur dioxide, etc.	0.01

The amount of water vapor present in moist air at a temperature range of 0 to 100°F (−17.8 to 37.8°C) varies from 0.05 to 3 percent by mass. It has a significant influence on the characteristics of moist air.

Water vapor is lighter than air. A cloud in the sky is composed of microscopic beads of liquid water that are surrounded by a thin layer of water vapor. These layers give the cloud the needed buoyancy to float in the air.

Equation of State of an Ideal Gas

The equation of state of an ideal gas indicates the relationship between its thermodynamic properties, or

$$pv = RT_R \tag{2.1}$$

where p = pressure of gas, psf (Pa)
v = specific volume of gas, ft^3/lb (m^3/kg)
R = gas constant, $ft \cdot lb_f/lb_m \cdot °R$ ($J/kg \cdot K$)
T_R = absolute temperature of gas, °R (K)

Since $v = V/m$, then Eq. (2.1) becomes

$$pv = mRT_R \tag{2.2}$$

where V = total volume of gas, ft^3 (m^3)
m = mass of gas, lb (kg)

Using the relationship $m = nM$, and $R = R_o/M$, we can write Eq. (2.2) as

$$pv = nR_oT_R \tag{2.3}$$

where n = number of moles, mol
M = molecular weight
R_o = universal gas constant, $ft \cdot lb_f/lb_m \cdot °R$ ($J/mol \cdot K$)

Equation of State of a Real Gas

A modified form of the equation of state for a real gas can be expressed as

$$\frac{pv}{RT_R} = 1 + Ap + Bp^2 + Cp^3 + \cdots = Z \tag{2.4}$$

where A, B, C, $\cdots$ = virial coefficients and Z = compressibility factor. The compressibility factor Z illustrates the degree of deviation of the behavior of the real gas, moist air, from the ideal gas due to the following:

1. Effect of air dissolved in water

2. Variation of the properties of water vapor attributable to the effect of pressure

3. Effect of intermolecular forces on the properties of water vapor itself

For an ideal gas, $Z = 1$. According to the information published by the former National Bureau of Standards of the United States, for dry air at standard atmospheric pressure (29.92 in. Hg, or 760 mm Hg) and a temperature of 32 to 100°F (0 to 37.8°C) the maximum deviation is about 0.12 percent. For water vapor in moist air under saturated conditions at a temperature of 32 to 100°F (0 to 37.8°C), the maximum deviation is about 0.5 percent.

Calculation of the Properties of Moist Air

The most exact calculation of the thermodynamic properties of moist air is based on the formulations developed by Hyland and Wexler of the U.S. National Bureau of Standards. The psychrometric chart and tables of ASHRAE are constructed and calculated from these formulations.

Calculations based on the ideal gas equations are the simplest and can be easily formulated. According to the analysis of Nelson and Pate, at a temperature between 0 and 100°F (−17.8 and 37.8°C), calculations of enthalpy and specific volume using ideal gas equations show a maximum deviation of 0.5 percent from the exact calculations by Hyland and Wexler. Therefore, ideal gas equations will be used in this text for the formulation and calculation of the thermodynamic properties of moist air.

Although air contaminants may seriously affect the health of occupants of the air conditioned space, they have little effect on the thermodynamic properties of moist air since their mass concentration is low. For simplicity, moist air is always considered as a binary mixture of dry air and water vapor during the analysis and calculation of its properties.

2.2 DALTON'S LAW AND THE GIBBS-DALTON LAW

Dalton's law shows that for a mixture of gases occupying a given volume at a certain temperature, the total pressure of the mixture is equal to the sum of the partial pressures of the constituents of the mixture, i.e.,

$$p_m = p_1 + p_2 + \cdots \tag{2.5}$$

where p_m = total pressure of mixture, psia (Pa)
$p_1, p_2, \ldots$ = partial pressure of constituents 1, 2, . . . , psia (Pa)

The partial pressure exerted by each constituent in the mixture is independent of the existence of other gases in the mixture. Figure 2.1 shows the variation of mass and pressure of dry air and water vapor, at an atmospheric pressure of 14.697 psia (101,325 Pa) and a temperature of 75°F (23.9°C).

The principle of conservation of mass for nonnuclear processes gives the following relationship:

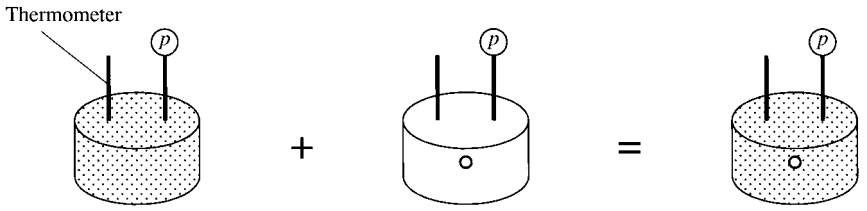
$$m_m = m_a + m_w \tag{2.6}$$

where m_m = mass of moist air, lb (kg)
m_a = mass of dry air, lb (kg)
m_w = mass of water vapor, lb (kg)

Applying Dalton's law for moist air, we have

$$p_{\text{at}} = p_a + p_w \tag{2.7}$$

where p_{at} = atmospheric pressure or pressure of the outdoor moist air, psia (Pa)
p_a = partial pressure of dry air, psia (Pa)
p_w = partial pressure of water vapor, psia (Pa)



$T = 75\ °F$	$T = 75\ °F$	$T = 75\ °F$
$m_a = 1$ lb	$m_a = 0$	$m_a = 1$ lb
$m_w = 0$	$m_w = 0.0092$ lb	$m_w = 0.0092$ lb
$p_a = 14.482$ psia	$p_a = 0$	$p_a = 14.482$ psia
$p_w = 0$	$p_w = 0.215$ psia	$p_w = 0.215$ psia
p (pressure of mixture) = 14.482 psia	$p = 0.215$ psia	$p_{at} = 14.697$ psia

FIGURE 2.1 Mass and pressure of dry air, water vapor, and moist air.

Dalton's law is based on experimental results. It is more accurate for gases at low pressures. Dalton's law can be further extended to state the relationship of the internal energy, enthalpy, and entropy of the gases in a mixture as the Gibbs-Dalton law:

$$\begin{aligned} m_m u_m &= m_1 u_1 + m_2 u_2 + \cdots \\ m_m h_m &= m_1 h_1 + m_2 h_2 + \cdots \\ m_m s_m &= m_1 s_1 + m_2 s_2 + \cdots \end{aligned} \tag{2.8}$$

where m_m = mass of gaseous mixture, lb (kg)
$m_1, m_2, \ldots$ = mass of the constituents, lb (kg)
u_m = specific internal energy of gaseous mixture, Btu/lb (kJ/kg)
$u_1, u_2, \ldots$ = specific internal energy of constituents, Btu/lb (kJ/kg)
h_m = specific enthalpy of gaseous mixture, Btu/lb (kJ/kg)
$h_1, h_2, \ldots$ = specific enthalpy of constituents, Btu/lb (kJ/kg)
s_m = specific entropy of gaseous mixture, Btu/lb·°R (kJ/kg·K)
$s_1, s_2, \ldots$ = specific entropy of constituents, Btu/lb·°R (kJ/kg·K)

2.3 AIR TEMPERATURE

Temperature and Temperature Scales

The temperature of a substance is a measure of how hot or cold it is. Two systems are said to have equal temperatures only if there is no change in any of their observable thermal characteristics when they are brought into contact with each other. Various temperature scales commonly used to measure the temperature of various substances are illustrated in Fig. 2.2.

In conventional inch-pound (I-P) units, at a standard atmospheric pressure of 14.697 psia (101,325 Pa), the Fahrenheit scale has a freezing point of 32°F (0°C) at the ice point, and a boiling point of 212°F (100°C). For the triple point with a pressure of 0.08864 psia (611.2 Pa), the magnitude on the Fahrenheit scale is 32.018°F (0.01°C). There are 180 divisions, or degrees, between the boiling and freezing points in the Fahrenheit scale. In the International System of Units (SI units), the Celsius or Centigrade scale has a freezing point of 0°C and a boiling point of 100°C. There are

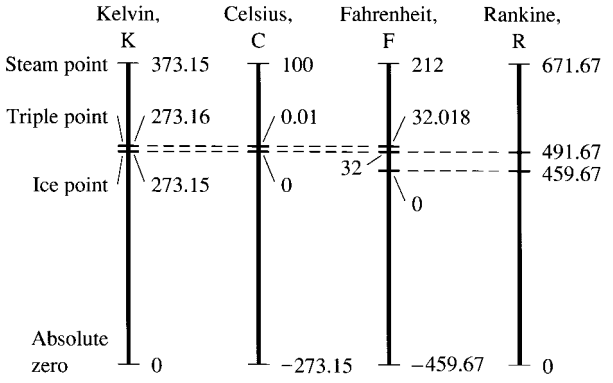


FIGURE 2.2 Commonly used temperature scales.

100 divisions between these points. The triple point is at 0.01°C. The conversion from Celsius scale to Fahrenheit scale is as follows:

$$°F = 1.8(°C) + 32 \tag{2.9}$$

For an ideal gas, at $T_R = 0$, the gas would have a vanishing specific volume. Actually, a real gas has a negligible molecular volume when T_R approaches absolute zero. A temperature scale that includes absolute zero is called an *absolute* temperature scale. The Kelvin absolute scale has the same boiling-freezing point division as the Celsius scale. At the freezing point, the Kelvin scale is 273.15 K. Absolute zero on the Celsius scale is −273.15°C. The Rankine absolute scale division is equal to that of the Fahrenheit scale. The freezing point is 491.67°R. Similarly, absolute zero is −459.67°F on the Fahrenheit scale.

Conversions between Rankine and Fahrenheit and between Kelvin and Celsius systems are

$$R = 459.67 + °F \tag{2.10}$$

$$K = 273.15 + °C \tag{2.11}$$

Thermodynamic Temperature Scale

On the basis of the second law of thermodynamics, one can establish a temperature scale that is independent of the working substance and that provides an absolute zero of temperature; this is called a *thermodynamic temperature scale*. The thermodynamic temperature T must satisfy the following relationship:

$$\frac{T_R}{T_{Ro}} = \frac{Q}{Q_o} \tag{2.12}$$

where Q = heat absorbed by reversible engine, Btu/h (kW)
Q_o = heat rejected by reversible engine, Btu/h (kW)
T_R = temperature of heat source of reversible engine, °R (K)
T_{Ro} = temperature of heat sink of reversible engine, °R (K)

Two of the ASHRAE basic tables, "Thermodynamic Properties of Moist Air" and "Thermodynamic Properties of Water at Saturation," in *ASHRAE Handbook 1993*, *Fundamentals*, are based on the thermodynamic temperature scale.

Temperature Measurements

During the measurement of air temperatures, it is important to recognize the meaning of the terms *accuracy*, *precision*, and *sensitivity*.

1. *Accuracy* is the ability of an instrument to indicate or to record the true value of the measured quantity. The error indicates the degree of accuracy.
2. *Precision* is the ability of an instrument to give the same reading repeatedly under the same conditions.
3. *Sensitivity* is the ability of an instrument to indicate change of the measured quantity.

Liquid-in-glass instruments, such as mercury or alcohol thermometers, were commonly used in the early days for air temperature measurements. In recent years, many liquid-in-glass thermometers have been replaced by remote temperature monitoring and indication systems, made possible by sophisticated control systems. A typical air temperature indication system includes sensors, amplifiers, and an indicator.

Sensors. Air temperature sensors needing higher accuracy are usually made from resistance temperature detectors (RTDs) made of platinum, palladium, nickel, or copper wires. The electrical resistance of these resistance thermometers characteristically increases when the sensed ambient air temperature is raised; i.e., they have a positive temperature coefficient α. In many engineering applications, the relationship between the resistance and temperature can be given by

$$R \approx R_{32}(1 + \alpha T)$$
$$\alpha \approx \frac{R_{212} - R_{32}}{180\, R_{32}} \tag{2.13}$$

where R = electric resistance, Ω
R_{32}, R_{212} = electric resistance, at 32 and 212°F (0 and 100°C), respectively, Ω
T = temperature, °F (°C)

The mean temperature coefficient α for several types of metal wires often used as RTDs is shown below:

	Measuring range,°F	α, Ω/°F
Platinum	−400 to 1350	0.00218
Palladium	400 to 1100	0.00209
Nickel	−150 to 570	0.0038
Copper	−150 to 400	0.0038

Many air temperature sensors are made from thermistors of sintered metallic oxides, i.e., semiconductors. They are available in a large variety of types: beads, disks, washers, rods, etc. Thermistors have a negative temperature coefficient. Their resistance decreases when the sensed air temperature increases. The resistance of a thermistor may drop from approximately 3800 to 3250 Ω when the sensed air temperature increases from 68 to 77°F (20 to 25°C). Recently developed high-quality thermistors are accurate, stable, and reliable. Within their operating range, commercially available thermistors will match a resistance-temperature curve within approximately 0.1°F (0.056°C). Some manufacturers of thermistors can supply them with a stability of 0.05°F (0.028°C) per year. For direct digital control (DDC) systems, the same sensor is used for both temperature indication, or monitoring, and temperature control. In DDC systems, RTDs with positive temperature coefficient are widely used.

Amplifier(s). The measured electric signal from the temperature sensor is amplified at the solid state amplifier to produce an output for indication. The number of amplifiers is matched with the number of the sensors used in the temperature indication system.

Indicator. An analog-type indicator, one based on directly measurable quantities, is usually a moving coil instrument. For a digital-type indicator, the signal from the amplifier is compared with an internal reference voltage and converted for indication through an analog-digital transducer.

2.4 HUMIDITY

Humidity Ratio

The *humidity ratio* of moist air w is the ratio of the mass of water vapor m_w to the mass of dry air m_a contained in the mixture of the moist air, in lb/lb (kg/kg). The humidity ratio can be calculated as

$$w = \frac{m_w}{m_a} \tag{2.14}$$

Since dry air and water vapor can occupy the same volume at the same temperature, we can apply the ideal gas equation and Dalton's law for dry air and water vapor. Equation (2.14) can be rewritten as

$$\begin{aligned} w = \frac{m_w}{m_a} &= \frac{p_w V R_a T_R}{P_a V R_w T_R} = \frac{R_a}{R_w}\frac{p_w}{p_{at} - p_w} \\ &= \frac{53.352}{85.778}\frac{p_w}{p_{at} - p_w} = 0.62198\frac{p_w}{p_{at} - p_w} \end{aligned} \tag{2.15}$$

where R_a, R_w = gas constant for dry air and water vapor, respectively, ft·lb$_f$/lb$_m$·°R(J/kg·K). Equation (2.15) is expressed in the form of the ratio of pressures; therefore, p_w and p_{at} must have the same units, either psia or psf (Pa).

For moist air at saturation, Eq. (2.15) becomes

$$w_s = 0.62198\frac{p_{ws}}{p_{at} - p_{ws}} \tag{2.16}$$

where p_{ws} = pressure of water vapor of moist air at saturation, psia or psf (Pa).

Relative Humidity

The *relative humidity* φ of moist air, or RH, is defined as the ratio of the mole fraction of water vapor x_w in a moist air sample to the mole fraction of the water vapor in a saturated moist air sample x_{ws} at the same temperature and pressure. This relationship can be expressed as

$$\varphi = \frac{x_w}{x_{ws}}\Big|_{T,p} \tag{2.17}$$

And, by definition, the following expressions may be written:

$$x_w = \frac{n_w}{n_a + n_w} \tag{2.18}$$

$$x_{ws} = \frac{n_{ws}}{n_a + n_{ws}} \tag{2.19}$$

where n_a = number of moles of dry air, mol
n_w = number of moles of water vapor in moist air sample, mol
n_{ws} = number of moles of water vapor in saturated moist air sample, mol

Moist air is a binary mixture of dry air and water vapor; therefore, we find that the sum of the mole fractions of dry air x_a and water vapor x_w is equal to 1, that is,

$$x_a + x_w = 1 \tag{2.20}$$

If we apply ideal gas equations $p_w V = n_w R_o T_R$ and $p_a V = n_a R_o T_R$, by substituting them into Eq. (2.19), then the relative humidity can also be expressed as

$$\varphi = \frac{p_w}{p_{ws}}\Big|_{T,p} \tag{2.21}$$

The water vapor pressure of saturated moist air p_{ws} is a function of temperature T and pressure p, which is slightly different from the saturation pressure of water vapor p_s. Here p_s is a function of temperature T only. Since the difference between p_{ws} and p_s is small, it is usually ignored.

Degree of Saturation

The *degree of saturation* μ is defined as the ratio of the humidity ratio of moist air w to the humidity ratio of the saturated moist air w_s at the same temperature and pressure. This relationship can be expressed as

$$\mu = \frac{w}{w_s}\Big|_{T,p} \tag{2.22}$$

Since from Eqs. (2.15), (2.20), and (2.21) $w = 0.62198\, x_w/x_a$ and $w_s = 0.62198\, x_{ws}/x_a$, Eqs. (2.20), (2.21), and (2.22) can be combined, so that

$$\varphi = \frac{\mu}{1 - (1 - \mu)x_{ws}} = \frac{\mu}{1 - (1 - \mu)(p_{ws}/p_{at})} \tag{2.23}$$

In Eq. (2.23), $p_{ws} \ll p_{at}$; therefore, the difference between φ and μ is small. Usually, the maximum difference is less than 2 percent.

2.5 PROPERTIES OF MOIST AIR

Enthalpy

The difference in specific enthalpy Δh for an ideal gas, in Btu/lb (kJ/kg), at a constant pressure can be defined as

$$\Delta h = c_p (T_2 - T_1) \tag{2.24}$$

where c_p = specific heat at constant pressure, Btu/lb·°F (kJ/kg·K)
T_1, T_2 = temperature of ideal gas at points 1 and 2, °F (°C)

As moist air is approximately a binary mixture of dry air and water vapor, the enthalpy of the moist air can be evaluated as

$$h = h_a + H_w \tag{2.25}$$

where h_a and H_w are, respectively, enthalpy of dry air and total enthalpy of water vapor, in Btu/lb (kJ/kg). The following assumptions are made for the enthalpy calculations of moist air:

1. The ideal gas equation and the Gibbs-Dalton law are valid.
2. The enthalpy of dry air is equal to zero at 0°F (−17.8°C).
3. All water vapor contained in the moist air is vaporized at 0°F (−17.8°C).
4. The enthalpy of saturated water vapor at 0°F (−17.8°C) is 1061 Btu/lb (2468 kJ/kg).
5. For convenience in calculation, the enthalpy of moist air is taken to be equal to the enthalpy of a mixture of dry air and water vapor in which the amount of dry air is exactly equal to 1 lb (0.454 kg).

Based on the preceeding assumptions, the enthalpy h of moist air can be calculated as

$$h = h_a + wh_w \tag{2.26}$$

where h_w = specific enthalpy of water vapor, Btu/lb (kJ/kg). In a temperature range of 0 to 100°F (−17.8 to 37.8°C), the mean value for the specific heat of dry air can be taken as 0.240 Btu/lb · °F (1.005 kJ/kg · K). Then the specific enthalpy of dry air h_a is given by

$$h_a = c_{pd}\,T = 0.240\,T \tag{2.27}$$

where c_{pd} = specific heat of dry air at constant pressure, Btu/lb · °F (kJ/kg · K)
T = temperature of dry air, °F (°C)

The specific enthalpy of water vapor h_w at constant pressure can be approximated as

$$h_w = h_{g0} + c_{ps}T \tag{2.28}$$

where h_{g0} = specific enthalpy of saturated water vapor at 0°F (−17.8°C)—its value can be taken as 1061 Btu/lb (2468 kJ/kg)
c_{ps} = specific heat of water vapor at constant pressure, Btu/lb · °F (kJ/kg · K)

In a temperature range of 0 to 100°F (−17.8 to 37.8°C), its value can be taken as 0.444 Btu/lb · °F (1.859 kJ/kg · K). Then the enthalpy of moist air can be evaluated as

$$h = c_{pd}T + w(h_{g0} + c_{ps}T) = 0.240\,T + w(1061 + 0.444\,T) \tag{2.29}$$

Here, the unit of h is Btu/lb of dry air (kJ/kg of dry air). For simplicity, it is often expressed as Btu/lb (kJ/kg).

Moist Volume

The *moist volume* of moist air v, ft^3/lb (m^3/kg), is defined as the volume of the mixture of the dry air and water vapor when the mass of the dry air is exactly equal to 1 lb (1 kg), that is,

$$v = \frac{V}{m_a} \tag{2.30}$$

where V = total volume of mixture, ft^3 (m^3)
m_a = mass of dry air, lb (kg)

In a moist air sample, the dry air, water vapor, and moist air occupy the same volume. If we apply the ideal gas equation, then

$$v = \frac{V}{m_a} = \frac{R_a T_R}{p_{\text{at}} - p_w} \tag{2.31}$$

where p_{at} and p_w are both in psf (Pa). From Eq. (2.15), $p_w = p_{at}w/(w + 0.62198)$. Substituting this expression into Eq. (2.31) gives

$$v = \frac{R_a T_R(1 + 1.6078\, w)}{P_{at}} \tag{2.32}$$

According to Eq. (2.32), the volume of 1 lb (1 kg) of dry air is always smaller than the volume of the moist air when both are at the same temperature and the same atmospheric pressure.

Density

Since the enthalpy and humidity ratio are always related to a unit mass of dry air, for the sake of consistency, *air density* ρ_a, in lb/ft^3 (kg/m^3), should be defined as the ratio of the mass of dry air to the total volume of the mixture, i.e., the reciprocal of moist volume, or

$$\rho_a = \frac{m_a}{V} = \frac{1}{v} \tag{2.33}$$

Sensible Heat and Latent Heat

Sensible heat is that heat energy associated with the change of air temperature between two state points. In Eq. (2.29), the enthalpy of moist air calculated at a datum state 0°F (−17.8°C) can be divided into two parts:

$$h = (c_{pd} + wc_{ps})T + wh_{g0} \tag{2.34}$$

The first term on the right-hand side of Eq. (2.34) indicates the sensible heat. It depends on the temperature T above the datum 0°F (−17.8°C).

Latent heat h_{fg} (sometimes called h_{ig}) is the heat energy associated with the change of the state of water vapor. The latent heat of vaporization denotes the latent heat required to vaporize liquid water into water vapor. Also, the latent heat of condensation indicates the latent heat to be removed in the condensation of water vapor into liquid water. When moisture is added to or removed from a process or a space, a corresponding amount of latent heat is always involved, to vaporize the water or to condense it.

In Eq. (2.34), the second term on the right-hand side, wh_{g0}, denotes latent heat. Both sensible and latent heat are expressed in Btu/lb (kJ/kg) of dry air.

Specific Heat of Moist Air at Constant Pressure

The specific heat of moist air at constant pressure c_{pa} is defined as the heat required to raise its temperature 1°F (0.56°C) at constant pressure. In (inch-pound) I-P units, it is expressed as Btu/lb·°F (in SI units, as J/kg·K). In Eq. (2.34), the sensible heat of moist air q_{sen}, Btu/h (W), is represented by

$$q_{sen} = \dot{m}_a(c_{pd} + wc_{ps})T = \dot{m}_a c_{pa} T \tag{2.35}$$

where $\dot{m}_a$ = mass flow rate of moist air, lb/h (kg/s). Apparently

$$c_{pa} = c_{pd} + wc_{ps} \tag{2.36}$$

Since c_{pd} and c_{ps} are both a function of temperature, c_{pa} is also a function of temperature and, in addition, a function of the humidity ratio.

For a temperature range of 0 to 100°F (−17.8 to 37.8°C), c_{pd} can be taken as 0.240 Btu/lb·°F (1005 J/kg·K) and c_{ps} as 0.444 Btu/lb·°F (1859 J/kg·K). Most of the calculations of $c_{pa}(T_2 - T_1)$

have a range of w between 0.005 and 0.010 lb/lb (kg/kg). Taking a mean value of $w = 0.0075$ lb/lb (kg/kg), we find that

$$c_{pa} = 0.240 + 0.0075 \times 0.444 = 0.243 \text{ Btu/lb} \cdot {}^\circ\text{F } (1020 \text{ J/kg} \cdot \text{K})$$

Dew-Point Temperature

The *dew-point temperature* T_{dew} is the temperature of saturated moist air of the same moist air sample, having the same humidity ratio, and at the same atmospheric pressure of the mixture p_{at}. Two moist air samples at the same T_{dew} will have the same humidity ratio w and the same partial pressure of water vapor p_w. The dew-point temperature is related to the humidity ratio by

$$w_s(p_{\text{at}}, T_{\text{dew}}) = w \tag{2.37}$$

where w_s = humidity ratio of saturated moist air, lb/lb (kg/kg). At a specific atmospheric pressure, the dew-point temperature determines the humidity ratio w and the water vapor pressure p_w of the moist air.

2.6 THERMODYNAMIC WET-BULB TEMPERATURE AND THE WET-BULB TEMPERATURE

Ideal Adiabatic Saturation Process

If moist air at an initial temperature T_1, humidity ratio w_1, enthalpy h_1, and pressure p flows over a water surface of infinite length in a well-insulated chamber, as shown in Fig. 2.3, liquid water will evaporate into water vapor and will disperse in the air. The humidity ratio of the moist air will gradually increase until the air can absorb no more moisture.

As there is no heat transfer between this insulated chamber and the surroundings, the latent heat required for the evaporation of water will come from the sensible heat released by the moist air. This process results in a drop in temperature of the moist air. At the end of this evaporation process, the moist air is always saturated. Such a process is called an *ideal adiabatic saturation process*, where an adiabatic process is defined as a process without heat transfer to or from the process.

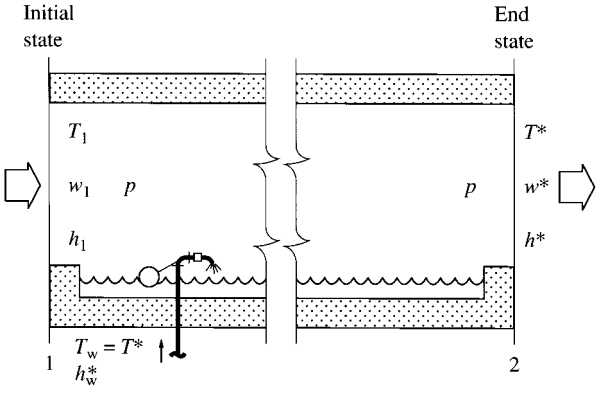


FIGURE 2.3 Ideal adiabatic saturation process.

Thermodynamic Wet-Bulb Temperature

For any state of moist air, there exists a thermodynamic wet-bulb temperature T^* that exactly equals the saturated temperature of the moist air at the end of the ideal adiabatic saturation process at constant pressure. Applying a steady flow energy equation, we have

$$h_1 + (w_s^* - w_1)h_w^* = h_s^* \tag{2.38}$$

where h_1, h_s^* = enthalpy of moist air at initial state and enthalpy of saturated air at end of ideal adiabatic saturation process, Btu/lb (kJ/kg)
w_1, w_s^* = humidity ratio of moist air at initial state and humidity ratio of saturated air at end of ideal adiabatic saturation process, lb/lb (kg/kg)
h_w^* = enthalpy of water as it is added to chamber at a temperature T^*, Btu/lb (kJ/kg)

The thermodynamic wet-bulb temperature T^*, °F (°C), is a unique property of a given moist air sample that depends only on the initial properties of the moist air—w_1, h_1 and p. It is also a fictitious property that only hypothetically exists at the end of an ideal adiabatic saturation process.

Heat Balance of an Ideal Adiabatic Saturation Process

When water is supplied to the insulation chamber at a temperature T^* in an ideal adiabatic saturation process, then the decrease in sensible heat due to the drop in temperature of the moist air is just equal to the latent heat required for the evaporation of water added to the moist air. This relationship is given by

$$c_{pd}(T_1 - T^*) + c_{ps}w_1(T_1 - T^*) = (w_s^* - w_1)h_{fg}^* \tag{2.39}$$

where T_1 = temperature of moist air at initial state of ideal adiabatic saturation process, °F (°C)
h_{fg}^* = latent heat of vaporization at thermodynamic wet-bulb temperature, Btu/lb (J/kg)

Since $c_{pa} = c_{pd} + w_1 c_{ps}$, we find, by rearranging the terms in Eq. (2.39),

$$\frac{w_s^* - w_1}{T_1 - T^*} = \frac{c_{pa}}{h_{fg}^*} \tag{2.40}$$

Also

$$T^* = T_1 - \frac{(w_s^* - w_1)h_{fg}^*}{c_{pa}} \tag{2.41}$$

Psychrometer

A psychrometer is an instrument that permits one to determine the relative humidity of a moist air sample by measuring its dry-bulb and wet-bulb temperatures. Figure 2.4 shows a psychrometer, which consists of two thermometers. The sensing bulb of one of the thermometers is always kept dry. The temperature reading of the dry bulb is called the *dry-bulb temperature*. The sensing bulb of the other thermometer is wrapped with a piece of cotton wick, one end of which dips into a cup of distilled water. The surface of this bulb is always wet; therefore, the temperature that this bulb measures is called the *wet-bulb temperature*. The dry bulb is separated from the wet bulb by a radiation-shielding plate. Both dry and wet bulbs are cylindrical.

Wet-Bulb Temperature

When unsaturated moist air flows over the wet bulb of the psychrometer, liquid water on the surface of the cotton wick evaporates, and as a result, the temperature of the cotton wick and the wet bulb

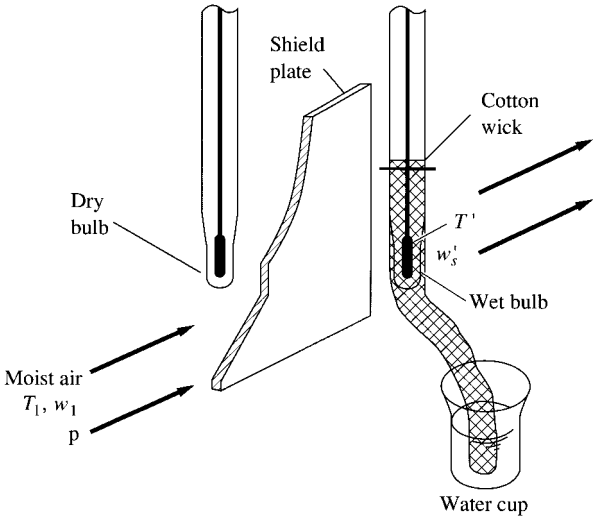


FIGURE 2.4 A psychrometer.

drops. This depressed wet-bulb reading is called the wet-bulb temperature T', and the difference between the dry-bulb and wet-bulb temperatures is called the *wet-bulb depression.*

Let us neglect the conduction along the thermometer stems to the dry and wet bulbs and also assume that the temperature of the water on the cotton wick is equal to the wet-bulb temperature of the moist air. Since the heat transfer from the moist air to the cotton wick is exactly equal to the latent heat required for vaporization, then, at steady state, the heat and mass transfer per unit area of the wet-bulb surface can be calculated as

$$h_c(T - T') + h_r(T_{ra} - T') = h_d(w_s' - w_1)h_{fg}' \tag{2.42}$$

where h_c, h_r = mean convective and radiative heat transfer coefficients, Btu/h·ft²·°F (W/m²·K)
h_d = mean convective mass-transfer coefficient, lb/h·ft² (kg/s·m²)
T = temperature of undisturbed moist air at a distance from wet bulb, °F (°C)
T' = wet-bulb temperature, °F (°C)
T_{ra} = mean radiant temperature, °F (°C)
w_1, w_s' = humidity ratio of moist air and saturated air film at surface of cotton wick, lb/lb (kg/kg)
h_{fg}' = latent heat of vaporization at wet-bulb temperature, Btu/lb (J/kg)

Based on the correlation of cross-flow forced convective heat transfer for a cylinder, $\mathrm{Nu_D} = C\,\mathrm{Re}^n\,\mathrm{Pr}^{0.333}$, and based on the analogy between convective heat transfer and convective mass transfer, the following relationship holds: $h_d = h_c/(c_{pa}\mathrm{Le}^{0.6667})$. Here, Nu is the Nusselt number, Re the Reynolds number, and Le the Lewis number. Also C is a constant.

Substituting this relationship into Eq. (2.42), we have

$$w_1 = w_s' - K'(T - T') \tag{2.43}$$

In Eq. (2.43), K' represents the wet-bulb constant. It can be calculated as

$$K' = \frac{c_{pa}\,\mathrm{Le}^{0.6667}}{h_{fg}'}\left[1 + \frac{h_r(T_{ra} - T')}{h_c(T - T')}\right] \tag{2.44}$$

The term $T - T'$ indicates the wet-bulb depression.

Combining Eqs. (2.43) and (2.44) then gives

$$\frac{w_s' - w_1}{T - T'} = \frac{c_{pa}\text{Le}^{0.667}}{h_{fg}'}\left[1 + \frac{h_r(T_{ra} - T')}{h_c(T - T')}\right] \tag{2.45}$$

Relationship between Wet-Bulb Temperature and Thermodynamic Wet-Bulb Temperature

Wet-bulb temperature is a function not only of the initial state of moist air, but also of the rate of heat and mass transfer at the wet bulb. Comparing Eq. (2.40) with Eq. (2.45), we find that the wet-bulb temperature measured by using a psychrometer is equal to thermodynamic wet-bulb temperature only when the following relationship holds:

$$\text{Le}^{0.6667}\left[1 + \frac{h_r(T_{\text{ra}} - T')}{h_c(T - T')}\right] = 1 \tag{2.46}$$

2.7 SLING AND ASPIRATION PSYCHROMETERS

Sling and aspiration psychrometers determine the relative humidity through the measuring of the dry- and wet-bulb temperatures.

A sling psychrometer with two bulbs, one dry and the other wet, is shown in Fig. 2.5*a*. Both dry and wet bulbs can be rotated around a spindle to produce an airflow over the surfaces of the dry and wet bulbs at an air velocity of 400 to 600 fpm (2 to 3 m/s). Also a shield plate separates the dry and wet bulbs and partly protects the wet bulb against surrounding radiation.

An aspiration psychrometer that uses a small motor-driven fan to produce an air current flowing over the dry and wet bulbs is illustrated in Fig. 2.5*b*. The air velocity over the bulbs is usually kept at 400 to 800 fpm (2 to 4 m/s). The dry and wet bulbs are located in separate compartments and are entirely shielded from the surrounding radiation. When the space dry-bulb temperature is within a range of 75 to 80°F (24 to 27°C) and the space wet-bulb temperature is between 65 and 70°F (18 and 21°C), the following wet-bulb constants K' can be used to calculate the humidity ratio of the moist air:

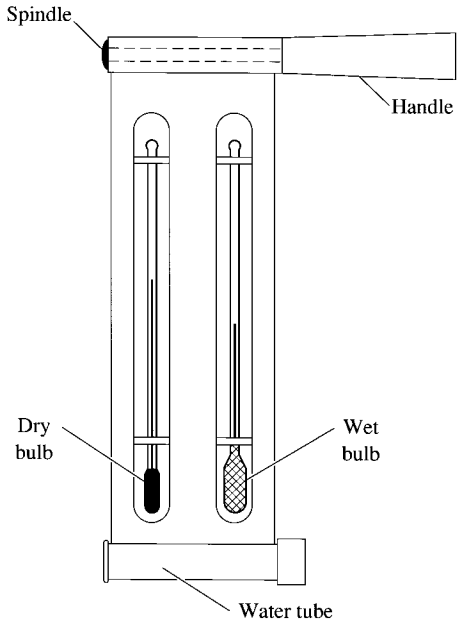
Aspiration psychrometer $K' = 0.000206$ 1/°F

Sling psychrometer $K' = 0.000218$ 1/°F

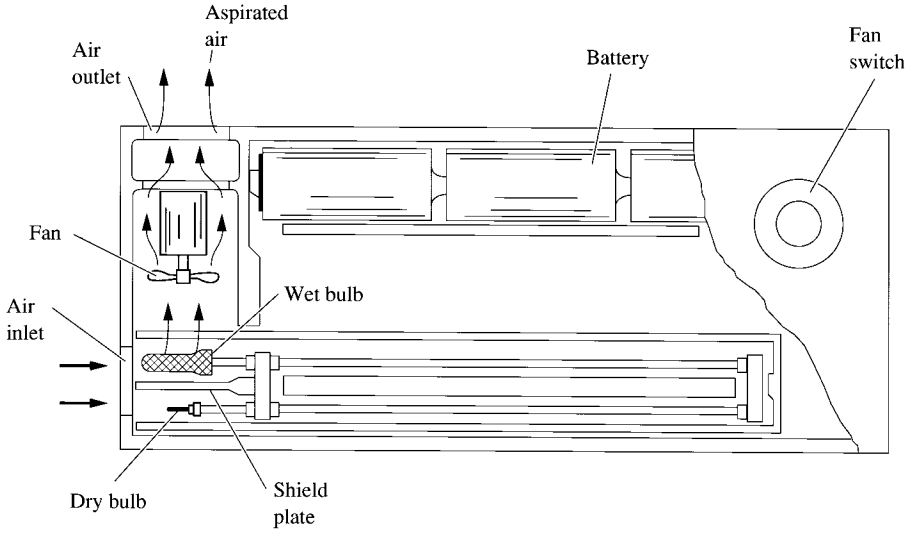
After the psychrometer has measured the dry- and wet-bulb temperatures of the moist air, the humidity ratio w can be calculated by Eq. (2.43). Since the saturated water vapor pressure can be found from the psychrometric table, the relative humidity of moist air can be evaluated through Eq. (2.15).

According to the analysis of Threlkeld (1970), for a wet-bulb diameter of 0.1 in. (2.5 mm) and an air velocity flowing over the wet bulb of 400 fpm (2 m/s), if the dry-bulb temperature is 90°F (32.2°C) and the wet-bulb temperature is 70°F (21.1°C), then $(T' - T^*)/(T - T')$ is about 2.5 percent. Under the same conditions, if a sling psychrometer is used, then the deviation may be reduced to about 1 percent. If the air velocity flowing over the wet bulb exceeds 400 fpm (2 m/s), there is no significant reduction in the deviation.

Distilled water must be used to wet the cotton wick for both sling and aspiration psychrometers. Because dusts contaminate them, cotton wicks should be replaced regularly to provide a clean surface for evaporation.



(a)



(b)

FIGURE 2.5 Sling and aspiration psychrometers (a) Sling psychrometer; (b) aspiration psychrometer.

2.8 HUMIDITY MEASUREMENTS

Humidity sensors used in HVAC&R for direct humidity indication or operating controls are separated into the following categories: mechanical hygrometers and electronic hygrometers.

Mechanical Hygrometers

Mechanical hygrometers operate on the principle that hygroscopic materials expand when they absorb water vapor or moisture from the ambient air. They contract when they release moisture to the surrounding air. Such hygroscopic materials include human and animal hairs, plastic polymers like nylon ribbon, natural fibers, wood, etc. When these materials are linked to mechanical linkages or electric transducers that sense the change in size and convert it into electric signals, the results in these devices can be calibrated to yield direct relative-humidity measurements of the ambient air.

Electronic Hygrometers

There are three types of electronic hygrometers: Dunmore resistance hygrometers, ion-exchange resistance hygrometers, and capacitance hygrometers.

Dunmore Resistance Hygrometer. In 1938, Dunmore of the National Bureau of Standards developed the first lithium chloride resistance electric hygrometer in the United States. This instrument depends on the change in resistance between two electrodes mounted on a hygroscopic material. Figure. 2.6*a* shows a Dunmore resistance sensor. The electrodes could be, e.g., a double-threaded winding of noble-metal wire mounted on a plastic cylinder coated with hygroscopic material. The wires can also be in a grid-type arrangement with a thin film of hygroscopic material bridging the gap between the electrodes.

At a specific temperature, electric resistance decreases with increasing humidity. Because the response is significantly influenced by temperature, the results are often indicated by a series of isothermal curves. Relative humidity is generally used as the humidity parameter, for it must be controlled in the indoor environment. Also the electrical response is more nearly a function of relative humidity than of the humidity ratio.

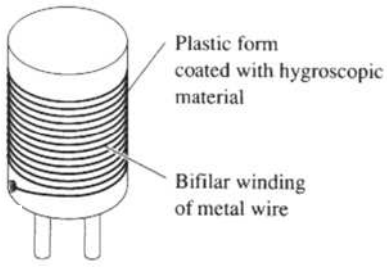
The time response to accomplish a 50 percent change in relative humidity varies directly according to the air velocity flowing over the sensor and also is inversely proportional to the saturated vapor pressure. If a sensor has a response time of 10 s at 70°F (21°C), it might need a response time of 100 s at 10°F (−12°C).

Because of the steep variation of resistance corresponding to a change in relative humidity, each of the Dunmore sensors only covers a certain range of relative-humidity measurements. A set of several Dunmore sensors is usually needed to measure relative humidity between 1 percent and 100 percent.

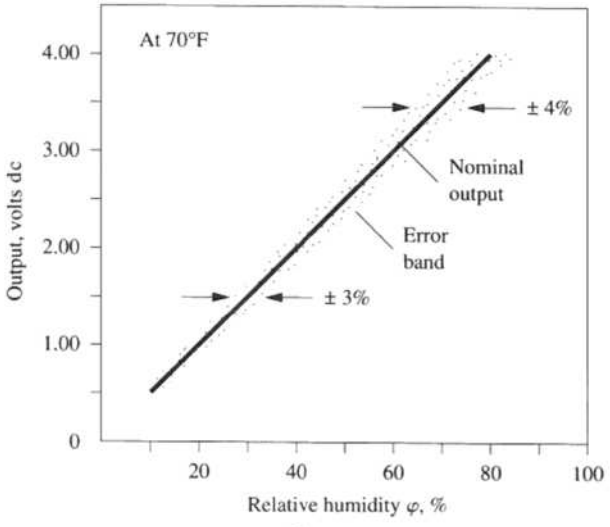
A curve for output, in direct-current (dc) volts, versus relative humidity is shown in Fig. 2.6*b* for a typical Dunmore sensor. It covers a measuring range of 10 to 80 percent, which is usually sufficient for the indication of relative humidity for a comfort air conditioned space. This typical Dunmore sensor has an accuracy of ±3 percent when the relative humidity varies between 10 and 60 percent at a temperature of 70°F (21°C, see Fig. 2.6*b*). Its accuracy reduces to ±4 percent when the relative humidity is in a range between 60 and 80 percent at the same temperature.

In addition to lithium chloride, lithium bromide is sometimes used as the sensor.

Ion-Exchange Resistance Hygrometer. The sensor of a ion-exchange resistance electric hygrometer is composed of electrodes mounted on a baseplate and a high-polymer resin film, used as a humidity-sensing material, cross-linking the electrodes as shown in Fig. 2.7*a* and *b*. Humidity is



(a)



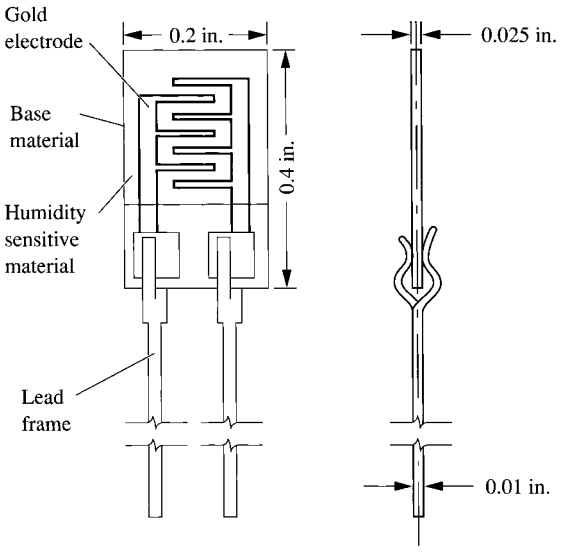
(b)

FIGURE 2.6 Dunmore resistance-type electric hygrometer. (a) Sensor; (b) output versus relative humidity φ. *(Reprinted by permission of Johnson Controls.)*

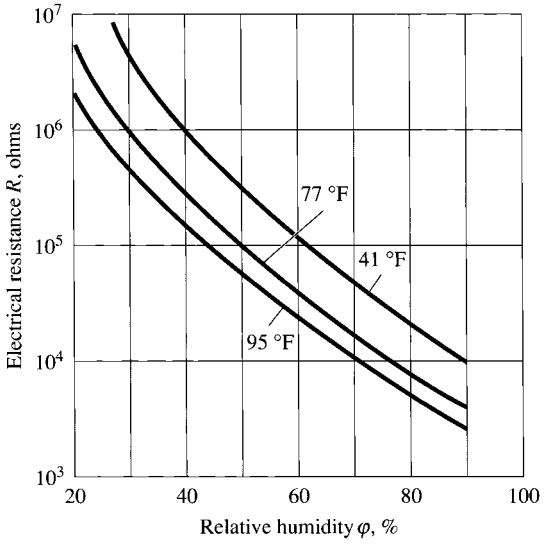
measured by the change in resistance between the electrodes. When the salt contained in the humidity-sensitive material bridging the electrodes becomes ion-conductive because of the presence of water vapor in the ambient air, mobile ions in the polymer film are formed. The higher the relative humidity of the ambient air, the greater the ionization becomes, and therefore, the greater the concentration of mobile ions. On the other hand, lower relative humidity reduces the ionization and results in a lower concentration of mobile ions. The resistance of the humidity-sensing material reflects the change of the relative humidity of the ambient air.

In Fig. 2.7*b*, the characteristic curves of an ion-exchange resistance electric hygrometer show that there is a nonlinear relationship between resistance R and relative humidity φ. These sensors cover a wider range than Dunmore sensors, from 20 to 90 percent relative humidity.

Capacitance Hygrometer. The commonly used capacitance sensor consists of a thin-film plastic foil. A very thin gold coating covers both sides of the film as electrodes, and the film is mounted



(a)



(b)

FIGURE 2.7 Ion-exchange resistance-type electric hygrometer. (a) Front and side view of sensor; (b) characteristic curve of R versus φ. (Reprinted by permission of General Eastern Instruments.)

inside a capsule. The golden electrodes and the dividing plastic foil form a capacitor. Water vapor penetrates the gold layer, which is affected by the vapor pressure of the ambient air and, therefore, the ambient relative humidity. The number of water molecules absorbed on the plastic foil determines the capacitance and the resistance between the electrodes.

Comparison of Various Methods

The following table summarizes the sensor characteristics of various methods to be used within a temperature range of 32 to 120°F (0 to 50°C) and a range of 10 to 95 percent RH:

	Operating method	Accuracy, % RH
Psychrometer	Wet-bulb depression	±3
Mechanical	Dimensional change	±3 to ±5
Dunmore	Electric resistance	±1.5
Ion-exchange	Electric resistance	±2 to ±5
Capacitance	Electric capacitance and resistance	±3 to ±5

Psychrometers are simple and comparatively low in cost. They suffer no irreversible damage at 100 percent RH, as do the sensors of electric hygrometers. Unfortunately, complete wet-bulb depression readings of psychrometers become difficult when relative humidity drops below 20 percent or when the temperature is below the freezing point. For remote monitoring, it is difficult to keep sufficient water in the water reservoir. Therefore, psychrometers are sometimes used to check the temperature and relative humidity in the air conditioned space manually.

Mechanical hygrometers directly indicate the relative humidity of the moist air. They are also simple and relatively inexpensive. Their main drawbacks are their lack of precision over an extensive period and their lack of accuracy at extreme high and low relative humidities. Electronic hygrometers, especially the polymer film resistance and the capacitance types, are commonly used for remote monitoring and for controls in many air conditioning systems. Both the electronic and mechanical hygrometers need regular calibration. Initial calibrations are usually performed either with precision humidity generators using two-pressure, two-temperature, and divided-flow systems or with secondary standards during manufacturing (refer to ASHRAE Standard 41.6-1982, *Standard Method for Measurement of Moist Air Properties*). Regular calibrations can be done with a precision aspiration psychrometer or with chilled mirror dew-point devices.

Air contamination has significant influence on the performance of the sensor of electronic and mechanical hygrometers. This is one of the reasons why they need regular calibration.

2.9 PSYCHROMETRIC CHARTS

Psychrometric charts provide a graphical representation of the thermodynamic properties of moist air, various air conditioning processes, and air conditioning cycles. The charts are very helpful during the calculation, analysis, and solution of the complicated problems encountered in air conditioning processes and cycles.

Basic Coordinates. The currently used psychrometric charts have two types of coordinates:

- *h-w* chart. Enthalpy *h* and humidity ratio *w* are basic coordinates. The psychrometric charts published by ASHRAE and the Chartered Institution of Building Services Engineering (CIBSE) are *h-w* charts.
- *T-w* chart. Temperature *T* and humidity ratio *w* are basic coordinates. Most of the psychrometric charts published by the large manufacturers in the United States are *T-w* charts.

For an atmospheric pressure of 29.92 in. Hg (760 mm Hg), an air temperature of 84°F (28.9°C), and a relative humidity of 100 percent, the humidity ratios and enthalpies found from the psychrometric charts published by ASHRAE and Carrier International Corporation are shown below:

	Humidity ratio, lb/lb	Enthalpy, Btu/lb
ASHRAE's chart	0.02560	48.23
Carrier's chart	0.02545	48.20

The last digit for humidity ratios and for enthalpies read from ASHRAE's chart is an approximation. Nevertheless, the differences between the two charts are less than 1 percent, and these are considered negligible.

In this handbook, for manual psychrometric calculations and analyses, ASHRAE's chart will be used.

Temperature Range and Barometric Pressure. ASHRAE's psychrometric charts are constructed for various temperature ranges and altitudes. In Appendix B only the one for normal temperature, that is, 32 to 120°F (0 to 50°C), and a standard barometric pressure at sea level, 29.92 in. Hg (760 mm Hg), is shown. The skeleton of ASHRAE's chart is shown in Fig. 2.8.

Enthalpy Lines. For ASHRAE's chart, the molar enthalpy of moist air is calculated from the formulation recommended by Hyland and Wexler (1983) in their paper "Formulations for the Thermodynamic Properties of Dry Air from 173.15 K to 473.15 K, and of Saturated Moist Air from 173.15 K to 473.15 K, at Pressures to 5 MPa." For ASHRAE's chart, the enthalpy h lines incline at an angle of 25° to the horizontal lines. The scale factor for the enthalpy lines C_h, Btu/lb · ft (kJ/kg · m), is

$$C_h = \frac{h_2 - h_1}{L_h} = \frac{20}{0.4128} = 48.45 \text{ Btu/lb} \cdot \text{ft } (370 \text{ kJ/kg} \cdot \text{m}) \tag{2.47}$$

where L_h = shortest distance between enthalpy lines h_2 and h_1, ft (m).

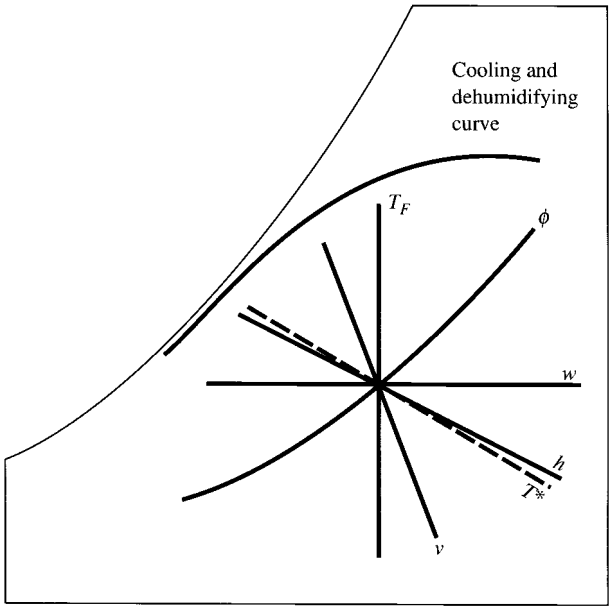


FIGURE 2.8 Skeleton of ASHRAE's psychrometric chart.

Humidity Ratio Lines. In ASHRAE's chart, the humidity ratio w lines are horizontal. They form the ordinate of the psychrometric chart. The scale factor C_w, lb/lb·ft (kg/kg·m), for w lines in ASHRAE's chart is

$$C_w = \frac{w_2 - w_1}{L_w} = \frac{0.020}{0.5} = 0.040 \text{ lb/lb·ft } (0.131 \text{ kg/kg·m}) \tag{2.48}$$

where L_w = vertical distance between w_2 and w_1, ft (m). For ASHRAE's chart, the humidity ratio w can be calculated by Eq. (2.15).

Constant-Temperature Lines. For ASHRAE's chart, since enthalpy is one of the coordinates, only the 120°F constant-temperature T line is a true vertical. All the other constant-temperature lines incline slightly to the left at the top. From Eq. (2.27), $T = h_a/c_{pd}$; therefore, one end of the T line in ASHRAE'S chart can be determined from the enthalpy scale at $w = 0$. The other end can be determined by locating the saturated humidity ratio w_s on the saturation curve.

Saturation Curve. A saturation curve is a locus representing a series of state points of saturated moist air. For ASHRAE's chart, the enthalpy of saturation vapor over liquid water or over ice at a certain temperature is calculated by the formula recommended in the Hyland and Wexler paper. The humidity ratios of the saturated moist air w_s between 0 and 100°F (−17.8 and 37.8°C) in the psychrometric chart can also be calculated by the following simpler polynomial:

$$w_s = a_1 + a_2 T_s + a_3 T_s^2 + a_4 T_s^3 + a_5 T_s^4 \tag{2.49}$$

where T_s = saturated temperature of moist air, °F (°C)
$a_1 = 0.00080264$
$a_2 = 0.000024525$
$a_3 = 2.5420 \times 10^{-6}$
$a_4 = -2.5855 \times 10^{-8}$
$a_5 = 4.038 \times 10^{-10}$

If we use Eq. (2.49) to calculate w_s, the error is most probably less than 0.000043 lb/lb (kg/kg). It is far smaller than the value that can be identified on the psychrometric chart, and therefore the calculated w_s is acceptable.

Relative-Humidity and Moist Volume Lines. For ASHRAE's chart, relative-humidity φ lines, thermodynamic wet-bulb T^* lines, and moist volume v lines all are calculated and determined based on the formulations in Hyland and Wexler's paper.

Thermodynamic Wet-Bulb Lines. For ASHRAE's chart, only thermodynamic wet-bulb T^* lines are shown. Since the dry-bulb and the thermodynamic wet-bulb temperatures coincide with each other on the saturation curve, one end of the T^* line is determined. The other end of the T^* line can be plotted on the w line where $w = 0$. Let the state point of the other end of the T^* line be represented by 1. Then, from Eq. (2.39), at $w_1 = 0$

$$c_{pd}(T_1 - T^*) = w_s^* h_{fg}^*$$

Solving for T^*, we have

$$T^* = \frac{c_{pd} T_1 - w_s^* h_{fg}^*}{c_{pd}} \tag{2.50}$$

Cooling and Dehumidifying Curves. The two cooling and dehumidifying curves plotted on ASHRAE's chart are based on data on coil performance published in the catalogs of U.S. manufacturers. These curves are very helpful in describing the actual locus of a cooling and dehumidifying process as well as determining the state points of air leaving the cooling coil.

2.10 DETERMINATION OF THERMODYNAMIC PROPERTIES ON PSYCHROMETRIC CHARTS

There are seven thermodynamic properties or property groups of moist air shown on a psychrometric chart:

1. Enthalpy h
2. Relative humidity φ
3. Thermodynamic wet-bulb temperature T^*
4. Barometric or atmospheric pressure p_{at}
5. Temperature T and saturation water vapor pressure p_{ws}
6. Density ρ and moist volume v
7. Humidity ratio w, water vapor pressure p_w, and dew-point temperature T_{dew}

The fifth, sixth, and seventh are thermodynamic property groups. These properties or properties groups are independent of each other except that the difference in slope between the enthalpy h line and thermodynamic wet-bulb temperature T^* line is small, and it is hard to determine their intersection. Usually, atmospheric pressure p_{at} is a known value based on the altitude of the location. Then, in the fifth property group, p_{ws} is a function of temperature T only. In the sixth property group, according to Eq. (2.33), $\rho_a = 1/v$; that is, air density and moist volume are dependent on each other. In the seventh property group, for a given value of p_{at}, properties w, p_w, and T_{dew} are all dependent on each other.

When p_{at} is a known value, and if the moist air is not saturated, then any two known independent thermodynamic properties can determine the magnitudes of the remaining unknown properties. If the moist air is saturated, then any independent property will determine the remaining magnitudes.

Example 2.1. The design indoor air temperature and relative humidity of an air conditioned space at sea level are 75°F (23.9°C) and 50 percent. Find the humidity ratio, the enthalpy, and the density of the indoor moist air

1. By using the ASHRAE chart
2. By calculation

Determine also the dew-point and thermodynamic wet-bulb temperatures of the moist air. The following information is required for the calculations:

Atmospheric pressure at sea level	14.697 psi (101,325 Pa)
Specific heat of dry air	0.240 Btu/lb · °F (1.005 kJ/kg · K)
Specific heat of water vapor	0.444 Btu/lb · °F (1.859 kJ/kg · K)
Enthalpy of saturated vapor at 0°F	1061 Btu/lb (2468 kJ/kg)
Gas constant of dry air	53.352 ft · lb_f/lb_m · °R (0.287 kJ/kg · K)

Solution

1. Plot the space point r on ASHRAE's chart by first finding the space temperature $T_r = 75$°F on

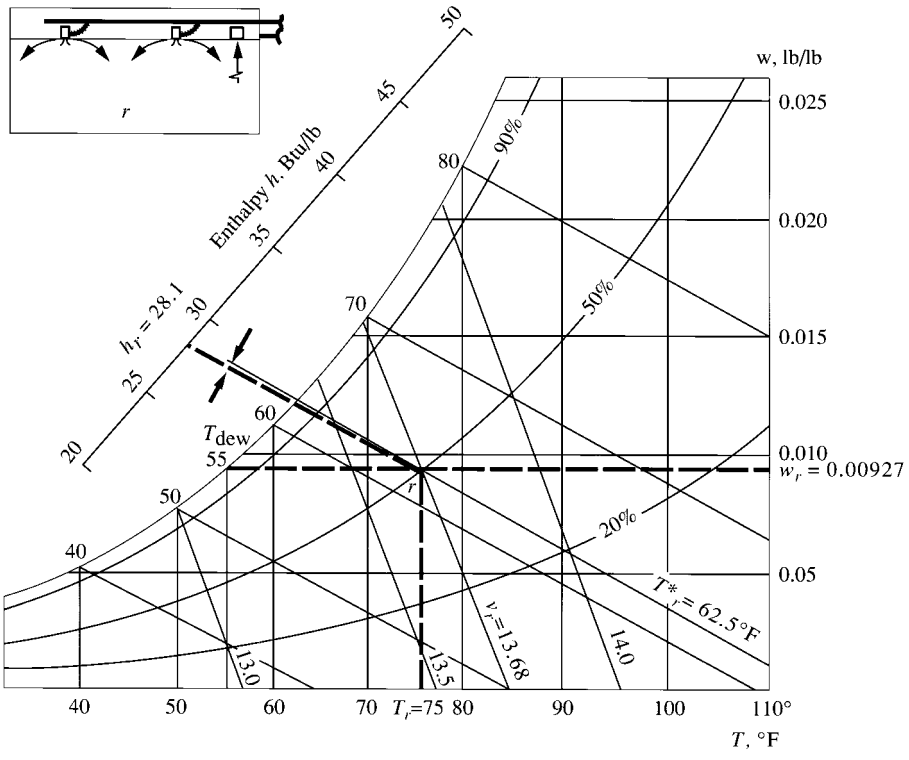


FIGURE 2.9 Thermodynamic properties determined from ASHRAE's psychrometric chart.

the abscissa and then following along the 75°F constant-temperature line up to a relative humidity $\varphi = 50$ percent, as shown in Fig. 2.9.

Draw a horizontal line from the space point *r*. This line meets the ordinate, humidity ratio *w*, at a value of $w_r = 0.00927$ lb/lb (kg/kg). This is the humidity ratio of the indoor space air. Draw a line parallel to the enthalpy line from the space point *r*. This line meets the enthalpy scale line at a value of $h_r = 28.1$ Btu/lb. This is the enthalpy of the indoor space air. Draw a horizontal line from the space point *r* to the left. This line meets the saturation curve at a dew-point temperature of 55°F (12.8°C). Draw a line parallel to the thermodynamic wet-bulb temperature lines through the space point *r*. The perpendicular scale to this line shows a thermodynamic wet-bulb temperature $T_r^* = 62.5$°F (16.9°C). Draw a line parallel to the moist volume lines through the space point *r*. The perpendicular scale to this line shows a moist volume $v_r = 13.68$ ft^3/lb (0.853 m^3/kg).

2. The calculations of the humidity ratio, enthalpy, and moist volume are as follows:

From Eq. (2.49), the humidity ratio of the saturated air at the dry-bulb temperature is

$$\begin{aligned} w_s &= 0.00080264 + 0.000024525T + 2.542\text{e-}06T^2 - 2.5855\text{e-}08T^3 + 4.038\text{e-}10T^4 \\ &= 0.00080264 + 0.0018394 + 0.014299 - 0.010908 + 0.012776 \\ &= 0.018809 \text{ lb/lb} \end{aligned}$$

According to Eq. (2.16), the saturated water vapor pressure of the indoor air is

$$p_{ws} = \frac{w_s p_{at}}{w_s + 0.62198} = \frac{0.018809 \times 14.697}{0.018809 + 0.62198} = 0.4314 \text{ psia}$$

From Eq. (2.21), the water vapor pressure of indoor air is

$$p_w = \varphi p_{ws} = 0.5 \times 0.4314 = 0.2157 \text{ psia}$$

Then, from Eq. (2.15), the humidity ratio of the indoor air is

$$w_r = \frac{0.62198 p_w}{p_{at} - p_w} = \frac{0.62198 \times 0.2157}{14.697 - 0.2157} = 0.009264 \text{ lb/lb } (0.009264 \text{ kg/kg})$$

From Eq. (2.29), the enthalpy of the indoor moist air is

$$h_r = c_{pd} T_F + w (h_g + c_{ps} T)$$
$$h_r = 0.240 \times 75 + 0.009264(1061 + 0.444 \times 75)$$
$$= 28.14 \text{ Btu/lb}$$

From Eq. (2.32), the moist volume of the indoor air is

$$v_r = \frac{R_a T}{p_{at} - p_w} = \frac{53.352 \times 535}{(14.697 - 0.2157)(144)} = 13.688 \text{ ft}^3\text{/lb}$$

Then, from Eq. (2.33), the density of the indoor moist air is

$$\rho_r = \frac{1}{v_r} = \frac{1}{13.688} = 0.07306 \text{ lb/ft}^3 \ (1.17 \text{ kg/m}^3)$$

Comparison of the thermodynamic properties read directly from ASHRAE's chart and the calculated values is as follows:

	ASHRAE's chart	Calculated value
Humidity ratio, lb/lb	0.00927	0.009264
Enthalpy, Btu/lb	28.1	28.14
Moist volume, ft^3/lb	13.68	13.69

Apparently, the differences between the readings from ASHRAE's chart and the calculated values are very small.

Example 2.2. An HVAC&R operator measured the dry- and wet-bulb temperatures in an air conditioned space as 75°F (23.9°C) and 63°F (17.2°C), respectively. Find the relative humidity of this air conditioned space by using ASHRAE's chart and by calculation. The humidity ratios of the saturated air at temperatures of 75 and 63°F are 0.018809 and 0.012355 lb/lb (kg/kg), respectively.

Solution. It is assumed that the difference between the wet-bulb temperature as measured by sling or aspiration psychrometer and the thermodynamic wet-bulb temperature is negligible. At a measured dry-bulb temperature of 75°F (23.9°C) and a wet-bulb temperature of 63°F (17.2°C), the relative humidity read directly from ASHRAE's chart is about 51.8 percent. From Sec. 2.6, the wet-bulb constant K' for a sling psychrometer is 0.000218 1/°F. Then, from Eq. (2.43), the humidity ratio of the space air can be calculated as

$$w = w_s' - 0.000218(T - T') = 0.012355 - 0.000218(75 - 63)$$
$$= 0.009739 \text{ lb/lb}$$

From Eq. (2.15), the vapor pressure of the space air is

$$p_w = \frac{wp_{at}}{w + 0.62198}$$
$$= \frac{0.009739 \times 14.697}{0.009739 + 0.62198} = 0.2266 \text{ psia}$$

And from Eq. (2.16), the saturated vapor pressure at a space temperature of 75°F is

$$p_{ws} = \frac{w_s p_{at}}{w_s + 0.62198}$$
$$= \frac{0.018809 \times 14.697}{0.018809 + 0.62198} = 0.4314 \text{ psia}$$

Hence, from Eq. (2.21), the calculated relative humidity of the space air is

$$\varphi = \frac{p_w}{p_{ws}} = \frac{0.2266}{0.4314} = 0.5253 \text{ or } 52.53\%$$

The difference between the value read directly from ASHRAE's chart and the calculated one is 52.53 percent − 51.8 percent = 0.7 percent.

Computer-Aided Psychrometric Calculation and Analysis

There are two kinds of psychrometric computer-aided software available on the market: psychrometric calculations and psychrometric graphics. Most of the psychrometric software is Windows-based computer programs.

Software for psychrometric calculations can determine any one of the thermodynamic properties of the moist air if two of the independent properties are known. Psychrometric calculation software usually also finds the thermodynamic property of the mixture of airstreams and provides altitude effect adjustments. Software programs for psychrometric graphics are far more powerful tools than psychrometric calculations. On the computer screen, the following is shown: a psychrometic chart, any number of labeled air conditioned state points, the corresponding air conditioned processes and the air conditioning cycle, and the spreadsheet that lists the thermodynamic properties, airflow in cubic feet per minute, and the heat transfer during the air conditioning processes. The thermodynamic properties of any of the state points and therefore the characteristics of the air conditioning process and cycle can be varied. As a result, the operation of the air system at either full load or part load, under cooling or heating modes, can be investigated and analyzed.

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CHAPTER 3
HEAT AND MOISTURE TRANSFER THROUGH BUILDING ENVELOPE

3.1 BUILDING ENVELOPE 3.2
3.2 HEAT-TRANSFER FUNDAMENTALS 3.2
Conductive Heat Transfer 3.3
Convective Heat Transfer 3.4
Radiant Heat Transfer 3.5
Overall Heat Transfer 3.6
Heat Capacity 3.8
3.3 HEAT-TRANSFER COEFFICIENTS 3.8
Coefficients for Radiant Heat Transfer 3.8
Coefficients for Forced Convection 3.9
Coefficients for Natural Convection 3.10
Surface Heat-Transfer Coefficients 3.10
3.4 MOISTURE TRANSFER 3.11
Sorption Isotherm 3.11
Moisture-Solid Relationship 3.12
Moisture Migration in Building Materials 3.13
Moisture Transfer from the Surface of the Building Envelope 3.14
Convective Mass-Transfer Coefficients 3.15
Moisture Transfer in Building Envelopes 3.16
3.5 CONDENSATION IN BUILDINGS 3.17
Visible Surface Condensation 3.17
Concealed Condensation within the Building Envelope 3.18
3.6 THERMAL INSULATION 3.18
Basic Materials and Thermal Properties 3.19
Moisture Content of Insulation Material 3.19
Economic Thickness 3.21
Thermal Resistance of Airspaces 3.21
3.7 SOLAR ANGLES 3.22
Basic Solar Angles 3.22
Hour Angle and Apparent Solar Time 3.24
Solar Angle Relationships 3.24
Angle of Incidence and Solar Intensity 3.24
3.8 SOLAR RADIATION 3.25
Solar Radiation for a Clear Sky 3.26
Solar Radiation for a Cloudy Sky 3.28
3.9 FENESTRATION 3.29
Types of Window Glass (Glazing) 3.29
Optical Properties of Sunlit Glazing 3.30
3.10 HEAT ADMITTED THROUGH WINDOWS 3.32
Heat Gain for Single Glazing 3.32
Heat Gain for Double Glazing 3.34
Shading Coefficients 3.36
Solar Heat Gain Factors and Total Shortwave Irradiance 3.37
Selection of Glazing 3.39
3.11 SHADING OF GLASS 3.40
Indoor Shading Devices 3.40
External Shading Devices 3.42
Shading from Adjacent Buildings 3.43
3.12 HEAT EXCHANGE BETWEEN THE OUTER BUILDING SURFACE AND ITS SURROUNDINGS 3.46
Sol-Air Temperature 3.47
3.13 COMPLIANCE WITH ASHRAE/IESNA STANDARD 90.1-1999 FOR BUILDING ENVELOPE 3.48
General Requirements 3.48
Mandatory Provisions 3.49
Prescriptive Building Envelope Option 3.49
Building Envelope Trade-Off Option 3.50
3.14 ENERGY-EFFICIENT AND COST-EFFECTIVE MEASURES FOR BUILDING ENVELOPE 3.50
Exterior Walls 3.50
Windows 3.50
Infiltration 3.51
Energy-Efficient Measures for Commercial Buildings in the United States 3.51
REFERENCES 3.51

3.1 BUILDING ENVELOPE

Building envelope consists of the building components that enclose conditioned spaces. Heat, moisture, and contaminants may be transferred to or from the outdoors or unconditioned spaces and therefore affect the indoor environment of the conditioned space.

Building envelope used for air conditioned space in buildings consists of mainly walls, roofs, windows, ceilings, and floors. There are two types of partitions: exterior partitions and demising partitions. An exterior partition is an opaque, translucent, or transparent solid barrier that separates conditioned space from outdoors or space which is not enclosed. A demising partition is a solid barrier that separates conditioned space from enclosed unconditioned space.

- An *exterior wall* is a solid exterior partition which separates conditioned space from the outdoors. Exterior walls are usually made from composite layers including any of the following: stucco, bricks, concrete, concrete blocks, wood, thermal insulation, vapor barrier, airspace, and interior finish. The gross exterior wall area is the sum of the window area, door area, and exterior wall area.
- A *partition wall* is an interior solid barrier which separates a conditioned space from others. Partition walls are usually made from interior surface finishes on two sides, wooden studs and boards, concrete blocks, concrete, bricks, and thermal insulation. A demising partition wall separates a conditioned space from an enclosed unconditioned space.
- *Wall below grade* is a solid barrier below ground level which often separates a basement or a crawl space from soil.
- A *roof* is an exterior partition that has a slope less than 60° from horizontal and has a conditioned space below directly or through a ceiling indirectly. Roofs are usually made from clay tile, waterproof membrane, concrete and lightweight concrete, wood, and thermal insulation.
- A *ceiling* is an interior partition that separates the conditioned space from a ceiling plenum. The ceiling plenum may or may not be air conditioned. Ceilings are usually made from acoustic tile or boards, thermal insulation, and interior surface finishes.
- An *exterior floor* is a horizontal exterior partition under conditioned space. A floor placed over a ventilated basement or a parking space is an exterior floor. Exterior floors are usually made from wood, concrete, thermal insulation, and face tiles.
- *Slab on grade* is a concrete floor slab on the ground. There is usually a vapor barrier, thermal insulation, and gravel and sand fill between the concrete slab and the ground.
- A *window* is glazing of any transparent or translucent material plus sash, frame, mullions, and dividers in the building envelope. Glazing is usually made from glass and transparent plastics. Frames are often made from wood, aluminium, and steel. The window area is the area of the surface of glazing plus the area of frame, sash, and mullions. A fenestration is any area on the exterior building envelope which admits light indoors. Fenestrations include windows, glass doors, and skylights.
- A *skylight* is glazing having a slope of less than 60° from the horizontal. There is often a conditioned space below skylight(s).

3.2 HEAT-TRANSFER FUNDAMENTALS

Heat transfer between two bodies, two materials, or two regions is the result of temperature difference. The science of heat transfer has provided calculations and analyses to predict rates of heat transfer. The design of an air conditioning system must include estimates of heat transfer between the conditioned space, its contents, and its surroundings, to determine cooling and heating loads. Heat-transfer analysis can be described in three modes: conduction, convection, and radiation.

Conductive Heat Transfer

Conduction is the mechanism of heat transfer in opaque solid media, such as through walls and roofs. For one-dimensional steady-state heat conduction q_k, Btu/h (W), Fourier's law gives the following relationship:

$$q_k = -kA\frac{dT}{dx} \tag{3.1}$$

where k = thermal conductivity, Btu/h · ft · °F (W/m · °C)
A = cross-sectional area normal to heat flow, ft^2 (m^2)
T = temperature, °F (°C)
x = coordinate dimension along heat flow, ft (m)

Equation (3.1) shows that the rate of heat transfer is directly proportional to the temperature gradient dT/dx, the thermal conductivity k, and the cross-sectional area A. The minus sign indicates that the heat must flow in the direction of decreasing temperature; i.e., if the temperature decreases as x increases, the gradient dT/dx is negative, so that heat conduction is a positive quantity.

For steady-state heat conduction through a plane composite wall with perfect thermal contact between each layer, as shown in Fig. 3.1, the rate of heat transfer through each section of the composite wall must be the same. From Fourier's law of conduction,

$$q_k = \frac{k_A A}{L_A}(T_1 - T_2) = \frac{k_B A}{L_B}(T_2 - T_3) = \frac{k_C A}{L_C}(T_3 - T_4) \tag{3.2}$$

where L_A, L_B, L_C = thickness of layers A, B, and C, respectively, of composite wall, ft (m)
T_1, T_2, T_3, T_4 = temperature at surfaces 1, 2, 3, and 4, respectively, °F (°C)

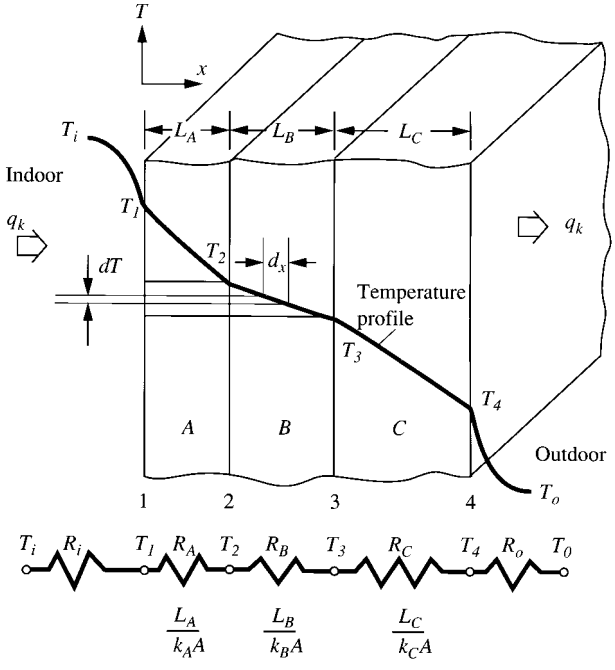

FIGURE 3.1 Steady-state one-dimensional heat conduction through a composite wall.

k_A, k_B, k_C = thermal conductivity of layers A, B, and C, respectively, of composite wall, Btu/h·ft·°F (W/m·°C)

Eliminating T_2 and T_3, we have

$$q_k = \frac{T_1 - T_4}{L_A/(k_A A) + L_B/(k_B A) + L_C/(k_C A)} \tag{3.3}$$

For a multilayer composite wall of n layers in perfect thermal contact, the rate of conduction heat transfer is given as

$$q_k = \frac{T_1 - T_{n+1}}{L_1/(k_1 A) + L_2/(k_2 A) + \cdots + L_n/(k_n A)} \tag{3.4}$$

Subscript n indicates the nth layer of the composite wall.

In Eq. (3.2), conduction heat transfer of any of the layers can be written as

$$q_k = \frac{kA}{L}\Delta T = \frac{\Delta T}{R^*} \qquad R^* = \frac{L}{kA} \tag{3.5}$$

where R^* = thermal resistance, h·°F/Btu (°C/W). In Eq. (3.5), an analogy can be seen between heat flow and Ohm's law for an electric circuit. Here the temperature difference $\Delta T = T_1 - T_2$ indicates thermal potential, analogous to electric potential. Thermal resistance R^* is analogous to electric resistance, and heat flow q_k is analogous to electric current.

The total conductive thermal resistance of a composite wall of n layers R_T, h·°F/Btu (°C/W), can be calculated as

$$R_T^* = R_1^* + R_2^* + \cdots + R_n^* \tag{3.6}$$

where $R_1^*, R_2^*, \cdots R_n^*$ = thermal resistances of layers 1, 2, · · ·, n layer of the composite wall, h·°F/Btu (°C/W). The thermal circuit of a composite wall of three layers is shown in the lower part of Fig. 3.1.

Convective Heat Transfer

Convective heat transfer occurs when a fluid comes in contact with a surface at a different temperature, such as the heat transfer taking place between the airstream in a duct and the duct wall.

Convective heat transfer can be divided into two types: forced convection and natural or free convection. When a fluid is forced to move along the surface by an outside motive force, heat is transferred by forced convection. When the motion of the fluid is caused by the density difference of the two streams in the fluid as a product of contacting a surface at a different temperature, the result is called natural or free convection.

The rate of convective heat transfer q_c, Btu/h (W), can be expressed in the form of Newton's law of cooling as

$$q_c = h_c A(T_s - T_\infty) \tag{3.7}$$

where h_c = average convective heat-transfer coefficient, Btu/h·ft²·°F (W/m²·°C)
T_s = surface temperature, °F (°C)
T_∞ = temperature of fluid away from surface, °F (°C)

In Eq. (3.7), the convective heat-transfer coefficient h_c is usually determined empirically. It is related to a dimensionless group of fluid properties, such as the correlation of flat-plate forced

convection, as shown in the following equation:

$$\mathrm{Nu}_L = \frac{h_c L}{k} = C\,\mathrm{Re}_L^n\,\mathrm{Pr}^m \tag{3.8}$$

where C = constant and k = thermal conductivity, Btu/h·ft·°F (W/m·°C). In Eq. (3.8), Nu_L is the Nusselt number, which is based on a characteristic length L. Characteristic length can be the length of the plate, the diameter of the tube, or the distance between two plates, in feet (meters). The Reynolds number Re_L is represented by the following dimensionless group:

$$\mathrm{Re}_L = \frac{\rho v L}{\mu} = \frac{vL}{\nu} \tag{3.9}$$

where ρ = density of fluid, lb/ft^3 (kg/m^3)
v = velocity of fluid, ft/s (m/s)
μ = absolute viscosity of the fluid, lb/ft·s (kg/m·s)
ν = kinematic viscosity of fluid, ft^2/s (m^2/s)

The Prandtl number Pr is represented by

$$\mathrm{Pr} = \frac{3600\mu c_p}{k} \tag{3.10}$$

where c_p = specific heat at constant pressure of the fluid, Btu/lb·°F (J/kg·°C). Fluid properties used to calculate dimensionless groups are usually related to the fluid temperature T_f. That is,

$$T_f = \frac{T_s - T_\infty}{2} \tag{3.11}$$

Convective heat transfer can also be considered analogous to an electric circuit. From Eq. (3.7), the convective thermal resistance R_c^*, h·°F/Btu (°C/W), is given as

$$R_c^* = \frac{1}{h_c A} \tag{3.12}$$

Radiant Heat Transfer

In radiant heat transfer, heat is transported in the form of electromagnetic waves traveling at the speed of light. The net rate of radiant transfer q_r, Btu/h (W), between a gray body at absolute temperature T_{R1} and a black surrounding enclosure at absolute temperature T_{R2} (for example, the approximate radiation exchange between occupant and surroundings in a conditioned space) can be calculated as

$$q_r = \sigma A_1 \epsilon_1 (T_{R1}^4 - T_{R2}^4) \tag{3.13}$$

where σ = Stefan-Boltzmann constant = 0.1714×10^{-8} Btu/h·ft^2·°R^4 (W/m^2·K^4)
A_1 = area of gray body, ft^2 (m^2)
ϵ_1 = emissivity of surface of gray body
T_{R1}, T_{R2} = absolute temperature of surfaces 1 and 2, °R (K)

If we multiply the right-hand side of Eq. (3.13) by $(T_{R1} - T_{R2})/(T_{R1} - T_{R2})$, and let the thermal resistance

$$R_r^* = \frac{T_{R1} - T_{R2}}{\sigma A_1 \epsilon_1 (T_{R1}^4 - T_{R2}^4)} = \frac{1}{h_r A_1} \tag{3.14}$$

where h_r = radiant heat-transfer coefficient, Btu/h·ft²·°F (W/m²·°C), we find that as a consequence,

$$q_r = h_r A_1 (T_1 - T_2) \tag{3.15}$$

where T_1, T_2 = temperature of surfaces 1 and 2, °F (°C).

If either of the two surfaces is a black surface or can be approximated as a black surface, as was previously assumed for the conditioned space surroundings, the net rate of radiant heat transfer between surfaces 1 and 2 can be evaluated as

$$q_r = \sigma A_1 F_{1-2} (T_{R1}^4 - T_{R2}^4) \tag{3.16}$$

where F_{1-2} = shape factor for a diffuse emitting area A_1 and a receiving area A_2. The thermal resistance can be similarly calculated.

Overall Heat Transfer

In actual practice, many calculations of heat-transfer rates are combinations of conduction, convection, and radiation.

Consider the composite wall shown in Fig. 3.1; in addition to the conduction through the wall, convection and radiation occur at inside and outside surfaces 1 and 4 of the composite wall. At the inside surface of the composite wall, the rate of heat transfer q_i, Btu/h (W), consists of convec-tive heat transfer between fluid, the air, and solid surface q_c and the radiant heat transfer q_r, as follows:

$$\begin{aligned} q_i &= q_c + q_r = h_c A_1 (T_i - T_1) + h_r A_1 (T_i - T_1) \\ &= h_i A_1 (T_i - T_1) \end{aligned} \tag{3.17}$$

where T_i = indoor temperature, °F (°C). From Eq. (3.17), the inside surface heat-transfer coefficient h_i at the liquid-to-solid interface, Btu/h·ft²·°F (W/m²·°C) is

$$h_i = h_c + h_r$$

and the thermal resistance R_i^* of the inner surface due to convection and radiation, h·°F/Btu (°C/W), is

$$R_i^* = \frac{1}{h_i A_1} \tag{3.18}$$

Similarly, at the outside surface of the composite wall, the rate of heat-transfer q_o, Btu/h·ft²·°F (W/m²·°C), is

$$q_o = h_o A_4 (T_4 - T_o) \tag{3.19}$$

where h_o = outside surface heat-transfer coefficient at fluid-to-solid interface, Btu/h·ft²·°F (W/m²·°C)
A_4 = area of surface 4, ft² (m²)
T_4 = temperature of surface 4, °F (°C)
T_o = outdoor temperature, °F (°C)

The outer thermal resistance R_o^*, h·°F/Btu (°C/W), is

$$R_o^* = \frac{1}{h_o A_4}$$

For one-dimensional steady-state heat transfer, the overall heat-transfer rate of the composite

wall q, Btu/h (W), can be calculated as

$$q = q_i = q_k = q_o = UA(T_i - T_o) = \frac{T_i - T_o}{R_T^*} \tag{3.20}$$

where U = overall heat-transfer coefficient, often called the U value, Btu/h·ft²·°F (W/m²·°C)
A = surface area perpendicular to heat flow, ft² (m²)
R_T^* = overall thermal resistance of composite wall, h·°F/Btu (°C/W)

and

$$R_T^* = R_i^* + R_A^* + R_B^* + R_C^* + R_o^* = \frac{1}{UA} \tag{3.21}$$

Also, the thermal resistances can be written as

$$R_A^* = \frac{L_A}{k_A A} \qquad R_B^* = \frac{L_B}{k_B B} \qquad R_C^* = \frac{L_C}{k_C C}$$

Therefore, the overall heat transfer coefficient U is given as

$$U = \frac{1}{1/h_i + L_A/k_A + L_B/k_B + L_C/k_C + 1/h_o} \tag{3.22}$$

For plane surfaces, area $A = A_1 = A_2 = A_3 = A_4$. For cylindrical surfaces, because the inside and outside surface areas are different, and because $UA = 1/R_T^*$, it must be clarified whether the area is based on the inside surface area A_i, outside surface area A_o, or any chosen surface area.

For convenient HVAC&R heat-transfer calculations, the reciprocal of the overall heat-transfer coefficient, often called the *overall R value* R_T, h·ft²·°F/Btu (m²·°C/W), is used. So R_T can be expressed as

$$R_T = \frac{1}{U} = R_i + R_1 + R_2 + \cdots + R_n + R_o \tag{3.23a}$$

where R_i, R_o = R values of inside and outside surfaces of composite wall, h·ft²·°F/Btu (m²·°C/W)
$R_1, R_2, \cdots, R_n$ = R values of components 1, 2, . . ., n, h·ft²·°F/Btu (m²·°C/W)

and

$$R_i = \frac{1}{h_i} \qquad R_1 = \frac{L_1}{k_1} \qquad R_2 = \frac{L_2}{k_2} \qquad \cdots \qquad R_n = \frac{L_n}{k_n} \qquad R_o = \frac{1}{h_o} \tag{3.23b}$$

Sometimes, for convenience, the unit of R is often omitted; for example, R-10 means the R value equals 10 h·ft²·°F/Btu (m²·°C/W).

A building envelope assembly, or a building shell assembly, includes the exterior wall assembly (i.e., walls, windows, and doors), the roof and ceiling assembly, and the floor assembly. The area-weighted average overall heat-transfer coefficient of an envelope assembly U_{av}, Btu/h·ft²·°F (W/m²·°C), can be calculated as

$$U_{av} = \frac{U_1A_1 + U_2A_2 + \cdots + U_nA_n}{A_o} \tag{3.24}$$

where $A_1, A_2 \ldots A_n$ = area of individual elements 1, 2, . . ., n of envelope assembly, ft² (m²)
A_o = gross area of envelope assembly, ft² (m²)
$U_1, U_2, \ldots, U_n$ = overall heat-transfer coefficient of individual paths 1, 2, . . ., n of the envelope assembly, such as paths through windows, paths through walls, and paths through roof, Btu/h·ft²·°F (W/m²·°C)

Heat Capacity

The heat capacity (HC) per square foot (meter) of an element or component of a building envelope or other structure depends on its mass and specific heat. Heat capacity HC, Btu/ft^2 · °F (kJ/m^2 · °C), can be calculated as

$$\frac{HC}{A} = \rho Lc \tag{3.25}$$

where m = mass of building material, lb (kg)
c = specific heat of building material, Btu/lb·°F (kJ/kg · °C)
A = area of building material, ft^2 (m^2)
ρ = density of building material, lb/ft^3 (kg/m^3)
L = thickness or height of building material, ft (m)

3.3 HEAT-TRANSFER COEFFICIENTS

Determination of heat-transfer coefficients to be used for load calculations or year-round energy estimates is complicated by the following types of variables:

- Building envelopes, exterior wall, roof, glass, partition wall, ceiling, or floor
- Fluid flow, turbulent flow, or laminar flow, forced or free convection
- Heat flow, horizontal heat flow in a vertical surface, or an upward or downward heat flow in a horizontal surface
- Space air diffusion, ceiling or sidewall inlet, or others
- Time of operation — summer, winter, or other seasons

Among the three modes of heat transfer, convection processes and their related coefficients are the least understood, making analysis difficult.

Coefficients for Radiant Heat Transfer

For a radiant exchange between the inner surface of an exterior wall and the surrounding surfaces (such as the surfaces of partition walls, ceilings, and floors) in an air conditioned room, the sum of the shape factors ΣF_{1-n} can be considered as unity. If all surfaces are assumed to be black, then the radiative heat-transfer coefficient h_r, Btu/h · ft^2 · °F (W/m^2 · °C), can be calculated as

$$h_r = \frac{\sigma (T_{Ris}^4 - T_{R\text{rad}}^4)}{T_{Ris} - T_{R\text{rad}}} \tag{3.26}$$

where T_{Ris} = absolute temperature of inner surface of exterior wall, roof, or external window glass, °R (K)
$T_{R\text{rad}}$ = absolute mean radiant temperature of surrounding surfaces, °R (K)

Often $T_{R\text{rad}}$ is approximately equal to the air temperature of the conditioned space T_{Rr} when both are expressed in degrees Rankine (kelvins).

Radiant heat transfer coefficients calculated according Eq. (3.26) for various surface temperatures and temperature differences between the inner surface of any building envelope and the surrounded surfaces are presented in Table 3.1. From Table 3.1, h_r depends on the absolute temperature of inner surface T_{Ris} and the temperature difference $T_{is} - T_{\text{rad}}$.

TABLE 3.1 Radiative Heat Transfer Coefficients h_r

Temperature difference $T_{is}-T_{rad}$, °F	Inner surface temperature T_{is}, °F				
	60 (520°R)	70 (530°R)	75 (535°R)	80 (540°R)	85 (545°R)
	Btu/h·ft²·°F				
1	0.871	0.909	0.935	0.968	0.990
2	0.866	0.910	0.936	0.966	0.990
3	0.862	0.908	0.935	0.963	0.989
5	0.856	0.904	0.930	0.958	0.984
10	0.844	0.892	0.918	0.945	0.971
20	0.819	0.868	0.893	0.919	0.945

***Note*:** Calculations made by assuming that $T_{rad} = T_r$ and the emissivity of inner surface $\epsilon_i = 1$.

Coefficients for Forced Convection

Before one can select the mean convective heat-transfer coefficient h_c in order to calculate the rate of convective heat transfer or to determine the overall heat-transfer coefficient U value during cooling load calculations, the type of convection (forced or natural) must be clarified. Forced convection between the surfaces of the building envelope and the conditioned space air occurs when the air-handling system is operating or there is a wind over the outside surface. In an indoor space, free convection is always assumed when the air-handling system is shut off, or there is no forced-air motion over the surface involved.

The convective heat-transfer coefficient h_c also depends on the air velocity v flowing over the surface, the configuration of the surface, the type of space air diffusion, and the properties of the fluid. According to Kays and Crawford (1980), a linear or nearly linear relationship between h_c and v holds. Even though the air velocity v in the occupied zone may be only 30 fpm (0.15 m/s) or even lower when the air-handling system is operating, however, the mode of heat transfer is still considered as forced convection and can be expressed as

$$h_c = A + Bv^n \tag{3.27}$$

where n = exponential index, usually between 0.8 and 1
v = bulky air velocity of fluid 0.5 to 1 ft (0.15 to 0.30 m) from surface, fpm (m/min)
A, B = constants

On the basis of test data from Wong (1990), Sato et al. (1972), and Spitler et al. (1991), as well as many widely used energy estimation computer programs, the convective heat-transfer coefficient for forced convection h_c, Btu/h·ft²·°F (W/m²·°C), for indoor surfaces can be determined as

$$h_c = 1.0 + 0.008v \tag{3.28}$$

For outside surfaces, the surface heat-transfer coefficient $h_o = h_c + h_r$, Btu/h·ft²·°F (W/m²·°C), can be calculated as

$$h_o = 1.8 + 0.004v_{wind} \tag{3.29}$$

where v_{wind} = wind speed, fpm.

Coefficients for Natural Convection

For natural convection, the empirical relationship between the dimensionless groups containing the convective heat-transfer coefficient h_c can be expressed as follows:

$$\mathrm{Nu}_L = C(\mathrm{Gr}_L\mathrm{Pr})^n = C\,\mathrm{Ra}_L^n \tag{3.30}$$

In Eq. (3.30), Gr_L, called the *Grashof number*, is based on the characteristic length L.

$$\mathrm{Gr}_L = \frac{\beta g \rho^2 (T_s - T_\infty)}{\mu^2} \tag{3.31}$$

where β = coefficient of volume expansion of fluid, 1/°R (1/K)
g = acceleration of gravity, ft/s^2 (m/s^2)
ρ = density of fluid, lb/ft^3 (kg/m^3)
C = constant

And Ra is called the *Rayleigh number*, and Ra = GrPr.

Natural convective heat transfer is difficult to evaluate because of the complexity of the recirculating convective stream of room air that is the result of the temperature distribution of the surfaces and the variation of temperature profile of the stream. Computer programs using numerical techniques have been developed recently. In actual practice, simplified calculations are often adopted.

Altmayer et al. (1983) in their recent experiments and analyses found that "The ASHRAE free convection correlations provide a fair prediction of the heat flux to the room air from cold and hot surfaces." The errors are mainly caused by the variation of temperature of the convective airstreams after contact with cold or hot surfaces. In simplified calculations, this variation can only be included in the calculation of the mean space air temperature T_∞.

Many rooms have a vertical wall height of about 9 ft (2.7 m). If the temperature difference $T_s - T_\infty > 1$°F (0.56°C), the natural convection flow is often turbulent. If $T_s - T_\infty \le 1$°F (0.56°C), as in many cooling load calculations between a partition wall and space air, the flow is laminar.

Based on data published in *ASHRAE Handbook 1997, Fundamentals*, the natural convection coefficients h_c, Btu/h·ft^2·°F (W/m^2·°C), are given as follows:

Vertical plates:

$$\text{Large plates, turbulent flow} \qquad h_c = 0.19(\Delta T_{sa})^{0.33} \tag{3.32}$$

$$\text{Small plates, laminar flow} \qquad h_c = 0.29\left(\frac{\Delta T_{sa}}{L}\right)^{0.25} \tag{3.33}$$

Horizontal plates, facing upward when air is heated or facing downward when air is cooled:

$$\text{Large plates, turbulent flow} \qquad h_c = 0.22(\Delta T_{sa})^{0.33} \tag{3.34}$$

$$\text{Small plates, laminar flow} \qquad h_c = 0.27\left(\frac{\Delta T_{sa}}{L}\right)^{0.25} \tag{3.35}$$

Horizontal plates, facing upward when air is cooled or facing downward when air is heated:

$$\text{Small plates} \qquad h_c = 0.12\left(\frac{\Delta T_{sa}}{L}\right)^{0.25} \tag{3.36}$$

Here, ΔT_{sa} indicates the temperature difference between the surface and the air.

Surface Heat-Transfer Coefficients

The surface heat-transfer coefficient h, sometimes called the *surface conductance*, Btu/h·ft^2·°F (W/m^2·°C), is the combination of convective and radiant heat-transfer coefficients; that is,

TABLE 3.2 Surface Heat-Transfer Coefficients h, Btu/h · ft^2 · °F

		Surface emissivity ϵ						
		0.90				0.20		
		Indoor surface				Indoor surface		
		ΔT_{sa} = 10°F				ΔT_{sa} = 10°F		
Description	Direction of heat flow	Summer	Winter	ΔT_{sa} = 1°F	Outdoor surface	Summer	Winter	ΔT_{sa} = 1°F
Forced convection								
<30 fpm		2.21	2.11	2.16		1.46	1.43	1.44
50 fpm		2.37	2.27	2.32		1.62	1.59	1.60
660 fpm (7.5 mph)					4.44			
1320 fpm (15 mph)					7.08			
Free convection								
Horizontal surface	Upward	1.36	1.27	1.17		0.62	0.60	0.44
Vertical surface		1.42	1.33	1.15		0.68	0.66	0.30
Horizontal surface	Downward			1.03				

***Note*:** Assume space temperature T_r = 74°F year-round and $T_r = T_{\text{rad}}$; here T_{rad} indicates the mean radiant temperature of the surroundings.

$h = h_c + h_r$. Table 3.2 lists the h values for various surface types at ΔT_{sa} = 10°F (5.6°C) and ΔT_{sa} = 1°F (0.56°C) during summer and winter design conditions. The values are based upon the following:

- $T_{\text{rad}} = T_a$, where T_a is the air temperature, °F (°C).
- ΔT_{sa} indicates temperature difference between surface and air, °F (°C).
- Emissivity of the surface ϵ = 0.9 and ϵ = 0.2.

3.4 MOISTURE TRANSFER

Moisture is water in the vapor, liquid, and solid states. Building materials exposed to excessive moisture may degrade or deteriorate as a result of physical changes, chemical changes, and biological processes. Moisture accumulated inside the insulating layer also increases the rate of heat transfer through the building envelope. Moisture transfer between the building envelope and the conditioned space air has a significant influence on the cooling load calculations in areas with hot and humid climates.

Sorption Isotherm

Moisture content X, which is dimensionless or else in percentage, is defined as the ratio of the mass of moisture contained in a solid to the mass of the bone-dry solid. An *absorption isotherm* is a constant-temperature curve for a material in which moisture content is plotted against an increased ambient relative humidity during an equilibrium state; i.e., the rate of condensation of water vapor on the surface of the material is equal to the rate of evaporation of water vapor from the material. A *desorption isotherm* is also a constant-temperature curve for a material. It is a plot of moisture content versus a decreased ambient relative humidity during equilibrium state.

Many building materials show different absorption and desorption isotherms. The difference in moisture content at a specific relative humidity between the absorption and desorption isotherms is called *hysteresis*. Figure 3.2 shows absorption and desorption isotherms of a building material.

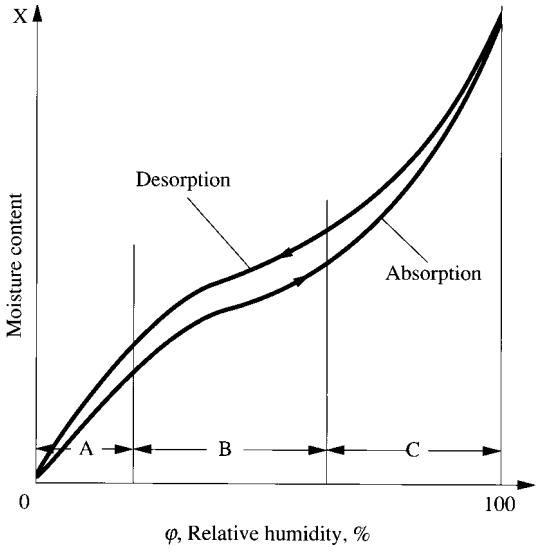

FIGURE 3.2 Sorption isotherms.

Temperature also has an influence on the moisture content of many building materials. At a constant relative humidity in ambient air, the moisture content of a building material will be lower at a higher temperature.

When a building material absorbs moisture, heat as heat of sorption is evolved. If liquid water is absorbed by the material, an amount of heat q_l, Btu/lb (kJ/kg) of water absorbed, similar to the heat of solution, is released. This heat results from the attractive forces between the water molecules and the molecules of the building material. If water vapor is absorbed, then the heat released q_v, Btu/lb (kJ/kg), is given by

$$q_v = q_l + h_{fg} \tag{3.37}$$

where h_{fg} = latent heat of condensation, Btu/lb (kJ/kg). Heat of sorption of liquid water q_l varies with equilibrium moisture content for a given material. The lower the X and the φ of ambient air, the higher will be the value of q_l. For pine, q_l may vary from 450 Btu/lb (1047 kJ/kg) for nearly bone-dry wood to 20 Btu/lb (46.5 kJ/kg) at a moisture content of 20 percent. Many building materials have very low q_l values compared with h_{fg}, such as a q_l of 40 Btu/lb (93 kJ/kg) for sand.

Moisture-Solid Relationship

Many building materials have numerous interstices and microcapillaries of radius less than 4×10^{-6} in. (0.1 μm), which may or may not be interconnected. These interstices and microcapillaries provide large surface areas to absorb water molecules. Moisture can be bound to the solid surfaces by retention in the capillaries and interstices, or by dissolution into the cellular structures of fibrous materials.

When the relative humidity φ of ambient air is less than 20 percent, moisture is tightly bound to individual sites in the monomolecular layer (region *A*, in Fig. 3.2). Moisture moves by vapor diffusion. The binding energy is affected by the characteristics of the surface, the chemical structure of the material, and the properties of water.

When relative humidity φ is 20 to 60 percent (region *B*, Fig. 3.2), moisture is bound to the surface of the material in multimolecular layers and is held in microcapillaries. Moisture begins to migrate in liquid phase, and its total pressure is reduced by the presence of moisture in microcapillaries. The binding energy involved is mainly the latent heat of condensation.

When $\varphi > 60$ percent, i.e., in region *C* of Fig. 3.2, moisture is retained in large capillaries. It is relatively free for removal and chemical reactions. The vapor pressure of the moisture is influenced only moderately because of the moisture absorbed in regions *A* and *B*. Moisture moves mainly in a liquid phase.

Because the density of the liquid is much greater than that of water vapor, the moisture content in building materials is mostly liquid. Unbound moisture can be trapped in interstices having a radius greater than 4×10^{-5} in. (1 μm), without significantly lowering the vapor pressure. Free moisture is the moisture in excess of the equilibrium moisture content in a solid. For insulating materials in which interstices are interconnected, air penetrates through these open-cell structures, and, moisture can be accumulated in the form of condensate and be retained in the large capillaries and pores.

Moisture Migration in Building Materials

Building envelopes are not constructed only with open-cell materials. The airstream and its associated water vapor cannot penetrate building envelopes. Air leakage can only squeeze through the cracks and gaps around windows and joints. However, all building materials are moisture-permeable; in other words, moisture can migrate across a building envelope because of differences in moisture content or other driving potentials.

According to Wong and Wang (1990), many theories have been proposed by scientists to predict the migration of moisture in solids. The currently accepted model of moisture flow in solids is based upon the following assumptions:

- Moisture migrates in solids in both liquid and vapor states. Liquid flow is induced by capillary flow and concentration gradients; vapor diffusion is induced by vapor pressure gradients.
- During the transport process, the moisture content, the vapor pressure, and the temperature are always in equilibrium at any location within the building material.
- Heat transfer within the building material is in the conduction mode. It is also affected by latent heat from phase changes.
- Vapor pressure gradients can be determined from moisture contents by means of sorption isotherms.
- Fick's law is applicable.
- Only one-dimensional flow across the building envelope is considered. Building materials are homogeneous.

If the temperature gradient is small, for simplicity, the mass flux $\dot{m}_w/A$ for one-dimensional flow, lb/h · ft^2 (kg/h · m^2), can be expressed as

$$\dot{m}_w/A = -\rho_w D_{lv} \frac{dX}{dx} \tag{3.38}$$

where D_{lv} = mass diffusivity of liquid and vapor, ft^2/h (m^2/h)
X = moisture content, lb/lb (kg/kg) dry solid
ρ_w = density of water, lb/ft^3 (kg/m^3)
x = coordinate dimension along moisture flow, ft (m)
A = area of building envelope perpendicular to moisture flow, ft^2 (m^2)

Mass diffusivities of some building materials as a function of moisture content are shown in Fig. 3.3.

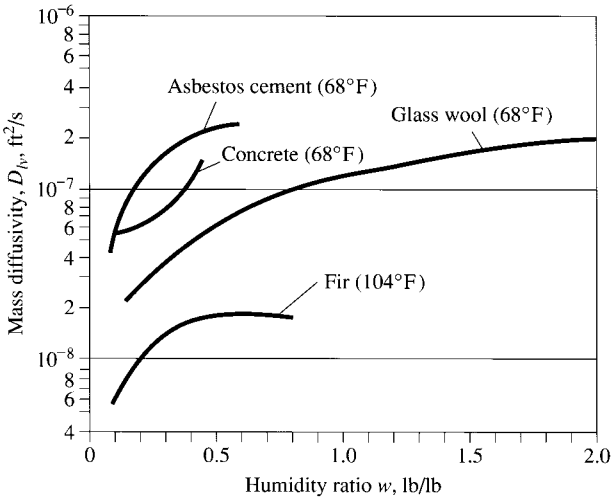

FIGURE 3.3 Mass diffusivity D_{lv} of some building materials. (*Abridged with permission from Bruin et al. Advances in Drying, Vol. 1, 1979.*)

Moisture Transfer from the Surface of the Building Envelope

At a certain time instant, moisture migrating from any part in the building envelope to its surface must be balanced by convective moisture transfer from the surface of the building envelope to the ambient air and the change of the moisture content as well as the corresponding mass concentration at the surface of the building envelope. Such a convective moisture transfer is often a part of the space latent cooling load. Analogous to the rate of convective heat transfer [Eq. (3.7)], the rate of convective moisture transfer $\dot{m}_w$, lb/h (kg/h), can be calculated as

$$\dot{m}_w = h_m A_w (C_{ws} - C_{wr}) \tag{3.39}$$

where h_m = convective mass-transfer coefficient, ft/h (m/h)
A_w = contact area between moisture and ambient air, ft^2 (m^2)
C_{ws}, C_{wr} = mass concentration of moisture at surface of building envelope and of space air, lb/ft^3 (kg/m^3)

It is more convenient to express the mass concentration difference $C_{ws} - C_{wr}$ in terms of a humidity ratio difference $w_s - w_r$. Here w_s and w_r represent the humidity ratio at the surface of the building envelope and of the space air, respectively, lb/lb (kg/kg). In terms of mass concentration we can write

$$C_{ws} = \rho_a w_s \tag{3.40}$$

$$C_{wr} = \rho_a w_r \tag{3.41}$$

Then the rate of convective moisture transfer can be rewritten as

$$\dot{m}_w = \rho_a h_m A_w (w_s - w_r) \tag{3.42}$$

In Eq. (3.42), the surface humidity ratio w_s depends on the moisture content X_s, the temperature T_s, and therefore the vapor pressure p_{ws} in the interstices of the surface of the building envelope. From the known X_s, T_s, and the sorption isotherm, the corresponding relative humidity φ_s at the surface can be determined. If the difference between relative humidity φ and the degree of saturation μ is

ignored, then

$$w_s = \mu_s w_{ss} \approx \varphi_s w_{ss} \tag{3.43}$$

where μ_s = degree of saturation at surface. In Eq. (3.43), w_{ss} represents the humidity ratio of the saturated air. It can be determined from Eq. (2.49) since T_s is a known value.

In Eq. (3.42), the contact area A_w between the liquid water at the surface of the building envelope and the space air is a function of moisture content X_s. A precise calculation of A_w is very complicated, but if the surface area of the building material is A_s, then a rough estimate can be made from

$$A_w = X_s A_s \tag{3.44}$$

Convective Mass-Transfer Coefficients

The Chilton-Colburn analogy relates the heat and mass transfer in these forms:

$$\begin{gathered} j_H = j_D \\ \frac{h_c}{\rho_a \nu c_{pa}} \mathrm{Pr}^{2/3} = \frac{h_m}{v} \mathrm{Sc}^{2/3} \\ \frac{h_c}{\rho_a c_{pa}} \left(\frac{\nu_a}{\alpha}\right)^{2/3} = h_m \left(\frac{v_w}{D_{aw}}\right)^{2/3} \end{gathered} \tag{3.45}$$

where v = air velocity remote from surface, ft/s (m/s)
c_{pa} = specific heat of moist air, Btu/lb·°F (J/kg·°C)
h_m = convective mass-transfer coefficient, ft/h (m/s)
ν = kinematic viscosity, ft^2/s (m^2/s)
α = thermal diffusivity of air, ft^2/s (m^2/s)
D_{aw} = mass diffusivity for water vapor diffusing through air, ft^2/s (m^2/s)
Sc = Schmidt number

Subscript a indicates dry air and w, water vapor.

For air at 77°F (25°C), $c_{pa} = 0.243$ Btu/lb·°F (1020 kJ/kg·°C), $\nu_a = 1.74 \times 10^{-4}$ ft^2/s (1.62×10^{-5} m^2/s), $\alpha = 2.44 \times 10^{-4}$ ft^2/s (2.27×10^{-5} m^2/s), $D_{aw} = 2.83$ ft^2/s (0.263 m^2/s), and $\rho_a = 0.0719$ lb/ft^3 (1.15 kg/m^3), we can show that

$$\mathrm{Pr} = \frac{\nu_a}{\alpha} = \frac{1.74 \times 10^{-4}}{2.44 \times 10^{-4}} = 0.713$$

and that

$$\mathrm{Sc} = \frac{\nu_a}{D_{aw}} = \frac{1.74 \times 10^{-4}}{2.83 \times 10^{-4}} = 0.614$$

At a space temperature of 77°F (25°C), therefore,

$$\begin{aligned} h_m &= \frac{h_c \mathrm{Pr}^{2/3}}{\rho_a c_{pa} \mathrm{Sc}^{2/3}} \\ &= \frac{h_c (0.713)^{2/3}}{0.0719 \times 0.243 \times (0.614)^{2/3}} = 63.2 h_c \end{aligned}$$

Moisture Transfer in Building Envelopes

Moisture transfer in building envelopes of a typical residential building can proceed along two paths, as shown in Fig. 3.4:

1. Moisture migrates inside the building material mainly in the form of liquid if the relative humidity of the ambient air is more than 50 percent. It will be transported to the indoor or outdoor air by convective mass transfer. The driving potentials are the moisture content of the building material, the vapor pressure gradient inside the building material, and the humidity ratio gradient between the surface and the ambient air.

2. Air leakage and the associated water vapor infiltrates or exfiltrates through the cracks and gaps around the windows, doors, fixtures, outlets, and between the joints. Air and moisture enter the cavities and the airspace in the building envelope. If the sheathing is not airtight, air leakage and its water vapor penetrate the perforated insulating board and discharge to the atmosphere, as shown in Fig. 3.4. If the sheathing is a closed-cell, airtight insulating board, then the airstream may infiltrate through gaps between the joints of the insulating board or through cracks between the window sill and the external wall and discharge to the atmosphere.

The driving potential of the air leakage and the associated water vapor is the total pressure differential between indoor and outdoor air across the building envelope due to wind effects, stack effect, mechanical ventilation, or a combination of these. Moisture is moving in the vapor form.

In leaky buildings, the moisture transfer by means of air leakage is often far greater than the moisture migration through solids. For better-sealed commercial buildings, moisture migration through the building envelope may be important.

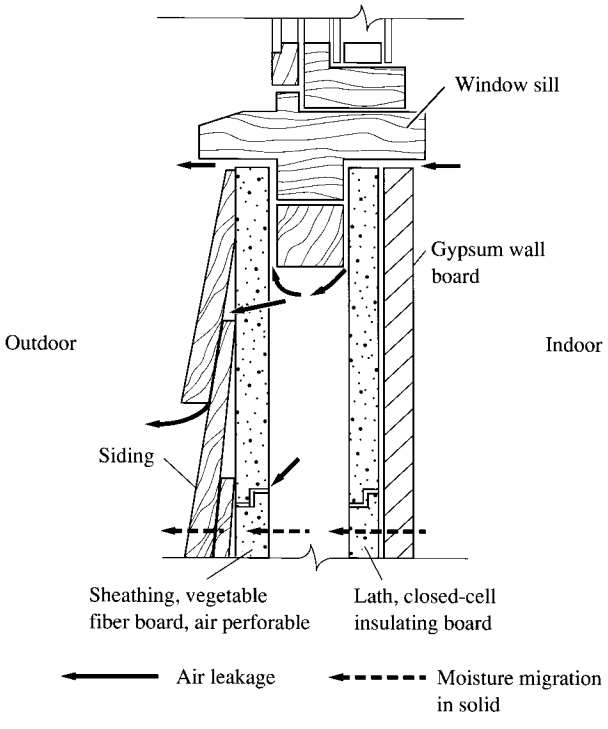

FIGURE 3.4 Moisture transfer in building envelope along two paths.

Installation of vapor retarders is an effective means to block or to reduce the moisture transfer in the building envelope. Vapor retarders are covered in greater detail later.

3.5 CONDENSATION IN BUILDINGS

When moist air contacts a solid surface whose temperature is lower than the dew point of the moist air, condensation occurs on the surface in the form of liquid water, or sometimes frost. Condensation can damage the surface finish, deteriorate the material and cause objectionable odors, stains, corrosion, and mold growth; reduce the quality of the building envelope with dripping water; and fog windows. Two types of condensation predominate in buildings:

1. Visible surface condensation on the interior surfaces of external window glass, below-grade walls, floor slabs on grade, and cold surfaces of inside equipment and pipes
2. Concealed condensation within the building envelope

Visible Surface Condensation

To avoid visible surface condensation, either the dew point of the indoor air must be reduced to a temperature below that of the surface, or the indoor surface temperature must be raised to a level higher than the dew point of the indoor air. The indoor surface temperature of the building envelope, such as T_1 of the composite exterior wall in Fig. 3.1, or any cold surface where condensation may occur can be calculated from Eqs. (3.17) and (3.20) as

$$T_1 = T_i - \frac{U(T_i - T_o)}{h_i} \tag{3.46}$$

where q = rate of heat transfer through building envelope, Btu/h (W)
h_i = inside surface heat-transfer coefficient, Btu/h·ft^2·°F (W/m^2·°C)

Increasing the thermal insulation is always an effective and economical way to prevent condensation because it saves energy and raises the surface temperature of the building envelope and other cold surfaces.

The dew point temperature of the indoor air $T_{r\cdot\text{dew}}$ is a function of humidity ratio w_r of the space air. At a specific space temperature T_r, the lower the relative humidity, the lower the $T_{r\cdot\text{dew}}$. There are several ways to lower the dew point of the indoor space air:

- By blocking and controlling infiltration of hot, humid outside air
- By reducing indoor moisture generation
- By using a vapor retarder, such as polyethylene film or asphalt layer, to prevent or decrease the migration of moisture from the soil and the outside wetted siding and sheathing
- By introducing dry outdoor air through mechanical or natural ventilation when doing so is not in conflict with room humidity criteria
- By using dehumidifiers to reduce the humidity ratio of the indoor air

During winter, any interior surface temperature of the wall, roof, or glass is always lower than the indoor space temperature. Better insulation and multiple glazing increase the interior surface temperature and, at the same time, reduce the heat loss, providing a more comfortable indoor environment than is possible by decreasing the relative humidity. To reduce excessive indoor humidity in many industrial applications, a local exhausting booth that encloses the moisture-generating source is usually the remedy of first choice.

During summer, the outer surface of chilled water pipes and refrigerant pipes (even the cold supply duct) is at a lower temperature than the indoor air temperature. Sufficient pipe and duct insulation is needed to prevent surface condensation. When an air-handling system is shut down, the heavy construction mass often remains at a comparatively lower surface temperature as the indoor humidity rises. To avoid indoor surface condensation, the enclosure must be tight enough to prevent the infiltration of hot, humid air from outdoors.

Concealed Condensation within the Building Envelope

Usually, concealed condensation within building materials does not accumulate as a result of moisture migration. Excessive free moisture, usually in the form of liquid or frost, at any location inside the building envelope results in a higher moisture content than in surrounding areas. Therefore, it produces a moisture migration outward from that location rather than into it.

During cold seasons, concealed condensation in building envelopes is mainly caused by the warm indoor air, usually at a dew-point temperature of 32°F (0°C) and above. It leaks outward through the cracks and openings and enters the cavities, the gaps between component layers, or even the penetrable insulating material. It ultimately contacts a surface at a temperature lower than the dew point of the indoor air. Donnelly et al. (1976) discovered that a residential stud wall panel with a poor vapor retarder accumulated about 3 lb (14.6 kg) of moisture per square foot (meter) of wall area during a period of 31 days in winter. This moisture accumulated in the mineral fiber insulating board adjacent to the cold side of the sheathing and at the interface between the insulating board and the sheathing.

The rate of condensation $\dot{m}_{\text{con}}$, lb/h (kg/h), can be calculated as

$$\dot{m}_{\text{con}} = 60\dot{V}_{lk}\rho_r(w_r - w_{s\,\text{con}}) \tag{3.47a}$$

where $\dot{V}_{lk}$ = volume flow rate of air leakage, cfm (m^3/min)
ρ_r = density of indoor air, lb/ft^3 (kg/m^3)
$w_{s\,\text{con}}$ = saturated humidity ratio corresponding to temperature of surface upon which condensation occurs, lb/lb (kg/kg)

Vapor retarders are effective for reducing moisture transfer through the building envelope. They not only decrease the moisture migration in the building material significantly, but also block air leakage effectively if their joints are properly sealed. Vapor retarders are classified as rigid, flexible, and coating types. Rigid type includes reinforced plastic, aluminum, and other metal sheets. Flexible type includes metal foils, coated films, and plastic films. Coatings are mastics, paints, or fusible sheets, composed of asphalt, resin, or polymeric layers. The vapor retarder is generally located on the warm side of the insulation layer during winter heating.

In an area where summer cooling is dominant, a vapor retarder to prevent condensation in the external brick wall and insulation layer would often be located on the outside of the insulation. The absorptive brick wall takes substantial amounts of water during rainfall. The subsequent sunny period drives the water vapor farther inside the wall and into the insulation layer where the water vapor condenses.

3.6 THERMAL INSULATION

Thermal insulation materials, usually in the form of boards, slabs, blocks, films, or blankets, retard the rate of heat transfer in conductive, convective, and radiant transfer modes. They are used within building envelopes or applied over the surfaces of equipment, piping, or ductwork to achieve the following benefits:

- Savings of energy by reducing heat loss and heat gain from the surroundings

- Prevention of surface condensation by increasing the surface temperature above the dew point of the ambient air
- Reduction of temperature difference between the inside surface and the space air for the thermal comfort of the occupants, when radiant heating or cooling is not desired
- Protection of the occupant from injury due to contact with hot pipes and equipment

Basic Materials and Thermal Properties

Basic materials in the manufacture of thermal insulation for building envelopes or air conditioning systems include

- Fibrous materials such as glass fiber, mineral wool, wood, cane, or other vegetable fibers
- Cellular materials such as cellular glass, foam rubber, polystyrene, and polyurethane
- Metallic reflective membranes

Most insulating materials consist of numerous airspaces, either closed cells (i.e., airtight cellular materials) or open cells (i.e., fibrous materials, penetrable by air).

Thermal conductivity k, an important physical property of insulating material, indicates the rate of heat transfer by means of a combination of gas and solid conduction, radiation, and convection through an insulating material, expressed in $\text{Btu} \cdot \text{in./h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ or $\text{Btu/h} \cdot \text{ft} \cdot {}^\circ\text{F}$ ($\text{W/m} \cdot {}^\circ\text{C}$). The thermal conductivity of an insulating material depends on its physical structure (such as cell size and shape or diameter of the fibrous materials), density, temperature, and type and amount of binders and additives.

Most of the currently used thermal insulating materials have thermal conductivities within a range of 0.15 to 0.4 $\text{Btu} \cdot \text{in./h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ (0.021 to 0.058 $\text{W/m} \cdot {}^\circ\text{C}$). Thermal properties of some building and insulating materials, based on data published in the *ASHRAE Handbook 1993*, *Fundamentals*, are listed in Table 3.3.

The thermal conductivity of many cellular insulating materials made from polymers like polyurethane and extruded polystyrene is not significantly affected by change in density ρ. Other insulating materials have a density at which the thermal conductivity is minimum. Deviating from this ρ, conductivity k increases always, whether ρ is greater or smaller in magnitude. The thermal conductivity of an insulating material apparently increases as its mean temperature rises. For polystyrene, k increases from 0.14 to 0.32 $\text{Btu} \cdot \text{in./h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ (0.020 to 0.046 $\text{W/m} \cdot {}^\circ\text{C}$) when its mean temperature is raised from 300 to 570°F (150 to 300°C).

Moisture Content of Insulation Material

From the point of view of moisture transfer, penetrability is an important characteristic. Closed-cell airtight board or block normally cannot have concealed condensation except at the gap between the joints and at the interface of two layers. Only when air and its associated water vapor penetrate an open-cell insulating material can concealed condensation form if they contact the surfaces of the interstices and pores at a temperature lower than the dew point of the penetrating air. Concealed condensation might also accumulate in open-cell insulating materials. If there is excess free moisture in an insulating material, the thermal insulation may be degraded. The increase of moisture in the thermal insulation layer is often due to the absorption from the ambient air, space air, or moisture transfer from adjacent layer, ground, or wetted surface.

Smolenski (1996) brought forward the question of how much is too much. In roofing insulation, many consider the answer to be the moisture content to produce a more than 20 percent loss in thermal efficiency or a thermal resistance ratio (TRR) of less than 80 percent. The *thermal resistance ratio* is defined as the ratio of wet to dry thermal resistance of the thermal insulation, in percent, or

$$\text{TRR} = \frac{(\text{wet thermal resistance}) \times 100}{\text{dry thermal resistance}} \tag{3.47b}$$

TABLE 3.3 Thermal Properties of Selected Materials

	Density, lb/ft^3	Thermal conductivity, (Btu/h·ft·°F)	Specific heat, (Btu/lb·°F)	Emissivity	
Aluminum (alloy 1100)	171	128	0.214	0.09	
Asbestos: insulation	120	0.092	0.20	0.93	
Asphalt	132	0.43	0.22		
Brick, building	123	0.4	0.2	0.93	
Brass (65% Cu, 35% Zn)	519	69	0.09	0.033	Highly polished
Concrete (stone)	144	0.54	0.156		
Copper (electrolytic)	556	227	0.092	0.072	Shiny
Glass: crown (soda-lime)	154	0.59	0.18	0.94	Smooth
Glass wool	3.25	0.022	0.157		
Gypsum	78	0.25	0.259	0.903	Smooth plate
Ice (32°F)	57.5	1.3	0.487	0.95	
Iron: cast	450	27.6	0.12	0.435	Freshly turned
Mineral fiberboard:					
acoustic tile, wet-molded	23	0.035	0.14		
wet-felted	21	0.031	0.19		
Paper	58	0.075	0.32	0.92	
Polystyrene, expanded, molded beads	1.25	0.021	0.29		
Polyurethane, cellular	1.5	0.013	0.38		
Plaster, cement and sand	132	0.43		0.91	Rough
Platinum	1340	39.9	0.032	0.054	Polished
Rubber: vulcanized, soft	68.6	0.08	0.48	0.86	Rough
hard	74.3	0.092		0.95	Glossy
Sand	94.6	0.19	0.191		
Steel (mild)	489	26.2	0.12		
Tin	455	37.5	0.056	0.06	Bright
Wood: fir, white	27	0.068	0.33		
oak, white	47	0.102	0.57	0.90	Planed
plywood, Douglas fir	34	0.07	0.29		
Wool: fabric	20.6	0.037			

***Source*:** Adapted with permission from *ASHRAE Handbook 1989, Fundamentals*.

The equilibrium moisture content of most commomly used insulation material at 90 percent relative humidity ambient air, as well as the moisture content of insulation material at 80 percent TRR by weight (percent of dry weight) and by volume (percent of volume of insulating material) are listed below:

Insulation material	Density, lb/ft^3	$\varphi = 90\%$	Moisture content, %	
			80% TRR, by weight	80% TRR, by volume
Cellular glass	8.4	0.2	23	3.1
Expanded polystyrene	1.0	2.0	383	6.1
Glass fiber	9.2	1.1	42	6.2
Urethane	2.1	6.0	262	8.8
Phenolic foam	2.6	23.4	25	1.0

For instance, for cellular glass when in contact with an ambient air of 90 percent relative humidity at normal room temperaure that reaches an equilibrium state during moisture absorption, it has

only a moisture content of 0.2 percent by dry weight, which is far lower than the moisture content of 23 percent by weight at 80 percent TRR. Most above-listed insulating materials have a far lower equilibrium moisture content of dry weight at an ambient air of 90 percent relative humidity than moisture content at 80 percent TRR. Also, a closed-cell structure cellular glass requires months or years to reach an equilibrium moisture content by volume corrresponding to 80 percent TRR whereas mineral wool and calcium silica absorb moisture in only hours.

Economic Thickness

The economic thickness of insulation is the thickness with the minimum owning and operating costs. Owning cost is the net investment cost of the installed insulation C_{in}, in dollars, less any capital investment that can be made as a result of lower heat loss or gain C_{pt}, in dollars. Theoretically, for a new plant, some small savings might be made because of the reduction of the size of a central plant; but in actual practice, this is seldom considered. For an existing plant where add-on insulation is being considered, C_{pt} is zero because the plant investment has already been made.

Operating cost C_{en} includes the annualized cost of energy over the life of a new plant, or the remaining life of an existing plant, in n years. It can also be taken as the number of years over which an owner wishes to have a total return of the net investment, considering both the interest and the fuel escalation rate. Total cost of insulation C_t, in dollars, for any given thickness is

$$C_t = C_{\text{in}} - C_{pt} + C_{en} \tag{3.48}$$

When the thickness of the insulating material increases, the quantity $C_{\text{in}} - C_{pt}$ also increases, as shown in Fig. 3.5, and C_{en} decreases. As a result, C_t first decreases, drops to a minimum, and then increases. The optimum economic thickness occurs when C_t drops to a minimum. The closest commercially available thickness is the optimum thickness.

Thermal Resistance of Airspaces

Thermal resistance of an enclosed airspace R_a has a significant effect on the total thermal resistance R_T of the building envelope, especially when the value of R_T is low. Thermal resistance R_a depends on the characteristic of the surface (reflective or nonreflective), the mean temperature, the temperature difference of the surfaces perpendicular to heat flow, the width across the airspace along the heat flow,

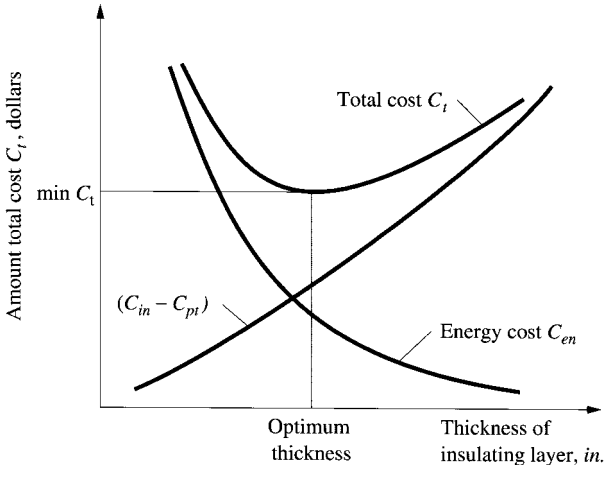

FIGURE 3.5 Optimum thickness of insulation material.

TABLE 3.4 *R* Values of Enclosed Airspace R_a, $h \cdot ft^2 \cdot °F/Btu$

Mean temperature, °F	Temperature difference, °F	Type of surface	Direction of heat flow	Width of airspace, in.	Emissivity $E^\dagger$		
					0.05	0.2	0.82
Summer, 90	10	Horizontal	Upward	0.5	2.03	1.51	0.73
				3.5	2.66	1.83	0.80
		Vertical	Horizontal	0.5	2.34	1.67	0.77
				3.5	3.40	2.15	0.85
		Horizontal	Downward	0.5	2.34	1.67	0.77
				3.5	8.19	3.41	1.00
Winter, 50	10	Horizontal	Upward	0.5	2.05	1.60	0.84
				3.5	2.66	1.95	0.93
		Vertical	Horizontal	0.5	2.54	1.88	0.91
				3.5	3.40	2.32	1.01
		Horizontal	Downward	0.5	2.55	1.89	0.92
				3.5	9.27	4.09	1.24

†Emissity $E = 1/(1/e_1 + 1/e_2 - 1)$, where e_1, e_2 indicate the emittances on two sides of the airspace.
***Source*:** Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

and the direction of airflow. The *R* values of the enclosed airspaces R_a, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$), abridged from data published in *ASHRAE Handbook 1989*, *Fundamentals*, are presented in Table 3.4.

3.7 SOLAR ANGLES

Basic Solar Angles

The basic solar angles between the sun's rays and a specific surface under consideration are shown in Figs. 3.6 and 3.7.

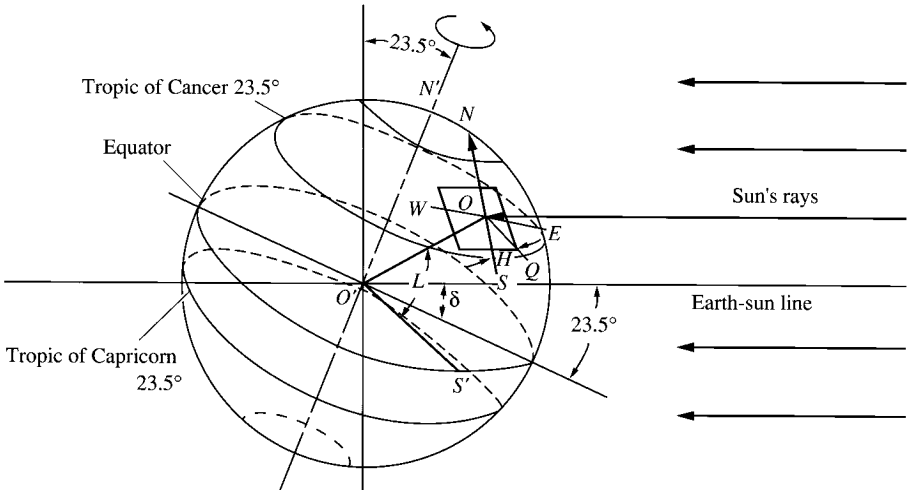

FIGURE 3.6 Basic solar angles and position of sun's rays at summer solstice.

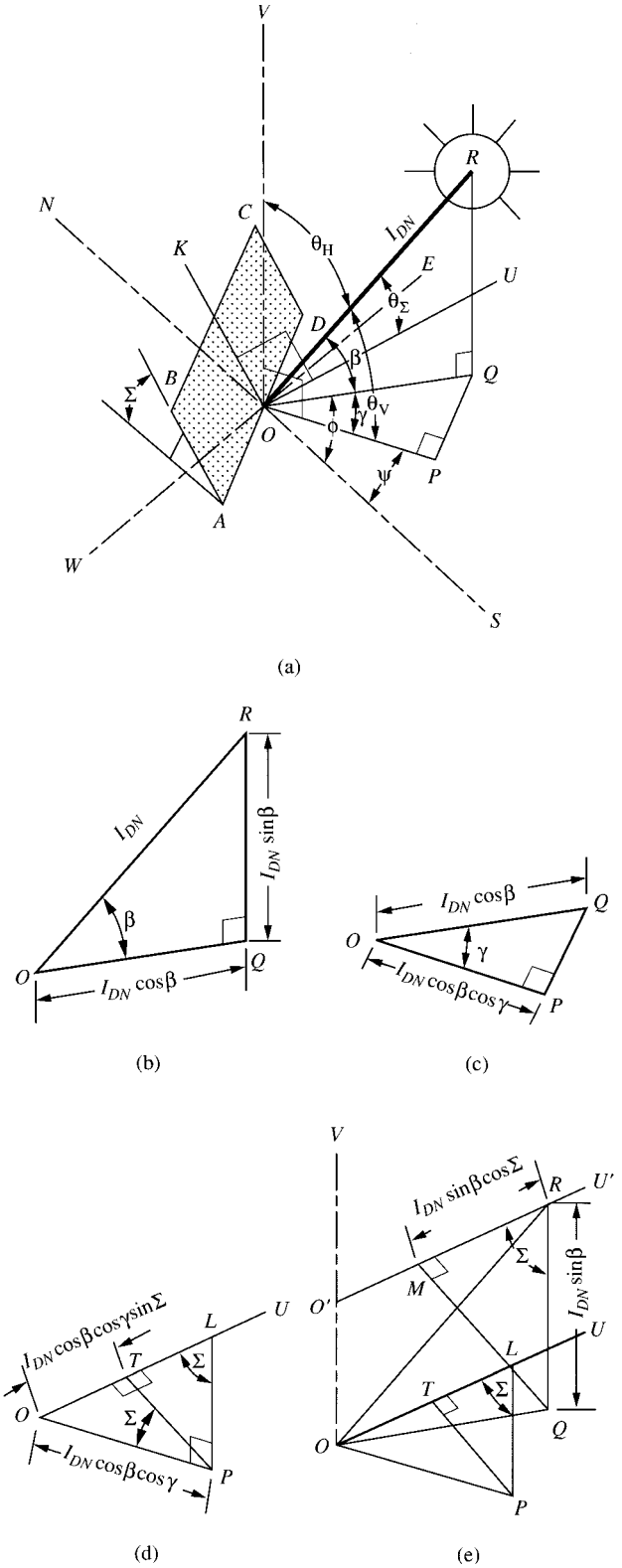

FIGURE 3.7 Solar intensity and angle of incidence.

- Solar altitude angle β (Fig. 3.7*a* and *b*) is the angle *ROQ* on a vertical plane between the sun's ray *OR* and its projection on a horizontal plane on the surface of the earth.
- Solar azimuth Φ (Fig. 3.7*a*) is the angle *SOQ* on a horizontal plane between the due-south direction line *OS* and the horizontal projection of the sun's ray *OQ*.
- Solar declination angle δ (Fig. 3.6) is the angle between the earth-sun line and the equatorial plane. Solar declination δ changes with the times of the year. It is shown in Fig. 3.6 on June 21.
- Surface-solar azimuth γ (Fig. 3.7*a* and *c*) is the angle *POQ* on a horizontal plane between the normal to a vertical surface *OP* and the horizontal projection of the sun's rays *OQ*.
- Surface azimuth Ψ (Fig. 3.7*a*) is the angle *POS* on a horizontal plane between *OP* and the direction line *SN*.
- Latitude angle *L* (Fig. 3.6) is the angle $S'O'O$ on the longitudinal plane between the equatorial plane and the line $O'O$ that connects the point of incidence of the sun's ray on the surface of earth *O* and the center of the earth O'.

Hour Angle and Apparent Solar Time

Hour angle *H* (Fig. 3.6) is the angle *SOQ* on a horizontal plane between the line *OS* indicating the noon of local solar time t_{ls} and the horizontal projection of the sun's ray *OQ*. The values of the hour angle *H* before noon are taken to be positive. At 12 noon, *H* is equal to 0. After 12 noon, *H* is negative. Hour angle *H*, in degrees, can be calculated as

$$H = 0.25 \text{ (time in minutes from local solar noon)} \tag{3.49}$$

The relationship between apparent solar time t_{as}, as determined by a sundial and expressed in apparent solar time, and local standard time t_{st}, both in minutes, is as follows:

$$t_{as} = t_{\text{eq}} + t_{st} \pm 4(M - G) \tag{3.50}$$

where M = local standard time meridian and G = local longitude, both in degrees. In Eq. (3.50), t_{eq}, in minutes, indicates the difference in time between the mean time indicated by a clock running at a uniform rate and the solar time due to the variation of the earth's orbital velocity throughout the year.

Solar Angle Relationships

The relationship among the solar angles is given as

$$\sin\beta = \sin\delta \sin L + \cos\delta \cos H \cos L \tag{3.51}$$

and

$$\cos\Phi = \frac{\sin\beta \sin L - \sin\delta}{\cos\beta \cos L} \tag{3.52}$$

Angle of Incidence and Solar Intensity

The angle of incidence θ (Fig. 3.7*a*) is the angle between the sun's rays radiating on a surface and the line normal to this surface. For a horizontal surface, the angle of incidence θ_H is *ROV*; for a vertical surface, the angle of incidence θ_V is *ROP*; and for a tilted surface, the angle of incidence θ_Σ is *ROU*. Here, Σ is the angle between the tilted surface *ABCD* and the horizontal surface.

Let I_{DN} be the intensity of direct normal radiation, or solar intensity, on a surface normal to the sun's ray, in Btu/h·ft^2 (W/m^2). In Fig. 3.7*b*, I_{DN} is resolved into the vertical component $RQ = I_{DN}$

sin β and the horizontal component $OQ = I_{DN} \cos\beta$. In Fig. 3.7*c*, for the right triangle OPQ, angle $POQ = \gamma$, and hence $OP = I_{DN} \cos\beta \cos\gamma$. In Fig. 3.7*d*, OPL is a right triangle. From point P, a line PT can be drawn perpendicular to the line normal OU, and hence, two right triangles are formed: OTP and LTP. In OTP, because angle $OPT = \Sigma$, the horizontal component of I_{DN} along the line normal of the tilting surface OU is $OT = I_{DN} \cos\beta \cos\gamma \sin\Sigma$. In Fig. 3.7*e*, draw line $O'U'$ parallel to line OU. Again, line QM can be drawn from point Q perpendicular to $O'U'$. Then the right triangles PTL and QMR are similar. Angles PLT and QRM are both equal to Σ. In the right triangle QMR, the component of RQ that is parallel to the line normal to the tilting surface is $MR = I_{DN} \sin\beta \cos\Sigma$.

The intensity of solar rays normal to a tilted surface I_Σ, Btu/h·ft^2 (W/m^2) is the vector sum of the components of the line normal to the tilted surface, or

$$\begin{aligned} I_\Sigma &= I_{DN} \cos\theta_\Sigma \\ &= I_{DN} (\cos\beta \cos\gamma \sin\Sigma + \sin\beta \cos\Sigma \end{aligned} \tag{3.53}$$

3.8 SOLAR RADIATION

Solar radiation provides most of the energy required for the earth's occupants, either directly or indirectly. It is the source of indoor daylight and helps to maintain a suitable indoor temperature during the cold seasons. At the same time, its influence on the indoor environment must be reduced and controlled during hot weather. The sun is located at a mean distance of 92,900,000 mi, (149,500,000 km) from the earth, and it has a surface temperature of about 10,800°F (6000°C). It emits electromagnetic waves at wavelengths of 0.29 to 3.5 μm (micrometers). Visible light has wavelengths of 0.4 to 0.7 μm and is responsible for 38 percent of the total energy striking the earth. The infrared region contains 53 percent. At the outer edge of the atmosphere at a mean earth-sun distance, the solar intensity, called the *solar constant* I_{sc}, is 434.6 Btu/h·ft^2 (1371 W/m^2). The extraterrestrial intensity I_o, Btu/h·ft^2 (W/m^2) varies as the earth-sun distance changes during the earth's orbit.

Based on the data from Miller et al. (1983), the breakdown of solar radiation reaching the earth's surface and absorbed by the earth is listed in Table 3.5. As listed in Table 3.5, only 50 percent of the solar radiation that reaches the outer edge of the earth's atmosphere is absorbed by the clouds and the earth's surface. At any specific location, the absorption, reflection, and scattering of solar radiation depend on the composition of the atmosphere and the path length of the sun's rays through the atmosphere, expressed in terms of the air mass m. When the sun is directly overhead, $m = 1$.

TABLE 3.5 Components of Solar Radiation That Traverse the Earth's Atmosphere

Components		Breakdowns	
Scattered by air	11%	Reflected to space	6%
		Scattered to earth	5
Absorbed by water vapor, dust, etc.	16		
Intercepted by clouds	45	Reflected to space	20
		Absorbed by clouds	4
		Diffused through clouds and absorbed by earth	21
Traversed through air	28	Absorbed by earth	24
		Reflected by earth	4

In Table 3.5, the part of solar radiation that gets through the atmosphere and reaches the earth's surface, in a direction that varies with solar angles over time, is called *direct radiation*. The part that is diffused by air molecules and dust, arriving at the earth's surface in all directions, is called *diffuse radiation*. The magnitude of solar radiation depends on the composition of the atmosphere, especially on the cloudiness of the sky. Therefore, different models are used to calculate the solar radiation reaching the surface of building envelopes.

Solar Radiation for a Clear Sky

ASHRAE recommends use of the following relationships to calculate the solar radiation for a clear sky. The solar intensity of direct normal radiation I_{DN}, Btu/h·ft^2 (W/m^2), can be calculated as

$$I_{DN} = \frac{AC_n}{\exp\,(B/\sin\,\beta)} \tag{3.54}$$

where A = apparent solar radiation when air mass $m = 0$ (its magnitudes are listed in Table 3.6), Btu/h·ft (W/m^2)
B = atmospheric extinction coefficient (Table 3.6), which depends mainly on the amount of water vapor contained in the atmosphere

In Eq. (3.54), the term 1/sin β in exponential form denotes the length of the direct radiation path through the atmosphere. The term C_n is the clearness number of the sky; C_n takes into account the dryness of the atmosphere and the dust contained in the air at a geographic location. Estimated C_n values for nonindustrial locations in the United States are shown in Fig. 3.8.

From Eq. (3.53), the direct radiation radiated onto a horizontal surface through a clear sky I_{DH}, Btu/h·ft^2 (W/m^2), is the vertical component of I_{DN}, that is,

$$I_{DH} = I_{DN} \cos\theta_H = I_{DN} \sin\beta \tag{3.55}$$

and the direct radiation irradiated onto a vertical surface for clear sky I_{DV}, Btu/h·ft^2 (W/m^2), is the horizontal component of I_{DN}, or

$$I_{DV} = I_{DN} \cos\theta_V = I_{DN} \cos\beta \cos\gamma \tag{3.56}$$

TABLE 3.6 Extraterrestrial Solar Radiation and Related Data for 21st Day of Each Month, Base Year 1964

				A	*B*	*C*
	I_0, Btu/h·ft^2	Equation of time, min	Declination, deg	Btu/h·ft^2	(Dimensionless ratios)	
January	448.8	− 11.2	− 20.0	390	0.142	0.058
Febuary	444.2	− 13.9	− 10.8	385	0.144	0.060
March	437.7	− 7.5	0.0	376	0.156	0.071
April	429.9	+1.1	+ 11.6	360	0.180	0.097
May	423.6	+3.3	+ 20.0	350	0.196	0.121
June	420.2	− 1.4	+ 23.45	345	0.205	0.134
July	420.3	− 6.2	+ 20.6	344	0.207	0.136
August	424.1	− 2.4	+ 12.3	351	0.201	0.122
September	430.7	+7.5	0.0	365	0.177	0.092
October	437.3	+ 15.4	− 10.5	378	0.160	0.073
November	445.3	+ 13.8	− 19.8	387	0.149	0.063
December	449.1	+1.6	− 23.45	391	0.142	0.057

A: apparent solar radiation; *B*: atmospheric extinction coefficient; *C*: diffuse radiation factor.
***Source*:** *ASHRAE Handbook 1997, Fundamentals*. Reprinted with permission.

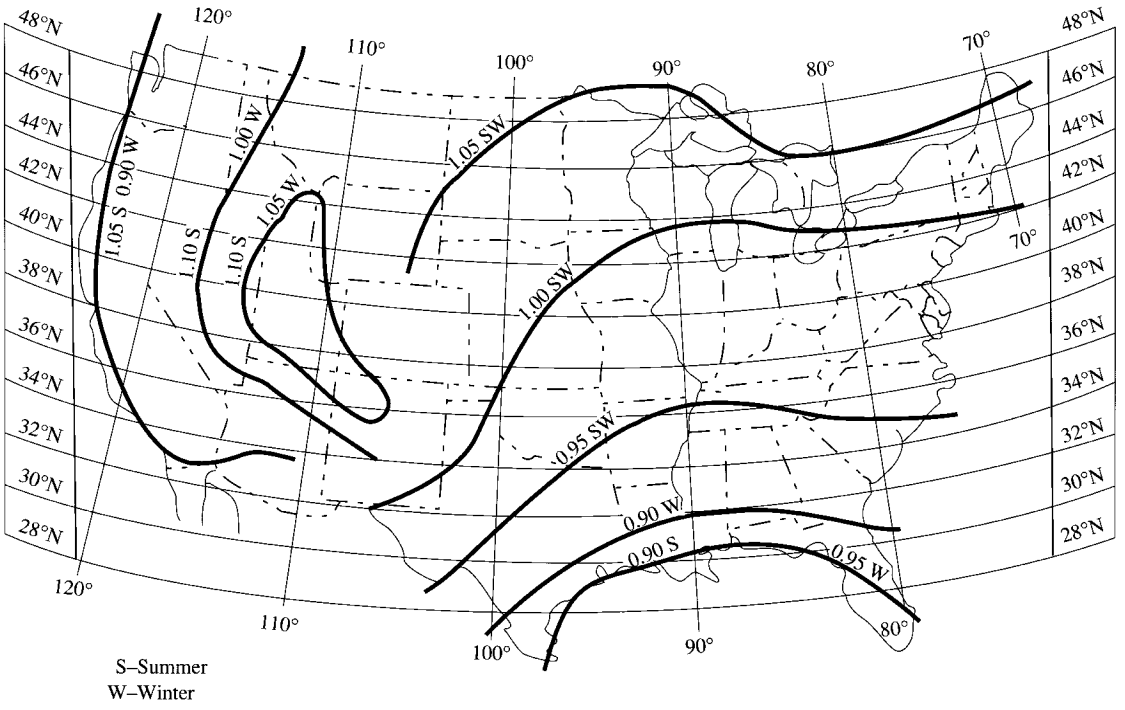

FIGURE 3.8 Estimated atmospheric clearness numbers in the United States for nonindustrial localities.

From Eq. (3.53), for the direct radiation radiated onto a tilted surface at an angle Σ with the horizontal plane through clear sky I_D, Btu/h·ft^2 (W/m^2), can be evaluated as

$$I_D = I_\Sigma = I_{DN} \cos \theta_\Sigma = I_{DN} (\cos\beta \cos\gamma \sin\Sigma + \sin\beta \cos\Sigma) \tag{3.57}$$

Most of the ultraviolet solar radiation is absorbed by the ozone layer in the upper atmosphere. Direct solar radiation, through an air mass $m = 2$, arrives on the earth's surface at sea level during a clear day with a spectrum of < 3 percent in the ultraviolet, 44 percent in the visible, and 53 percent in the infrared.

The diffuse radiation I_d, Btu/h·ft^2 (W/m^2), is proportional to I_{DN} on cloudless days and can be approximately calculated as

$$I_d = \frac{C I_{DN} F_{ss}}{C_n^2} \tag{3.58}$$

where C = diffuse radiation factor, as listed in Table 3.6, and C_n = clearness number of sky from Fig. 3.8. In Eq. (3.58), F_{ss} indicates the shape factor between the surface and the sky, or the fraction of shortwave radiation transmitted through the sky that reaches the surface. For a vertical surface $F_{ss} = 0.5$, for a horizontal surface $F_{ss} = 1$, and for any tilted surface with an angle Σ

$$F_{ss} = \frac{1.0 + \cos\Sigma}{2} \tag{3.59}$$

The total or global radiation on a horizontal plane I_G, Btu/h·ft^2 (W/m^2), recorded by the U.S.

National Climatic Data Center (NCDC), can be calculated as

$$I_G = I_D + I_d = I_{DN}\left(\sin\beta + \frac{C}{C_n^2}\right) \tag{3.60}$$

The reflection of solar radiation from any surface I_{ref}, Btu/h·ft^2 (W/m^2), is given as

$$I_{\text{ref}} = \rho_s F_{sr}(I_D + I_d) \tag{3.61}$$

where ρ_s = reflectance of the surface and F_{sr} = shape factor between the receiving surface and the reflecting surface. The ground-reflected diffuse radiation falling on any surface I_{sg}, Btu/h·ft^2 (W/m^2), can be expressed as

$$I_{sg} = \rho_g F_{sg} I_G \tag{3.62}$$

where F_{sg} = shape factor between the surface and the ground and ρ_g = reflectance of the ground. For concrete, $\rho_g = 0.23$, and for bitumen and gravel, $\rho_g = 0.14$. A mean reflectance $\rho_g = 0.2$ is usually used for ground.

The total intensity of solar radiation I_t, Btu/h·ft^2 (W/m^2), falling on a surface at a direction normal to the surface on clear days, is given by

$$\begin{aligned} I_t &= I_D + I_d + I_{\text{ref}.S} \\ &= (I_{DN} + I_{\text{ref}.DN})\cos\theta + I_d \end{aligned} \tag{3.63}$$

where I = component of reflected solar intensity in direction normal to surface, Btu/h·ft^2 (W/m^2)
$I_{ref.DN}$ = component of reflected solar intensity in direction of sun ray, Btu/h·ft^2 (W/m^2)

Solar Radiation for a Cloudy Sky

For cloudy skies, the global horizontal irradiation I_G^*, Btu/h·ft^2 (W/m^2), usually can be obtained from the NCDC. If it is not available, then it can be predicted from the following relationship:

$$I_G^* = \left(1 + \frac{C_{cc}Q}{P} + \frac{C_{cc}^2 R}{P}\right) I_G \tag{3.64a}$$

Here C_{cc} indicates the cloud cover, on a scale of 0 to 10, and can be calculated by

$$C_{cc} = C_T - 0.5\sum_{j=1}^{4} C_{\text{cir}.j} \tag{3.64b}$$

where C_T = total cloud amount
$C_{\text{cir}.j}$ = clouds covered by cirriforms, including cirrostratus, cirrocumulus, and cirrus, in $j = 1$ to 4 layers

Both C_T and $C_{\text{cir}.j}$ values can be obtained from the major weather stations. The values of coefficients P, Q, and R, according to Kimura and Stephenson (1969), are listed below:

	P	Q	R
Spring	1.06	0.012	− 0.0084
Summer	0.96	0.033	− 0.0106
Autumn	0.95	0.030	− 0.0108
Winter	1.14	0.003	− 0.0082

The direct radiation for a cloudy sky I_D^*, Btu/h·ft^2 (W/m^2), can be calculated as

$$I_D^* = \frac{I_G^* \sin \beta\,(1 - C_{cc}/10)}{\sin \beta + C/C_n^2} \tag{3.65}$$

The diffuse radiation for a cloudy sky I_d^*, Btu/h·ft^2 (W/m^2), is calculated as

$$I_d^* = I_G^* - I_D^* \tag{3.66}$$

3.9 FENESTRATION

Fenestration is the term used for assemblies containing glass or light-transmitting plastic, including appurtenances such as framing, mullions, dividers, and internal, external, and between-glass shading devices, as shown in Fig. 3.12*a*. The purposes of fenestrations are to (1) provide a view of the outside world, (2) permit entry of daylight, (3) admit solar heat as a heating supplement in winter, (4) act as an emergency exit for single-story buildings, and (5) add to aesthetics.

Solar radiation admitted through a glass or window pane can be an important heat gain for commercial buildings, with greater energy impact in the sun belt. HVAC&R designers are asked to control this solar load while providing the required visibility, daylight, and winter heating benefits as well as fire protection and safety features.

Types of Window Glass (Glazing)

Most window glasses, or glazing, are vitreous silicate consisting of silicon dioxide, sodium oxide, calcium oxide, and sodium carbonate. They can be classified as follows:

- *Clear plate or sheet glass or plastic*. Clear plate glass permits good visibility and transmits more solar radiation than other types.
- *Tinted heat-absorbing glass*. Tinted heat-absorbing glass is fabricated by adding small amounts of selenium, nickel, iron, or tin oxides. These produce colors from pink to green, including gray or bluish green, all of which absorb infrared solar heat and release a portion of this to the outside atmosphere through outer surface convection and radiation. Heat-absorbing glass also reduces visible light transmission.
- *Insulating glass*. Insulating glass consists of two panes—an outer plate and a inner plate—or three panes separated by metal, foam, or rubber spacers around the edges and hermetically sealed in a stainless-steel or aluminum-alloy structure. The dehydrated space between the glass panes usually has a thickness of 0.125 to 0.75 in. (3.2–19 mm) and is filled with air, argon, or other inert gas. Air- or gas-filled space increases the thermal resistance of the fenestration.
- *Reflective coated glass*. Reflective glass has a microscopically thin layer of metallic or ceramic coating on one surface of the glass, usually the inner surface of a single-pane glazing or the outer surface of the inner plate for an insulating glass. For a single pane, the coating is often protected by a layer of transparent polyester. The chromium and other metallic coatings give excellent reflectivity in the infrared regions but reduced transmission of visible light compared to clear plate and heat-absorbing glass. Reflections from buildings with highly reflective glass may blind drivers, or even kill grass in neighboring yards.
- *Low-emissivity (low-E) glass coatings*. Glazing coated with low-emissivity, or low-*E*, films has been in use since 1978. It is widely used in retrofit applications. A low-emissivity film is usually a vacuum-deposited metallic coating, usually aluminum, on a polyester film, at a thickness of about 4×10^{-7} in. (0.01 μm). Because of the fragility of the metal coating, protection by another polyester film against abrasion and chemical corrosion must be provided. Recently, copper and

silver coatings on polyethylene and polypropylene film for protection have been used for better optical transmission. A low-*E* film coating reduces the *U* value about 25 to 30 percent for single panes. When combined with other solar control devices, these films can reduce solar heat gain further.

Optical Properties of Sunlit Glazing

When solar radiation inpinges on the outer surface of a plate of glass with an intensity of I at an angle of incidence θ, as shown in Fig. 3.9, a portion of the solar radiation is transmitted, another portion is reflected from both inner and outer surfaces, and the remaining portion is absorbed.

Let r indicate the portion reflected and a the portion absorbed. Also let θ be the angle of incidence. In Fig. 3.9, it can be seen that the portion of the solar radiation transmitted through the glass is actually the sum of the transmittals after successive multiple reflections from the outer and inner surfaces. The decimal portion of I transmitted through the glass, or τ, represents the *transmittance* of the window glass. Similarly, the portion of solar radiation reflected from the window glass is the sum of the successive reflections from the outer surfaces after multiple reflections and absorptions,

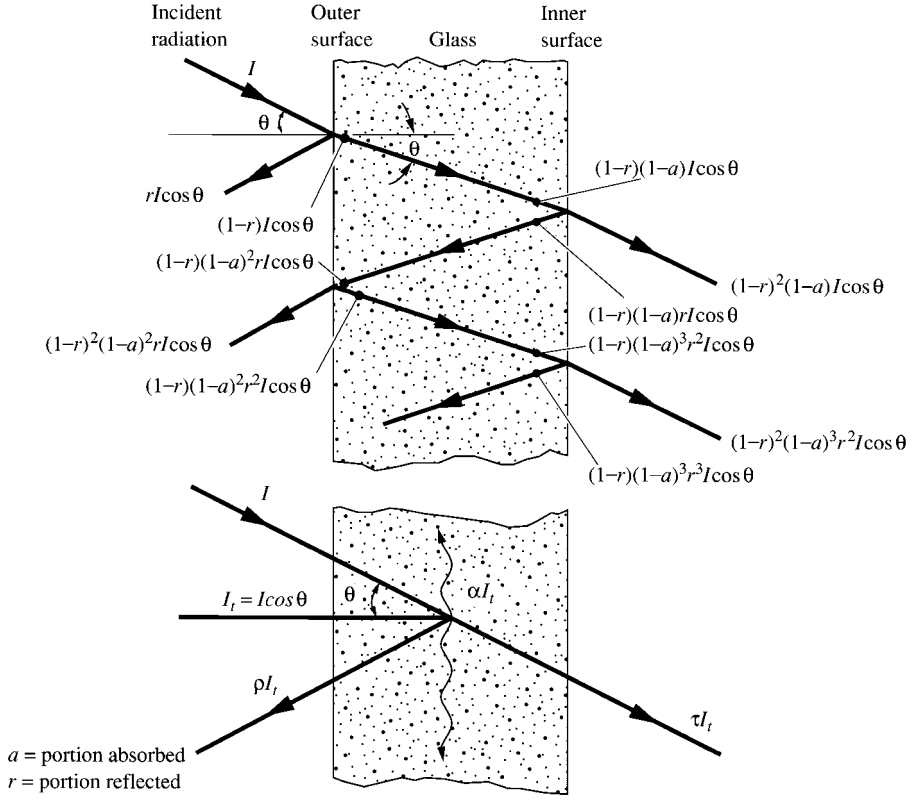

FIGURE 3.9 Simplified representation of multiple transmissions, reflections, and absorptions of solar radiation at glass surfaces.

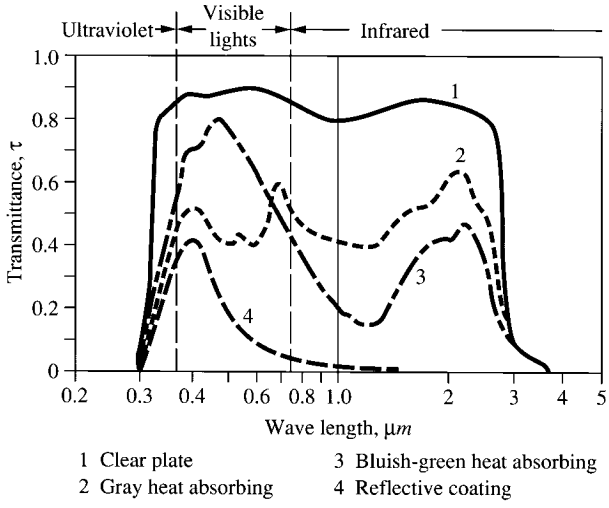

FIGURE 3.10 Spectral transmittance for various types of window glasses. (*Source: ASHRAE Handbook 1989, Fundamentals. Reprinted by permission.*)

and it is identified as ρ, the *reflectance*. The portion absorbed is the sum of the successive absorptions within the glass α, or its *absorptance*.

Figure 3.10 illustrates the spectral transmittance of several types of glazing. All are transparent for shortwave solar radiation at a wavelength between 0.29 and 3 μm and are opaque to longwave radiation in the infrared range with a wavelength greater than 3 μm. Most interior furnishings, equipment, and appliances have an outer surface temperature lower than 250°F (120°C), emitting almost all longwave radiation. At such temperatures, glass is opaque to longwave radiation emitted from inside surfaces and lets only shortwave radiation through. This trapping of longwave radiation indoors is called the greenhouse effect.

In Fig. 3.10 one can also see that clear plate glass has a high transmittance $\tau = 0.87$ for visible light, and that $\tau = 0.8$ for infrared from 0.7 to 3 μm. Heat-absorbing glass has a lower τ and higher absorptance α for both visible light and infrared radiation. Bluish-green heat-absorbing glass has a higher τ in the visible light range and a lower τ in the infrared range than gray heat-absorbing glass. Some reflective glazing has a high reflectance ρ and a significantly lower τ in the visible light range and is opaque to radiation at wavelengths greater than 2 μm. Such characteristics for heat-absorbing and reflective glasses are effective for reducing the amount of solar heat entering the conditioned space during cooling as well as heating seasons. This fact sometimes presents a dilemma for the designer, who must finally compromise to get the optimum combination of conflicting properties.

Another important property of glass is that both τ and α decrease and ρ increases sharply as the incident angle θ increases from 60 to 90°. At 90°, τ and α are 0 and ρ is equal to 1. That is why the solar radiation transmitted through vertical glass declines sharply at noon during summer with solar altitudes $\beta > 70°$.

For all types of plate glass, the sum of these radiation components is always equal to 1, that is,

$$\tau + \alpha + \rho = 1$$

and

$$I = \tau I + \alpha I + \rho I \tag{3.67}$$

3.10 HEAT ADMITTED THROUGH WINDOWS

For external glazing without shading, the heat gain admitted into the conditioned space through each square foot of sunlit area A_s of window Q_{wi}/A_s, Btu/h · ft^2 (W/m^2), can be calculated as follows:

$$\begin{array}{c}\text{Heat gain through each ft}^2\\ \text{of sunlit window}\end{array} = \begin{array}{c}\text{solar radiation transmitted}\\ \text{through window glass}\end{array} + \begin{array}{c}\text{inward heat flow from glass}\\ \text{inner surface into conditioned space}\end{array}$$

That is,

$$\frac{Q_{wi}}{A_s} = \frac{\tau I_t + Q_{RCi}}{A_s} \tag{3.68}$$

where Q_{RCi} = inward heat flow from the inner surface of an unshaded sunlit window, Btu/h (W).

Heat Gain for Single Glazing

For an external, sunlit single-glazed window without shading, the inward heat flow from the inner surface of the glass, as shown in Fig. 3.11, can be evaluated as

$$Q_{RCi} = \text{inward absorbed radiation} + \text{conductive heat transfer}$$

$$= UA_s\left(\frac{\alpha I_t}{h_o} + T_o - T_i\right) \tag{3.69}$$

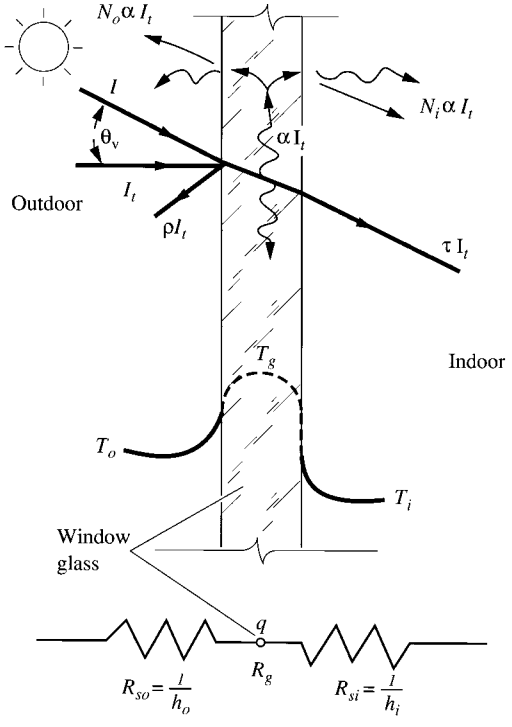

FIGURE 3.11 Heat admitted through a single-glazing window glass.

where h_o = heat-transfer coefficient for outdoor surface of window glass, Btu/h·ft²·°F (W/m²·°C)
T_o = outdoor air temperature, °F (°C)
T_i = indoor air temperature,°F (°C)

Heat admitted through a unit area of the single-glazing window glass is

$$\frac{Q_{wi}}{A_s} = \tau I_t + U\left(\frac{\alpha I_t}{h_o} + T_o - T_i\right)$$
$$= \text{SHGC} I_t + U(T_o - T_i) \tag{3.70}$$

Solar heat gain coefficient (SHGC) is the ratio of solar heat gain entering the space through the window glass to the incident solar radiation, total shortwave irradiance for a single-glazed window is given as

$$\text{SHGC} = Q_{ws}/I_t A_s$$
$$= \tau + \frac{U\alpha}{h_o} \tag{3.71}$$

In these last three equations, U indicates the overall heat-transfer coefficient of the window, in Btu/h·ft²·°F (W/m²·°C), and can be calculated as

$$U = \frac{U_{wg}A_{wg} + U_{eg}A_{eg} + U_f A_f}{A_{wg} + A_{eg} + A_f} \tag{3.72}$$

In Eq. (3.72), A represents area, in square feet (square meters); subscript *wg* indicates glass, and *eg* signifies the edge of the glass including the sealer and spacer of the insulating glass. The edge of glass has a width of about 2.5 in. (64 mm). The subscript *f* means the frame of the window. Some U values for various types of windows at winter design conditions are listed in Table 3.7. For summer

TABLE 3.7 Overall Heat-Transfer Coefficient U Values for Windows at Winter Conditions* with Commercial Type of Frame, Btu/h·ft²·°F)

Type	Gas between glasses	Space between glasses, in.	Emittance† of low-*E* film	Glass	Edge	Aluminum frame of $U_f = 1.9$	Aluminum frame with thermal break, with $U_f = 1.0$	Wood or vinyl frame of $Uf = 0.41$
			Overall coefficient Btu/h·ft²·°F					
Single glass				1.11		1.23	1.10	0.98
Double glass	Air	3/8		0.52	0.62	0.74	0.60	0.51
Double glass	Air	3/8	0.40	0.43	0.55	0.67	0.54	0.45
Double glass	Air	3/8	0.15	0.36	0.51	0.62	0.48	0.39
Double glass	Argon	3/8	0.15	0.30	0.48	0.57	0.43	0.34
Triple glass	Air	3/8		0.34	0.50	0.60	0.46	0.38
Triple glass	Air	3/8	0.40	0.30	0.48	0.57	0.44	0.35
Triple or double glass with polyester film suspended in between	Argon	3/8	0.15	0.17	0.43	0.47	0.34	0.25

* *Winter conditions* means 70°F indoor, 0°F outdoor temperature and a wind speed of 15 mph.
† Low-*E* film can be applied to surface 2 for double glass (see Fig. 3.12*b*) or surface 2, 3, 4, or 5 for triple glass (any surface other than outer and inner surfaces).
***Source*:** Abridged with permission from *ASHRAE Handbook, 1989, Fundamentals*.

design conditions at 7.5 mph (3.3 m/s) wind speed, the listed U values should be multiplied by a factor of 0.92. The U values of windows depend on the construction of the window, the emissivity of the surfaces of glass or plastic sheets, and the air velocity flowing over the outdoor and indoor surfaces. Q_{ws} indicates the solar heat gain entering the space, in Btu/h (W).

As shown in Fig. 3.11, solar radiation having a total intensity I, Btu/h·ft^2 (W/m^2), when it is radiated on the outer surface of a vertical pane with an angle of incidence θ_v, the line normal indicating total shortwave irradiance I_t, in Btu/h·ft^2 (W/m^2) actually consists of

$$I = I_{DN} + I_d + I_{\text{ref. }DN}$$
$$I_t = I\cos\theta_v \tag{3.73}$$

Heat Gain for Double Glazing

For a double-glazed window, the inward heat flow per square foot of the inner surface of the glass, as shown in Fig. 3.12*b*, is

$$\frac{Q_{RCi}}{A_s} = N_{io}\alpha I_o + N_{ii}\alpha I_i + U(T_o - T_i)$$
$$= U\left[\frac{\alpha I_o}{h_o} + \left(\frac{1}{h_o} + \frac{1}{h_a}\right)\alpha I_i + T_o - T_i\right] \tag{3.74}$$

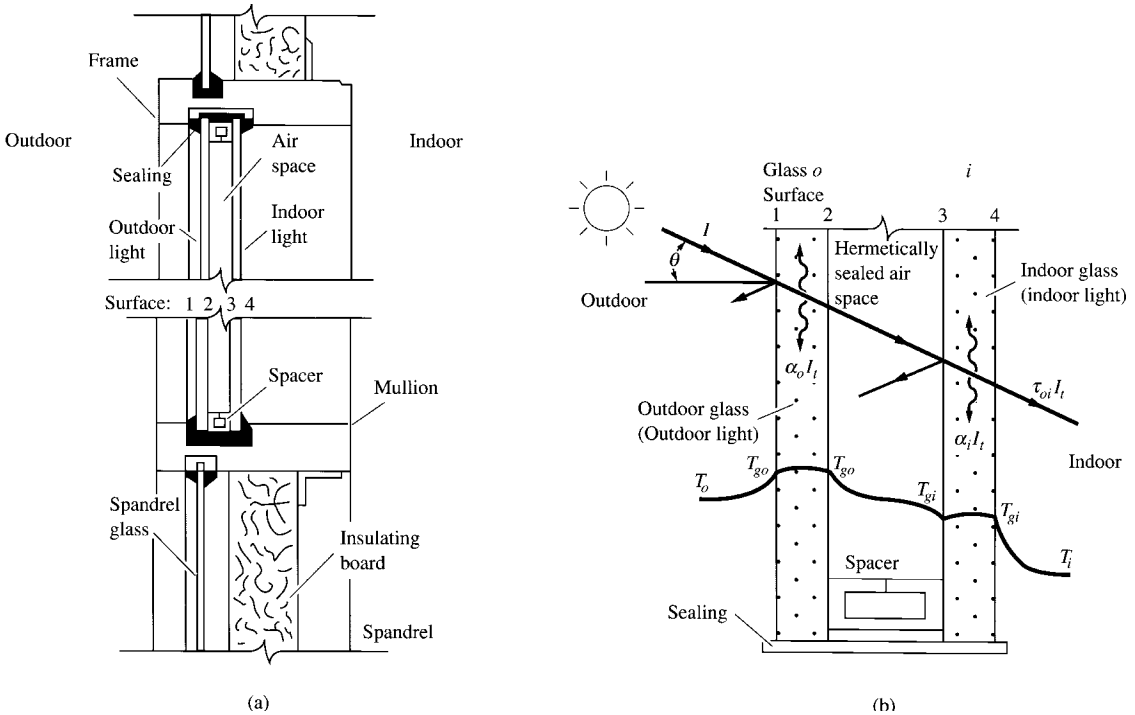

FIGURE 3.12 Heat flow through an insulating glass (double-pane) window. (*a*) Construction of a typical insulating glass; (*b*) heat flow and temperature profiles.

where N_{io} = inward fraction of solar radiation absorbed by outdoor glass
N_{ii} = inward fraction of solar radiation absorbed by indoor glass
I_o, I_i = solar intensity irradiated on outdoor and indoor glass, Btu/h·ft^2 (W/m^2)

The solar radiation absorbed by the outdoor and indoor panes, in Btu/h·ft^2 (W/m^2), is

$$\alpha I_o = \alpha_o I_t \qquad \alpha I_i = \alpha_i I_t \tag{3.75}$$

Absorptance of the outdoor panes α_o is

$$\alpha_o = \alpha_1 + \frac{\alpha_2 \tau_{oi} \rho_3}{1 - \rho_2 \rho_3} \tag{3.76}$$

Absorptance of the indoor pane is

$$\alpha_i = \frac{\alpha_3 \tau_o}{1 - \rho_2 \rho_3} \tag{3.77}$$

Subscript o indicates outdoor; i, indoor; and 1, 2, 3, and 4, the surfaces of the panes as shown in Fig. 3.12*b*.

The airspace heat-transfer coefficient h_a, Btu/h·ft^2·°F (W/m^2·°C), is the reciprocal of the R value of the airspace R_a listed in Table 3.4, that is,

$$h_a = \frac{1}{R_a} \tag{3.78}$$

The transmittance of solar radiation through both outdoor and indoor panes τ_{oi} can be calculated as

$$\tau_{oi} = \frac{\tau_o \tau_i}{1 - \rho_2 \rho_3} \tag{3.79}$$

Then the heat admitted per square foot (square meter) through a double-glass window Q_{woi}/A_s Btu/h·ft^2 (W/m^2), can be calculated as

$$\begin{aligned} \frac{Q_{woi}}{A_s} &= \tau_{oi} I_t + U\left(\frac{\alpha_o}{h_o} + \frac{\alpha_i}{h_o} + \frac{\alpha_i}{h_a}\right) + I_t + U(T_o - T_i) \\ &= \text{SHGC}_{oi} I_t + U(T_o - T_i) \end{aligned} \tag{3.80}$$

The SHGC_{oi} for a double-glass window can be calculated as

$$\text{SHGC}_{oi} = \tau_{oi} + U\left(\frac{\alpha_o}{h_o} + \frac{\alpha_i}{h_a}\right) \tag{3.81}$$

Because a glass plate is usually between 0.125 and 0.25 in. (3 and 6 mm) thick, with a thermal conductivity k of about 0.5 Btu/h·ft·°F (W/m·°C), there is only a small temperature difference between the inner and outer surfaces of the plate when solar radiation is absorbed. For the sake of simplicity, it is assumed that the temperature of the plate T_g is the same in the direction of heat flow.

For a double-glazed window, the glass temperature of the outdoor pane T_{go}, °F (°C), can be calculated as

$$T_{go} = T_o + \left(\alpha I_o + \alpha I_i - \frac{Q_{RCi}}{A_s}\right)\left(\frac{1}{h_o} + \frac{R_{go}}{2}\right) \tag{3.82}$$

The temperature of the indoor pane T_{gi}, °F (°C), is calculated as

$$T_{gi} = T_r + Q_{RCi}\left(\frac{1}{h_i} + \frac{R_{gi}}{2}\right) \tag{3.83}$$

where R_{go}, R_{gi} = R values of outdoor and indoor panes, $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$)
h_i = heat-transfer coefficient of inside surface 4 (see Fig. 3.12*b*) of inside plate, $Btu/h \cdot ft^2 \cdot °F$ ($W/m^2 \cdot °C$)

Shading Coefficients

The *shading coefficient* is defined as the ratio of solar heat gain of a glazing assembly of specific construction and shading devices at a summer design solar intensity and outdoor and indoor temperatures, to the solar heat gain of a reference glass at the same solar intensity and outdoor and indoor temperatures. The reference glass is double-strength sheet glass (DSA) with transmittance $\tau = 0.86$, reflectance $\rho = 0.08$, absorptance $\alpha = 0.06$, and $F_{\text{DSA}} = 0.87$ under summer design conditions. The shading coefficient SC is an indication of the characteristics of a glazing and the associated shading devices, and it can be expressed as

$$\text{SC} = \frac{\text{solar heat gain of specific type of window glass}}{\text{solar heat gain of double-strength sheet glass}}$$

$$= \frac{\text{SHGC}_w}{\text{SHGC}_{\text{DSA}}} = \frac{\text{SHGC}_w}{0.87} = 1.15\ \text{SHGC}_{wi} \tag{3.84}$$

where SHGC_w = solar heat gain coefficient of specific type of window glass
SHGC_{DSA} = solar heat gain coefficient of standard reference double-strength sheet glass

Shading coefficients of various types of glazing and shading devices are presented in Table 3.8.

TABLE 3.8 Shading Coefficients for Window Glass with Indoor Shading Devices

Type of glass	Thickness of glass, in.	Solar transmittance	Glass	Venetian blinds Med.*	Venetian blinds Light†	Roller shade Opaque, white	Roller shade Translucent	Draperies Med.‡	Draperies Light§
Clear	$\frac{3}{32}$	0.87 to 0.79		0.74	0.67	0.39	0.44	0.62	0.52
Heat-absorbing	$\frac{3}{16}$ or $\frac{1}{4}$	0.46		0.57	0.53	0.30	0.36	0.46	0.44
	$\frac{3}{8}$	0.34		0.54	0.52	0.28	0.32		
		0.24		0.42	0.40	0.28	0.31	0.38	0.36
Reflective-coated			0.30	0.25	0.23				
			0.40	0.33	0.29				
			0.50	0.42	0.38				
Insulating glass		Outer Inner							
Clear out Clear in	$\frac{3}{32}$, or $\frac{1}{8}$	0.87 0.87		0.62	0.58	0.35	0.40		
Heat-absorbing out Clear in	$\frac{1}{4}$	0.46 0.80		0.39	0.36	0.22	0.30		
Reflective glass			0.20	0.19	0.18				
			0.30	0.27	0.26				
			0.40	0.34	0.33				

*Med. indicates medium color.
†Light indicates light color.
‡Draperies Med. represents draperies of medium color with a fabric openness of 0.10 to 0.25 and yarn reflectance of 0.25 to 0.50.
§Draperies Light repesents draperies of light color with a fabric openness below 0.10 and yarn reflectance over 0.50.
Source: Adapated with permission from *ASHRAE Handbook 1989, Fundamentals*.

Solar Heat Gain Factors and Total Shortwave Irradiance

The solar heat gain factor (SHGF), Btu/h·ft^2 (W/m^2), is designated as the average solar heat gain during cloudless days through DSA glass. In the *ASHRAE Handbook 1993, Fundamentals* are tabulated SHGF value for various latitudes, solar times, and orientations for load and energy calculations. For calculating the summer cooling peak load, the concept of maximum SHGF has been introduced. This is the maximum value of SHGF on the 21st of each month for a specific latitude, as listed in Table 3.9. For high elevations and on very clear days, the actual SHGF may be 15 percent higher than the value, listed in Table 3.9. In dusty industrial areas or at very humid locations, the actual SHGF may be lower.

According to *ASHRAE Handbook 1997*, *Fundamentals*, Gueymard (1995) provides a comprehensive model for calculating the spectral and broadband total shortwave irradiance I_t, in Btu/h·ft^2 (W/m^2) for cloudless sky conditions and allows the input of the concentrations of a variety atmospheric constituents.

Example 6.1. A double-glass window of a commercial building facing west consists of an outdoor clear plate glass of 0.125-in. and an indoor reflective glass of 0.25-in. thickness with a reflective film on the outer surface of the indoor glass, as shown in Fig. 3.12*b*. This building is located at

TABLE 3.9 Maximum Solar Heat Gain Factors (Max SHGF)

	Max SHGF, Btu/h·ft^2									
	N (shade)	NNE/NNW	NE/NW	ENE/WNW	E/W	ESE/WSW	SE/SW	SSE/SSW	S	HOR*
	North latitude, 40°									
Jan.	20	20	20	74	154	205	241	252	254	133
Feb.	24	24	50	129	186	234	246	244	241	180
Mar.	29	29	93	169	218	238	236	216	206	223
Apr.	34	71	140	190	224	223	203	170	154	252
May	37	102	165	202	220	208	175	133	113	265
June	48	113	172	205	216	199	161	116	95	267
July	38	102	163	198	216	203	170	129	109	262
Aug.	35	71	135	185	216	214	196	165	149	247
Sept.	30	30	87	160	203	227	226	209	200	215
Oct.	25	25	49	123	180	225	238	236	234	177
Nov.	20	20	20	73	151	201	237	248	250	132
Dec.	18	18	18	60	135	188	232	249	253	113
	North latitude, 32°									
Jan.	24	24	29	105	175	229	249	250	246	176
Feb.	27	27	65	149	205	242	248	232	221	217
Mar.	32	37	107	183	227	237	227	195	176	252
Apr.	36	80	146	200	227	219	187	141	115	271
May	38	111	170	208	220	199	155	99	74	277
June	44	122	176	208	214	189	139	83	60	276
July	40	111	167	204	215	194	150	96	72	273
Aug.	37	79	141	195	219	210	181	136	111	265
Sept.	33	35	103	173	215	227	218	189	171	244
Oct.	28	28	63	143	195	234	239	225	215	213
Nov.	24	24	29	103	173	225	245	246	243	175
Dec.	22	22	22	84	162	218	246	252	252	158

*Horizontal surface

***Source*:** Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

40° north latitude. The detailed optical properties of their surfaces are as follows:

$\tau = 0.80$	$\rho_1 = 0.08$	$\alpha_1 = 0.12$	$e_1 = 0.84$
	$\rho_2 = 0.08$	$\alpha_2 = 0.12$	$e_2 = 0.84$
$\tau_i = 0.16$	$\rho_3 = 0.68$	$\alpha_3 = 0.16$	$e_3 = 0.15$
	$\rho_4 = 0.08$	$\alpha_4 = 0.76$	$e_4 = 0.84$

The R value of the 0.25-in. (6-mm) thickness indoor glass is $R_g = 0.035\ \text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}$ (0.0063 $\text{m}^2\cdot{}^\circ\text{C/W}$), and that of the enclosed airspace is $R_a = 1.75\ \text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}$ (0.32 $\text{m}^2\cdot{}^\circ\text{C/W}$).

At 4 P.M. on July 21, the outdoor temperature at this location is 100°F (37.8°C), the indoor temperature is 76°F (24.4°C), and the total solar intensity at a direction normal to this west window is 248 $\text{Btu/h}\cdot\text{ft}^2$ (782 W/m^2). The outdoor surface heat-transfer coefficient $h_o = 4.44\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ (25.2 $\text{W/m}^2\cdot{}^\circ\text{C}$), and the heat-transfer coefficient of the inner surface of the indoor glass $h_i = 2.21\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$ (12.5 $\text{W/m}^2\cdot{}^\circ\text{C}$). Calculate the following:

1. The inward heat flow of this window
2. The temperatures of the outdoor and indoor glasses
3. The shading coefficient of this double-glass window
4. The total heat gain admitted through this window

Solution

1. From Eq. (3.76), the absorption coefficient of the outdoor glass can be calculated as

$$\alpha_o = \alpha_1 + \frac{\alpha_2\tau_o\rho_3}{1 - \rho_2\rho_3}$$

$$= 0.12 + \frac{0.12 \times 0.8 \times 0.68}{1 - 0.08 \times 0.68} = 0.189$$

And from Eq. (3.77), the absorption coefficient of the indoor glass is

$$\alpha_i = \frac{\alpha_3\tau_o}{1 - \rho_2\rho_3} = \frac{0.16 \times 0.8}{1 - 0.08 \times 0.68} = 0.135$$

From Eq. (3.75), the heat absorbed by the outdoor glass is

$$\alpha I_o = \alpha_o I_t = 0.189 \times 248 = 46.9\ \text{Btu/h}\cdot\text{ft}^2$$

Also, the heat absorbed by the indoor glass is calculated as

$$\alpha I_i = a_i I_t = 0.135 \times 248 = 33.5\ \text{Btu/h}\cdot\text{ft}^2$$

And then, from Eq. (3.22), the overall heat-transfer coefficient of this double-glass window is

$$U = \frac{1}{1/h_i + R_g + R_a + R_g + 1/h_o}$$

$$= \frac{1}{1/2.21 + 0.035 + 1.75 + 0.035 + 1/4.44} = 0.400\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

The inward heat flow from the inner surface of the indoor glass, from Eq. (3.74), is given as

$$\frac{Q_{RCi}}{A_s} = U\left[\frac{\alpha_o I_t}{h_o} + \alpha_i I_t \frac{1}{h_o} + \frac{1}{h_a} + T_o - T_r\right]$$

$$= 0.400\left[\frac{46.9}{4.44} + 33.5\,\frac{1}{4.44} + 1.75 + 100 - 76\right] = 40.29\ \text{Btu/h}\cdot\text{ft}^2\ (127.1\ \text{W/m}^2)$$

2. From Eq. (3.82), the temperature of the outdoor glass is

$$T_{go} = T_o + \left(\alpha_o I_t + \alpha_i I_t - \frac{Q_{RCi}}{A_s}\right)\left(\frac{1}{h_o} + \frac{R_g}{2}\right)$$

$$= 100 + (46.9 + 33.5 - 40.29)\left(\frac{1}{4.44} + \frac{0.035}{2}\right) = 109.7°\text{F}$$

and the temperature of the indoor glass is

$$T_{gi} = T_i + \frac{Q_{RCi}}{A_s}\left(\frac{1}{h_i} + \frac{R_g}{2}\right)$$

$$= 76 + 40.29\left(\frac{1}{2.21} + \frac{0.035}{2}\right) = 94.9°\text{F}\ (34.9°\text{C})$$

3. From Eq. (3.79), the transmittance for both panes is

$$\tau_{oi} = \frac{\tau_o \tau_i}{1 - \rho_2 \rho_3} = \frac{0.8 \times 0.16}{1 - 0.08 \times 0.68} = 0.135$$

and from Eq. (3.81), the solar heat gain coefficient for the double-glass window is

$$\text{SHGC}_{oi} = \alpha_{oi} + \frac{U\alpha_o}{h_o} + \left(\frac{U}{h_o} + \frac{U}{h_a}\right)\alpha_i$$

$$= 0.135 + \frac{0.40 \times 0.189}{4.44} + \left(\frac{0.40}{4.44} + 0.40 \times 1.75\right)0.135 = 0.259$$

Then, from Eq. (3.84), the shading coefficient of this double-glass window is given by

$$\text{SC} = \frac{\text{SHGC}_{oi}}{\text{SHGC}_{\text{DSA}}} = \frac{0.259}{0.87} = 0.297$$

4. From Eq. (3.80), the total heat gain admitted through this double-glass window per square foot of sunlit area is

$$\frac{Q_{wi}}{A_s} = \text{SHGC}_{oi} I_t + U(T_o - T_i) = 0.259 \times 248 + 0.40(100 - 76)$$

$$= 64.2 + 9.6 = 73.8\ \text{Btu/h}\cdot\text{ft}^2\ (233\ \text{W/m}^2)$$

Selection of Glazing

During the selection of glazing, the following factors should be considered: visual communication, use of daylight, thermal comfort, summer and winter solar heat gain, street noise attenuation, safety

and fire protection, and life-cycle cost analysis. Energy conservation considerations, including the control of solar heat with the optimum combination of absorbing and reflective glass and various shading devices, are covered in the next section.

To reduce the heat loss through glass during winter, one can install double or triple glazing, storm windows, or low-emission film coating on the surface of the glass. Elmahdy (1996) and de Abreu et al. (1996) tested the thermal performance of seven insulating glass units. If a clear, double-glazed insulating glass unit with a silicone foam spacer of 0.5 in. (13 mm) between two panes is taken as the base unit, the U values of these seven insulating glass units are as follows:

Unit	Glazing	Airspace, in.	Spacer	U value, Btu/h·ft^2·°F
1	Clear, double-glazed	0.5	Foam	0.51
2	Clear, double-glazed	0.5	Aluminum	0.51
3	Clear, double-glazed	0.25	Foam	0.58
4	Clear, double-glazed	0.75	Foam	0.51
5	Low-*e*, double-glazed	0.5	Foam	0.36
6	Clear, Triple-glazed	0.5	Foam	0.32
7	Clear, Triple-glazed	0.25	Foam	0.39

Low-*e* coating reduces the U value by about 30 percent, and the triple-glazing drops about one-third compared with a clear, double-glazed insulating glass unit. If the width of the airspace is decreased to 0.25 in. (6.5 mm), its U value will increase 15 percent. When the airspace is wider than 0.5 in. (13 mm), regardless of whether a metal or silicone foam spacer is adopted, neither has any significant effect on U value of the insulation glass unit.

3.11 SHADING OF GLASS

Shading projected over the surface of glass significantly reduces its sunlit area. Many shading devices increase the reflectance of the incident radiation. There are two types of shading: deliberately installed shading devices, which include indoor and external shading devices, and shading from adjacent buildings.

Indoor Shading Devices

Indoor shading devices not only provide privacy but also are usually effective in reflecting part of the solar radiation back to the outdoors. They also raise the air temperature of the space between the shading device and the window glass, which in turn reduces the conductive heat gain in summer. Indoor shading devices are easier to operate and to maintain and are more flexible in operation than external shading devices. Three types of indoor shading devices are commonly used: venetian blinds, draperies, and roller shades.

Venetian Blinds. Most horizontal venetian blinds are made of plastic or aluminum slats, spaced 1 to 2 in. (25 to 50 mm) apart, and some are made of rigid woven cloth. The ratio of slat width to slat spacing is generally 1.15 to 1.25. For light-colored metallic or plastic slats at a 45° angle, typical optical properties are $\tau = 0.05$, $\rho = 0.55$, and $\alpha = 0.40$. Vertical venetian blinds with wider slats are widely used in commercial buildings.

Consider a single-glazed window combined with indoor venetian blinds at a slat angle $\Psi = 45°$, as shown in Fig. 3.13. Let the subscripts g, v, and a represent the glass, the venetian blinds, and the

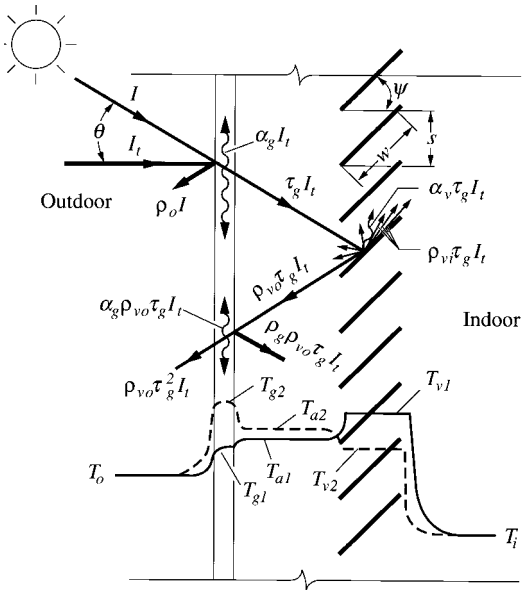

FIGURE 3.13 Heat transfer through a single-glazed window combined with venetian blinds.

air between the venetian blinds and the glass. Also let *o* indicate the outward direction and *i* the inward direction. Of the solar radiation transmitted through the glass and radiating on the surface of the slats $\tau_g I_t$,

- A fraction $\rho_{vo} \tau_g I_t$ is reflected outward from the surface of each slat.
- Another fraction $\rho_{vi} \tau_g I_t$ is reflected from the slat surface into the conditioned space.
- A third fraction absorbed by the slat is either convected away by the space air or reflected from the surface to the indoor surroundings in the form of longwave radiation.

If the glass has a high transmittance, the slat temperature T_{v1} will be higher, as shown by the lower solid temperature curve in Fig. 3.13. If the glass has a high absorptance, its temperature T_{g2} will be higher. In Table 3.8 the shading coefficients of various combinations of venetian blinds and glazing are listed.

According to results of a field survey by Inoue et al. in four office buildings in Japan in 1988, 60 percent of the venetian blinds were not operated during the daytime. The incident angle of the direct solar radiation had greater influence on operation of the blinds than the intensity. Automatic control of slat angle and of raising and lowering the blinds is sometimes used and may become more popular in the future.

Draperies. These are fabrics made of cotton, regenerated cellulose (such as rayon), or synthetic fibers. Usually they are loosely hung, wider than the window, and pleated; and they can be drawn open or closed as required. Drapery-glass combinations reduce the solar heat gain in summer and increase the thermal resistance in winter. Reflectance of the fabric is the dominant factor in the reduction of heat gain, and visibility is a function of the openness of the weave.

Roller Shades. These are sheets made of treated fabric or plastic that can be pulled down to cover the window or rolled up. Roller shades have a lower SC than do venetian blinds and draperies.

When glass is covered with shades, any outdoor visual communication is blocked, and the visible light transmittance is less than that of other indoor shading devices.

External Shading Devices

External shading devices include overhangs, side fins, egg-crate louvers, and pattern grilles, as shown in Fig. 3.14. They are effective in reducing the solar heat gain by decreasing the sunlit area. However, the external shading devices do not always fit into the architectural requirements and are less flexible and more difficult to maintain. Pattern grilles impair visibility significantly.

Figure 3.15 shows the shaded area of a glass pane constructed with both overhang and side fin. The *profile angle* Ω is defined as the angle between a horizontal plane and a tilted plane that includes the sun's rays. We see that $\tan \Omega = UQ/OR = \tan \beta/\cos \gamma$. Let S_W be the width of the shadow and S_H be the height of the shadow projected on the plane of the glass by direct solar radiation, in feet (meters). Also, let W be the width of the glass and H be the height. Then the shadow width on the plane of the glass is

$$S_W = P_V \tan \gamma \tag{3.85}$$

and the shadow height is

$$S_H = P_H \tan \Omega = P_H \frac{\tan \beta}{\cos \gamma} \tag{3.86}$$

where P_V = projection of side fin plus mullion and reveal, ft (m)
P_H = horizontal projection of overhang plus transom and reveal, ft (m)

The projection factor due to the external shading device F_{pro} can be calculated as

$$F_{\text{pro}} = \frac{P_H}{H} \tag{3.87}$$

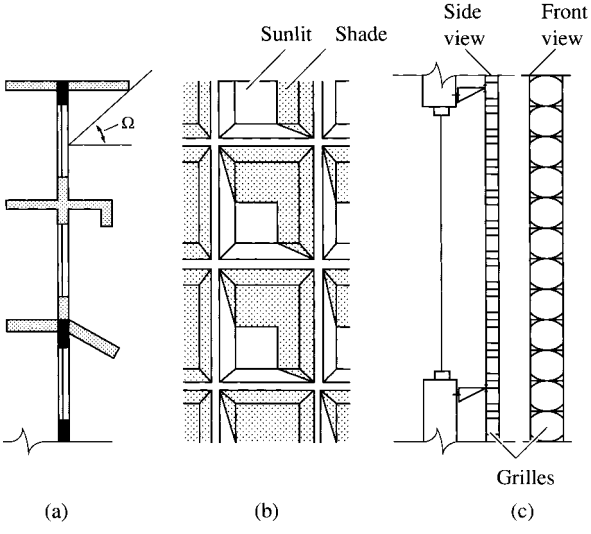

FIGURE 3.14 Types of external shades. (*a*) Overhang; (*b*) egg-crate louver; (*c*) pattern grilles.

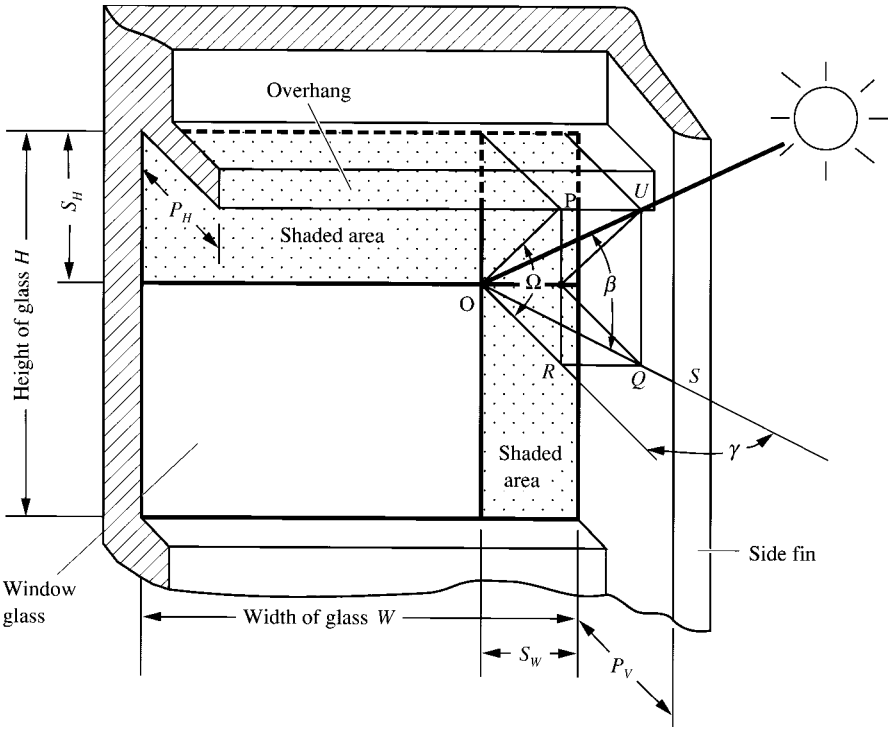

FIGURE 3.15 Shaded area of a window glass constructed with an overhang and side fin.

The net sunlit area of the glass A_s, ft^2 (m^2), can then be calculated as

$$A_s = (W - S_W)(H - S_H) \tag{3.88a}$$

and the shaded area of the glass A_{sh}, ft^2 (m^2), is given by

$$A_{sh} = A_g - A_s = WH - (W - S_W)(H - S_H) \tag{3.88b}$$

where A_g = area of the glass, ft^2 (m^2).

Shading from Adjacent Buildings

Shadows on glass cast by adjacent buildings significantly reduce the sunlit area of the glass. For example, in Fig. 3.16 we see the area on building A shaded because of the presence of building B. Let two sides of the shaded building A coincide with the X and Y axes on the plan view shown in Fig. 3.16. In the elevation view of Fig. 3.16, the shadow height on the facade of building A is

$$S_H = H_B - L_{AB} \tan \Omega \tag{3.89}$$

and the shadow width S_W on the facade of building A is

$$S_W = W_{OB} + W_B - L_{AB} \tan \gamma \tag{3.90}$$

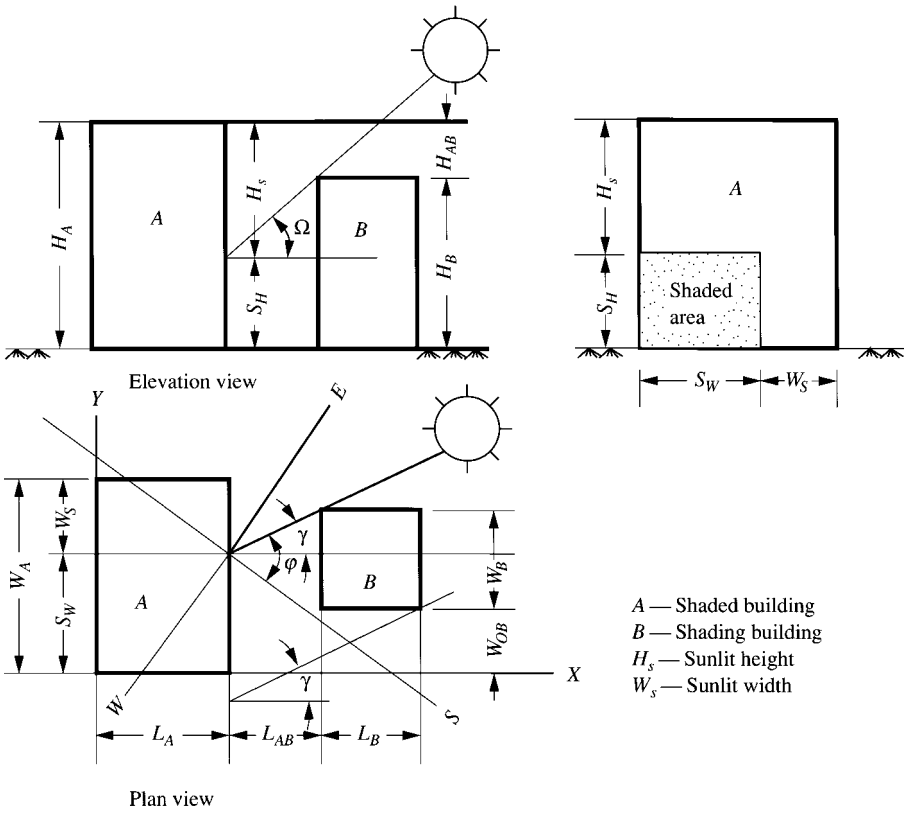

FIGURE 3.16 Shading from adjacent building.

where H_B = height of building B, ft (m)
W_B = width of building B, ft (m)
L_{AB} = distance between two buildings along X axis, ft (m)
W_{OB} = distance between X axis and building B, ft (m)

Subscript A indicates the shaded building and B the shading building.

Because of the sun's varying position, the solar altitude β and the surface solar azimuth γ change their values from time to time. Table 3.10 lists the solar altitude β and solar azimuth γ at north latitudes 32° and 40°. To evaluate the shaded area of the outer surface of a building, a computer program can determine which of several hundred representative points on this outer surface are sunlit or shaded at specific times of the day. Then a ratio of shaded area to total area can be calculated.

At a certain time instant, a window glass area can be shaded due to the effect of the overhang and vertical fins, or the effect of the adjacent building, or both. For the calculation of the combined shading effect of overhangs, vertical fins, and adjacent building, it is recommended that the number of sunlit and shaded windows of an outer surface of a building under the effect of the adjacent building be calculated first. According to Eq. (3.88*a*), the total sunlit area can be calculated as the product of the net sunlit area for each window because of the overhang and vertical fins and the number of windows in the sunlit area of that outer surface.

TABLE 3.10 Solar Altitude (ALT) and Solar Azimuth (AZ)

	North latitude							
	32°				40°			
	Solar time	Solar position		Solar time	Solar time	Solar position		Solar time
Date	AM	ALT	AZ	PM	AM	ALT	AZ	PM
Dec.	8	10	54	4	8	5	53	4
	9	20	44	3	9	14	42	3
	10	28	31	2	10	21	29	2
	11	33	16	1	11	25	15	1
	12	35	0	12	12	27	0	12
Jan.	7	1	65	5	8	8	55	4
+	8	13	56	4	9	17	44	3
Nov.	9	22	46	3	10	24	31	2
	10	31	33	2	11	28	16	1
	11	36	18	1	12	30	0	12
	12	38	0	12				
Feb.	7	7	73	5	7	4	72	5
+	8	18	64	4	8	15	62	4
Oct.	9	29	53	3	9	24	50	3
	10	38	39	2	10	32	35	2
	11	45	21	1	11	37	19	1
	12	47	0	12	12	39	0	12
Mar.	7	13	82	5	7	11	80	5
+	8	25	73	4	8	23	70	4
Sept.	9	37	62	3	9	33	57	3
	10	47	47	2	10	42	42	2
	11	55	27	1	11	48	23	1
	12	58	0	12	12	50	0	12
Apr.	6	6	100	6	6	7	99	6
+	7	19	92	5	7	19	89	5
Aug.	8	31	84	4	8	30	79	4
	9	44	74	3	9	41	67	3
	10	56	60	2	10	51	51	2
	11	65	37	1	11	59	29	1
	12	70	0	12	12	62	0	12
May	6	10	107	6	5	2	115	7
+	7	23	100	5	6	13	106	6
July	8	35	93	4	7	24	97	5
	9	48	85	3	8	35	87	4
	10	61	73	2	9	47	76	3
	11	72	52	1	10	57	61	2
	12	78	0	12	11	66	37	1
					12	70	0	12
June	5	1	118	7	5	4	117	7
	6	12	110	6	6	15	108	6
	7	24	103	5	7	26	100	5
	8	37	97	4	8	37	91	4
	9	50	89	3	9	49	80	3
	10	62	80	2	10	60	66	2
	11	74	61	1	11	69	42	1
	12	81	0	12	12	73	0	12

***Source*:** *ASHRAE Handbook 1981, Fundamentals*. Reprinted with permission.

3.12 HEAT EXCHANGE BETWEEN THE OUTER BUILDING SURFACE AND ITS SURROUNDINGS

Because atmospheric temperature is lower at high altitudes, there is always a radiant heat loss from the outer surface of the building to the sky vault without clouds. However, it may be offset or partly offset by reflected solar radiation from the ground on a sunny day. Radiant heat loss from the building needs to be calculated during nighttime and included in year-round energy estimation.

In commercial and institutional buildings using glass, concrete, or face brick on the outside surface of the building envelope, the migration of moisture through the glass pane is rather small. Because of the heavy mass of the concrete wall, the influence of the diurnal cyclic variation of the relative humidity of outdoor air on moisture transfer through the building envelope is also small. Therefore, for simplicity, the moisture transfer between the building envelope and the outside air can be ignored.

The heat balance at the outer building surface, as shown in Fig. 3.17, can be expressed as follows:

$$Q_{\text{sol}} + Q_{\text{ref}} = Q_{os} + (Q_{\text{rad}} - Q_{\text{at}}) + Q_{oi} \tag{3.91}$$

In Eq. (3.91), Q_{sol} represents the solar radiation absorbed by the outer surface of the building envelope, in Btu/h (W). It can be calculated as

$$Q_{\text{sol}} = \alpha_{os}[A_s(I_D + I_d) + A_{\text{sh}} I_d] \tag{3.92}$$

where α_{os} = absorptance of the outer surface of the building envelope. From Eq. (3.61), the reflection of solar radiation from any reflecting surface to the outer surface of the building and absorbed by it, or q_{ref}, Btu/h (W), is given by

$$Q_{\text{ref}} = A\alpha_{os} I_{\text{ref}} = A\rho_s \alpha_{os} F_{sr}(I_D + I_d) \tag{3.93}$$

where A = total area of the outer surface of the building envelope, $A = A_s + A_{\text{sh}}$, ft^2 (m^2). The term q_{os} indicates the convective heat transfer from the outer surface of the building outward, in

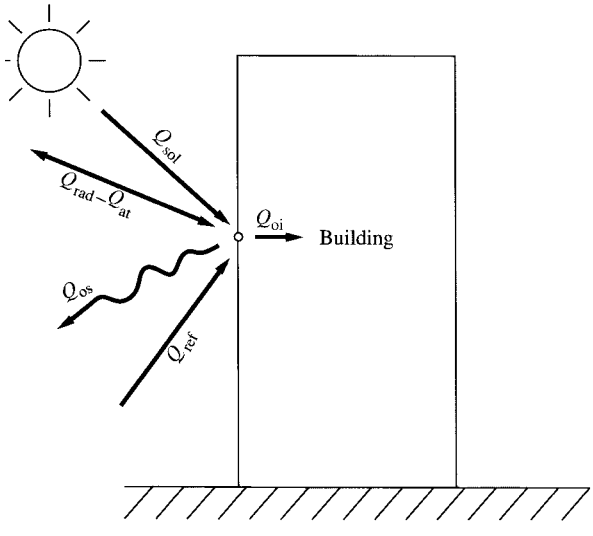

FIGURE 3.17 Heat balance at the outer surface of a building.

Btu/h (W). From Eq. (3.7), it can be calculated as

$$Q_{os} = h_c A(T_{os} - T_o) \tag{3.94}$$

where T_{os}, T_o = outer surface temperature of the building and the outdoor air temperature, respectively, °F (°C)
h_c = convective heat-transfer coefficient of outer surface, Btu/h·ft²·°F (W/m²·°C)

The term Q_{oi} denotes the inward heat flow from the outer building surface, in Btu/h (W). The term $Q_{\text{rad}} - Q_{\text{at}}$ indicates the net heat emitted from the outer surface of the building because of the radiation exchange between the surface and the atmosphere, in Btu/h (W). Here, Q_{rad} represents the longwave radiation emitted from the surface, and Q_{at} indicates the atmospheric radiation to the surface.

Kimura (1977) found that atmospheric radiation can be expressed as

$$Q_{\text{at}} = (1 - C_{cc}K_{cc})\tau T_{Ro}^4 \text{Br} + C_{cc}K_{cc}\tau T_{Rg}^4 \tag{3.95}$$

where C_{cc} = cloud cover factor, which can be obtained from local climate records, dimensionless
K_{cc} = cloudy reduction factor

Usually, the smaller the value of C_{cc}, the higher the clouds and the smaller the K_{cc} value. For simplicity, it can be calculated as

$$K_{cc} = 0.83 - 0.4C_{cc} \tag{3.96}$$

And Br is actually the emissivity of the atmosphere, and it can be expressed by an empirical formula developed by Brunt:

$$\text{Br} = 0.51 + 0.55\sqrt{p_w} \tag{3.97}$$

where p_w = water vapor pressure, psia. In Eq. (3.95), T_{Rg} represents the absolute ground temperature, °R (K), and T_{Ro} the absolute outdoor temperature, °R (K). Then, radiant heat loss can be written as

$$Q_{\text{rad}} - Q_{\text{at}} = \epsilon_{os} A F_{sat} \tau \{T_{Ros}^4 - [(1 - C_{cc}K_{cc})T_{Ro}^4 \text{Br} + C_{cc}K_{cc}T_{Rg}^4]\} \tag{3.98}$$

where ϵ_{os} = emissivity of outer surface of building
F_{sat} = shape factor between outer surface and atmosphere
T_{Ros} = absolute outer surface temperature of building, °R (K)

Sol-Air Temperature

For a sunlit outer surface of a building, if Q_{ref} is mainly from ground-reflected solar radiation, it is offset by $Q_{\text{rad}} - Q_{\text{at}}$, and Eq. (3.91) becomes

$$Q_{\text{sol}} = Q_{os} + Q_{oi}$$

Let $Q_{oi} = h_o A(T_{\text{sol}} - T_{os})$. Substituting for Q_{oi}, then, gives

$$\alpha_{os} I_t = h_o(T_{os} - T_o) + h_o(T_{\text{sol}} - T_{os})$$

and

$$T_{\text{sol}} = T_o + \frac{\alpha_{os} I_t}{h_o} \tag{3.99}$$

In Eq. (3.99), T_{sol} is called sol-air temperature, in °F (°C). It is a fictitious outdoor temperature that combines the effect of the solar radiation radiated on the outer surface of the building and the inward heat transfer due to the outdoor–indoor temperature difference.

Example 3.2. At midnight of July 21, the outdoor conditions of an office building are as follows:

Outdoor temperature	75°F (23.9°C)
Water vapor pressure of outdoor air	0.215 psia
Ground temperature	72°F (22.2°C)
Cloud cover factor C_{cc}	0.1
Emissivity of the outer surface	0.90
Shape factor between the surface and sky	0.5

If the outer surface temperature of this building is 76°F (24.4°C) and the Stefan-Boltzmann constant $\sigma = 0.1714 \times 10^{-8}$ Btu/h·ft^2·°R^4 (5.67×10^{-8} W/m^2·K^4), find the radiant heat loss from each square foot (square meter) of the vertical outer surface of this building.

Solution. From Eq. (3.97) and given data, we can see

$$\text{Br} = 0.51 + 0.55\sqrt{p_w} = 0.51 + 0.55\sqrt{0.215} = 0.765$$

And from Eq. (3.96), the cloudy reduction factor is

$$K_{cc} = 0.83 - 0.4C_{cc} = 0.83 - 0.4(0.1) = 0.79$$

From the given, $T_{Ros} = 76 + 460 = 536$°R, $T_{Ro} = 75 + 460 = 535$°R, and $T_{Rg} = 72 + 460 = 532$°R; then, from Eq. (3.98), the radiant heat loss from the outer surface of the building due to radiant exchange between the surface and the atmosphere is

$$\begin{aligned} Q_{\text{rad}} - Q_{\text{at}} &= \epsilon_{os} F_{\text{sat}} \sigma\{T_{Ros}^4 - [(1 - C_{cc}K_{cc})T_{Ro}^4\text{Br} + C_{cc}K_{cc}T_{Rg}^4]\} \\ &= 0.90 \times 0.5 \times 0.174\{(5.36)^4 - [(1 - 0.1 \times 0.79)(5.35)^4 \times 0.765 \\ &\quad + 0.1 \times 0.79 \times (5.32)^4]\} = 14.27 \text{ Btu/h·ft}^2 \text{ (45.02 W/m}^2\text{)} \end{aligned}$$

3.13 COMPLIANCE WITH ASHRAE/IESNA STANDARD 90.1-1999 FOR BUILDING ENVELOPE

An energy-efficient and cost-effective building envelope should meet the requirements and design criteria in ASHRAE/IESNA Standard 90.1-1999, Energy Standard for Buildings Except Low-Rise Residential Buildings; the DOE *Code of Federal Regulations*, Title10, Part 435, Subpart A, Performance Standard for New Commercial and Multi-Family High-Rise Residential Buildings; and local energy codes. The DOE Code is very similar to Standard 90.1.

The design and selection of the building envelope are generally the responsibility of an architect with the assistance of a mechanical engineer or contractor. Building envelopes are usually designed, or even constructed, before the HVAC&R system is designed. A speculative building is built of known use and type of occupancy, but the exact tenants are unknown. A shell building is built before the use and occupancy are determined.

Compliance with the ASHRAE/IES Standard 90.1-1999 for building envelopes includes the following. Refer to Standard 90.1-1999 for exceptions and details.

General Requirements

The requirements apply to the exterior building envelope which separates conditioned space from the outdoors and the semiexterior building envelope which separates the conditioned space from semiheated space or unconditioned space, or separated semiheated space from the unconditioned space or outdoors.

Semiheated space is an enclosed space, but not a conditioned space, within a building that is heated by a heating system whose output capacity is greater than or equal to 3.4 Btu/h·ft (10.71

W/m^2). Compliance includes mandatory provisions, the prescriptive building envelope option, and the building envelope trade-off option.

Mandatory Provisions

The standard mandates that insulation materials shall be installed in accordance with the manufacturer's recommendation to achieve rated *R* value of insulation. Rated *R* value of insulation is the thermal resistance of the insulation alone in units of $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$) at a mean temperature of 75°F (23.9°C). Rated *R* value refers to the thermal resistance of the added insulation in framing cavities or insulated sheathing only and does not include the thermal resistance of other building materials or air films.

Insulation shall be installed in a permanent manner in substantial contact with the inside surface. The roof insulation shall not be installed on a suspended ceiling with removable ceiling panels. Insulation outdoors shall be covered with a protective material to prevent damage from sunlight, moisture, landscaping operations, equipment maintenance, and wind. Insulation materials contacts with the ground shall have a water absoption rate no greater than 0.3 percent.

Fenestration performance shall be determined from production line units or representative units purchased. *U* factors shall be determined in accordance with National Fenestration Rating Council (NFRC) 100. Solar heat gain coefficient (SHGC) for the overall fenestration area shall be determined in accordance with NFRC 200. Visible light transmittance shall be determined in accordance with NFRC 300.

The following areas of building envelope shall be sealed, caulked, gasketed, or weather-stripped until air leakage is minimal:

- Joints around fenestration and door frames
- Junctions between walls and foundations, between wall corners, between walls and roofs, wall and floors, or walls and panels
- Openings because of penetrations of utility services through roofs, walls, and floors
- Fenestrations and doors built at site
- Building assemblies used as ducts and plenums
- Joints, seams, and penetrations due to vapor retarders
- Other openings in the building envelope

Air leakage of fenestration and doors shall be determined in accordance with NFRC 400. Air leakage shall not exceed 1.0 cfm/ft^2 (5.0 $L/s \cdot m^2$) for glazed swinging entrance doors and revolving doors and also 0.4 cfm/ft^2 (2.0 $L/s \cdot m^2$) for all other products. A door that separates conditioned space from the exterior shall be protected with an enclosed vestibule with all doors into and out of the vestibule installed with self-closing devices.

All mandatory provisions in Standard 90.1-1999 will be presented in the form "standard mandates that", whereas for nonmandatory provisions, only "standard specifics that" will be used.

Prescriptive Building Envelope Option

The exterior building envelope shall comply with the requirements for the appropriate conditioned space in Table 5.3 for various climate (located in Normative Appendix B of Standard 90.1-1999). In Table B1 to B26 of I-P edition:

- There are 26 tables, each of them has a number of heating degree days of 65°F (HDD65), and a number of cooling degree days of 50°F (CDD 50). Select a table with number of HDD 65 and CDD 50 equal to or most nearly equal to the values where the proposed building locates.
- There are three kinds of conditioned space: nonresidential, residential, semiheated.
- Total vertical fenestration area, including both fixed and operable fenestration shall be less than 50 percent of the gross wall area. The total skylight area, including glass and plastic skylights with or without a curb, shall be less than 5 percent of the roof area.

Fenestration, including fixed and operable vertical fenestration, shall have a *U* factor not greater than that specified in Table 5.3, 90.1-1999. Vertical fenestration shall have an SHGC not greater than that specified for all orientations in Table 5.3, 90.1-1999. There are only visible light transmittance criteria in the Building Envelope Trade-Off Option.

- All roofs shall have a rated *R* value of insulation not less than that specified in Table 5.3,90.1-1999. Skylight curbs shall be insulated to the level of the roofs with the insulation entirely above the deck or R-5, whichever is less.
- All above-grade walls shall have a rated value of insulation not less than that specified in Table 5.3, 90.1-1999. Mass wall heating capacity shall be determined from Table A-6 or A-7 in Standard 90.1-1999. Below-grade walls shall have a rated *R* value of insulation not less than that specified in Table 5.3, 90.1-1999.
- All floors and heated or unheated slab-on-grade floors shall have a rated *R* value of insulation not less than that specified in Table 5.3, 90.1-1999. Slab-on-grade floor insulation shall be installed around the perimeter of the slab-on-grade floor to the distance specified.
- All opaque doors shall have a *U* factor not greater than that specified in Table 5.3, 90.1-1999.

Building Envelope Trade-Off Option

The building envelope complies with the standard if the proposed building satisfied the provisions of 5.1 and 5.2 of Standard 90.1-1999 and the envelope performance factor of the proposed building is less than or equal to the envelope performance factor of the budget building. The envelope performance factor considers only the building envelope components. Schedules of operation, lighting power input, equipment power input, occupant density, and mechanical systems shall be the same for both the proposed building and budget building. Envelope performance factors shall be calculated using the procedures of Normative Appendix C, Standard 90.1-1999. Refer to Standard 90.1-1999 for details.

3.14 ENERGY-EFFICIENT AND COST-EFFECTIVE MEASURES FOR BUILDING ENVELOPE

According to the economic parametric analysis of the thermal design in Johnson et al. (1989), the following are energy-efficient and cost-effective measures for the design of building envelopes for office buildings.

Exterior Walls

An increase in the mass of the exterior wall, i.e., its thermal capacitance, reduces only the peak heating and cooling loads. The increase in electrical usage and capital investment often offsets the benefit of the decrease of peak loads. The variation of life-cycle costing is often negligible.

Increasing the insulation to more than 2-in. (50-mm) thick decreases only the natural gas usage for heating. This is cost-effective for areas with very cold winters. In some cases, the increase in the annual cost due to the increase in the capital investments of the insulation may balance the reduction of gas usage. A careful analysis is required.

Windows

The vertical fenestration area ratio (VFR) is the single parameter that most influences the building life-cycle costing among building envelopes for high-rise buildings. For many office buildings, VFR lies between 0.2 and 0.3.

Heat-absorbing and -reflective glasses produce a significant building cost savings in areas where solar heat control is important in summer. Double-panes, triple-panes, and low-emission films are effective in reducing the *U* value of the window assembly and therefore, the heating and cooling loads. Indoor shading devices with window management systems are cost-effective. For example, an indoor shading device can be turned on when solar heat gain exceeds 20 Btu/h·ft^2 (63 W/m^2). Although overhangs reduce the cooling loads, they may increase the need for electric lighting for daylit buildings. The effect of overhang usage on building life-cycle costing is not significant in many instances.

Infiltration

Infiltration has a significant influence on heating and cooling loads. Indoor air quality must be guaranteed by mechanical ventilation systems and sufficient outdoor air intake through these systems. It is desirable that windows and cracks in joints be well sealed. In multiple-story buildings, infiltration through elevator shafts, pipe shafts, and duct shafts should be reduced.

Energy-Efficient Measures for Commercial Buildings in the United States

According to the EIA's *Commercial Buildings Characteristics*, in 1992, the breakdown of energy conservation features for commercial buildings for a total area of 67,876 million ft^2 (6308 million m^2) in the United States is as follows:

Roof or ceiling insulation	74%
Wall insulation	49%
Storm or multiple glazing	44%
Tinted, reflective, or shading glass	37%
Exterior or interior shading devices	50%
Windows that are openable	43%

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CHAPTER 4
INDOOR AND OUTDOOR DESIGN CONDITIONS

4.1 INDOOR DESIGN CONDITIONS 4.1
4.2 HEAT EXCHANGE BETWEEN HUMAN BODY AND INDOOR ENVIRONMENT 4.2
Two-Node Model of Thermal Interaction 4.2
Steady-State Thermal Equilibrium 4.3
Transient Energy Balance 4.3
4.3 METABOLIC RATE AND SENSIBLE HEAT LOSSES FROM HUMAN BODY 4.4
Metabolic Rate 4.4
Mechanical Work 4.4
Sensible Heat Exchange 4.5
Clothing Insulation 4.7
4.4 EVAPORATIVE HEAT LOSSES 4.7
Respiration Losses 4.7
Evaporative Heat Loss from Skin Surface 4.7
Maximum Evaporative Heat Loss due to Regulatory Sweating 4.7
Diffusion Evaporative Heat Loss and Total Skin Wetness 4.8
4.5 MEAN RADIANT TEMPERATURE AND EFFECTIVE TEMPERATURE 4.9
Mean Radiant Temperature 4.9
Effective Temperature 4.14
4.6 FACTORS AFFECTING THERMAL COMFORT 4.14
4.7 THERMAL COMFORT 4.15
Fanger's Comfort Equation 4.15
ASHRAE Comfort Zones 4.17
Comfort-Discomfort Diagrams 4.17
4.8 INDOOR AIR TEMPERATURE AND AIR MOVEMENTS 4.20
Comfort Air Conditioning Systems 4.20
Design Considerations 4.21
Indoor Design Temperatures for Comfort Air Conditioning 4.21
Process Air Conditioning Systems 4.23
4.9 HUMIDITY 4.23
Comfort Air Conditioning Systems 4.23
Process Air Conditioning Systems 4.24
4.10 SICK BUILDING SYNDROME AND INDOOR AIR QUALITY 4.27
Indoor Air Contaminants 4.27
Basic Strategies to Improve Indoor Air Quality 4.29
Outdoor Air Requirements for Occupants 4.30
4.11 AIR CLEANLINESS 4.31
4.12 SOUND LEVEL 4.32
Sound and Sound Level 4.32
Sound Power Level and Sound Pressure Level 4.32
Octave Bands 4.33
Addition of Sound Levels 4.33
Human Response and Design Criteria 4.34
4.13 SPACE PRESSURE DIFFERENTIAL 4.37
4.14 OUTDOOR DESIGN CONDITIONS 4.38
Summer and Winter Outdoor Design Conditions 4.39
The Use of Outdoor Weather Data in Design 4.39
Outdoor Weather Characteristics and Their Influence 4.42
REFERENCES 4.42

4.1 INDOOR DESIGN CONDITIONS

Indoor design parameters are those that the air conditioning system influences directly to produce a required conditioned indoor environment in buildings. They are shown below and grouped as follows:

1. Basic design parameters

- Indoor air temperature and air movements
- Indoor relative humidity

2. Indoor air quality

- Air contaminants
- Outdoor ventilation rate provided
- Air cleanliness for processing

3. Specific design parameters

- Sound level
- Pressure differential between the space and surroundings

The indoor design parameters to be maintained in an air conditioned space are specified in the design document and become the targets to be achieved during operation. In specifying the indoor design parameters, the following points need to be considered:

1. Not all the parameters already mentioned need to be specified in every design project. Except for the indoor air temperature which is always an indoor design parameter in comfort air conditioning, it is necessary to specify only the parameters which are essential to the particular situation concerned.

2. Even for process air conditioning systems, the thermal comfort of the workers should also be considered. Therefore, the indoor design parameters regarding health and thermal comfort for the occupants form the basis of design criteria.

3. When one is specifying indoor design parameters, economic strategies of initial investment and energy consumption of the HVAC&R systems must be carefully investigated. Design criteria should not be set too high or too low. If the design criteria are too high, the result will be an excessively high investment and energy cost. Design criteria that are too low may produce a poor indoor air quality, resulting in complaints from the occupants, causing low-quality products, and possibly leading to expensive system alternations.

4. Each specified indoor design parameter is usually associated with a tolerance indicated as a $\pm$ sign, or as an upper or lower limit. Sometimes there is a traditional tolerance understood by both the designers and the owners of the building. For instance, although the summer indoor design temperature of a comfort air conditioning system is specified at 75 or 78°F (23.9 or 25.6°C), in practice a tolerance of $\pm 2-3$°F ($\pm 1.1-1.7$°C) is often considered acceptable.

5. In process air conditioning systems, sometimes a stable indoor environment is more important than the absolute value of the indoor parameter to be maintained. For example, it may not be necessary to maintain 68°F (20°C) for all areas in precision machinery manufacturing. More often, a 72°F (22.2°C) or even a still higher indoor temperature with appropriate tolerance will be more suitable and economical.

4.2 HEAT EXCHANGE BETWEEN HUMAN BODY AND INDOOR ENVIRONMENT

Two-Node Model of Thermal Interaction

In 1971, Gagge et al. recommended a two-node model of human thermal interaction. In this model, the human body is composed of two compartments: an inner body core, including skeleton, muscle and internal organs; and an outer shell of skin surface. The temperatures of the body core and the surface skin are each assumed to be uniform and independent. Metabolic heat production, external mechanical work, and respiratory losses occur only in the body core. Heat exchange between the body core and the skin surface depends on heat conduction from direct contact and the peripheral blood flow of the thermoregulatory mechanism of the human body.

Steady-State Thermal Equilibrium

When the human body is maintained at a steady-state thermal equilibrium, i.e., the heat storage at the body core and skin surface is approximately equal to zero, then the heat exchange between the human body and the indoor environment can be expressed by the following heat balance equation:

$$M = W + C + R + E_{sk} + E_{res} \tag{4.1}$$

where M = metabolic rate, Btu/h·ft^2 (W/m^2)
W = mechanical work performed, Btu/h·ft^2 (W/m^2)
$C + R$ = convective and radiative, or sensible heat loss from skin surface, Btu/h·ft^2 (W/m^2)
E_{sk} = evaporative heat loss from skin surface, Btu/h·ft^2 (W/m^2)
E_{res} = evaporative heat loss from respiration, Btu/h·ft^2 (W/m^2)

In Eq. (4.1), the ft^2 in the unit Btu/h·ft^2 applies to the skin surface area. The skin surface area of a naked human body can be approximated by an empirical formula proposed by Dubois in 1916

$$A_D = 0.657 m_b^{0.425} H_b^{0.725} \tag{4.2}$$

where A_D = Dubois surface area of naked body, ft^2 (m^2)
m_b = mass of human body, lb (kg)
H_b = height of human body, ft (m)

In an air conditioned space, a steady-state thermal equilibrium is usually maintained between the human body and the indoor environment.

Transient Energy Balance

When there is a transient energy balance between the human body and the indoor environment, the thermal interaction of the body core, skin surface, and indoor environment forms a rate of positive or negative heat storage both in the body core and on the skin surface.

The human body needs energy for physical and mental activity. This energy comes from the oxidation of the food taken into the human body. The heat released from this oxidation process is called metabolic heat. It dissipates from the skin surface of the human body into the surroundings. In a cold environment, the thermoregulatory mechanism reduces the rate of peripheral blood circulation, lowering the temperature of the skin and preventing any greater heat loss from the human body. However, if the heat loss and the mechanical work performed are greater than the rate of metabolic heat produced, then the temperatures of both the body core and the skin surface fall, and shivering or other spontaneous activities occur to increase the production of heat energy within the human body.

On the other hand, in a hot environment, if a large amount of heat energy needs to be dissipated from the human body, the physiological control mechanism increases the blood flow to the skin surface. This raises the skin temperature. If the heat produced is still greater than the heat actually dissipated and the temperature of the body core increased from its normal temperature of about 97.6 to about 98.6°F (36.4 to about 37.0°C), then liquid water is released from the sweat glands for evaporative cooling.

For a transient state of energy balance between the human body and the indoor environment, the rate of heat storage in the body core S_{cr} and the skin surface S_{sk}, both in Btu/h·ft^2 (W/m^2), can be calculated as

$$S_{cr} + S_{sk} = M - W - (C + R) - E_{sk} - E_{res} \tag{4.3}$$

4.3 METABOLIC RATE AND SENSIBLE HEAT LOSSES FROM HUMAN BODY

Metabolic Rate

The metabolic rate M is the rate of energy release per unit area of skin surface as a result of the oxidative processes in the living cells. Metabolic rate depends mainly on the intensity of the physical activities performed by the human body. The unit of metabolic rate is called the *met*. One met is defined as 18.46 Btu/h · ft^2 (58.24 W/m^2) of metabolic heat produced in the body core. In Table 4.1 are listed the metabolic rates of various activities.

Mechanical Work

Some of the energy released from the oxidative processes within the body core can be partly transformed to external mechanical work through the action of the muscles. Mechanical work W is usually expressed as a fraction of the metabolic rate and can be calculated as

$$W = \mu M \tag{4.4}$$

where μ = mechanical efficiency. For most office work, mechanical efficiency is μ less than 0.05. Only when there is a large amount of physical activity such as bicycling, lifting and carrying, or walking on a slope may increase μ to a value of 0.2 to 0.24.

TABLE 4.1 Metabolic Rate for Various Activities

	Metabolic rate	
Activity level	Met	Btu/h · ft^2
Resting		
Sleeping	0.7	13
Seated, quiet	1.0	18
Office work		
Reading, seated	1.0	18
Typing	1.1	20
Teaching	1.6	30
Domestic work		
Cooking	1.6–2.0	29–37
House cleaning	2.0–3.4	37–63
Walking		
Speed 2 mph	2.0	37
4 mph	3.8	70
Machine work		
Light	2.0–2.4	37–44
Heavy	4.0	74
Dancing, social	2.4–4.4	44–81
Sports		
Tennis, singles	3.6–4.0	66–74
Basketball	5.0–7.6	90–140
Wrestling	7.0–8.7	130–160

Source: Adapted with permission from *ASHRAE Handbook 1989, Fundamentals.*

Sensible Heat Exchange

Sensible heat loss or, occasionally, sensible heat gain $R + C$ represents the heat exchange between the human body and the indoor environment through convective and radiative heat transfer. Figure 4.1 shows the sensible heat exchange between the human body and the environment. The combined convective and radiative heat transfer can be calculated as

$$\begin{aligned} C + R &= f_{\text{cl}} h_c (T_{\text{cl}} - T_a) + f_{\text{cl}} h_r (T_{\text{cl}} - T_{\text{rad}}) \\ &= f_{\text{cl}} h (T_{\text{cl}} - T_o) \end{aligned} \tag{4.5}$$

where T_{cl} = mean surface temperature of clothing, °F (°C). The operative temperature T_o is defined as the weighted average of the mean radiant temperature T_{rad} and indoor air temperature T_a, both in °F (°C), that is,

$$T_o = \frac{h_r T_{\text{rad}} + h_c T_a}{h_r + h_c} \tag{4.6}$$

The surface heat-transfer coefficient is defined as

$$h = h_c + h_r \tag{4.7}$$

and the ratio of the clothed surface area to the naked surface area is

$$f_{\text{cl}} = \frac{A_{\text{cl}}}{A_D} \tag{4.8}$$

where h_c, h_r = convective and radiative heat-transfer coefficient, Btu/h · ft^2 · °F (W/m^2 · °C)
A_{cl} = surface area of clothed body, ft^2 (m^2)

The mean radiant temperature T_{rad} is discussed in greater detail in later sections.

According to Seppenan et al. (1972), the convective heat-transfer coefficient h_c for a person standing in moving air, when the air velocity is $30 \leq v \leq 300$ fpm ($0.15 \leq v \leq 1.5$ m/s), is

$$h_c = 0.0681 v^{0.69} \tag{4.9}$$

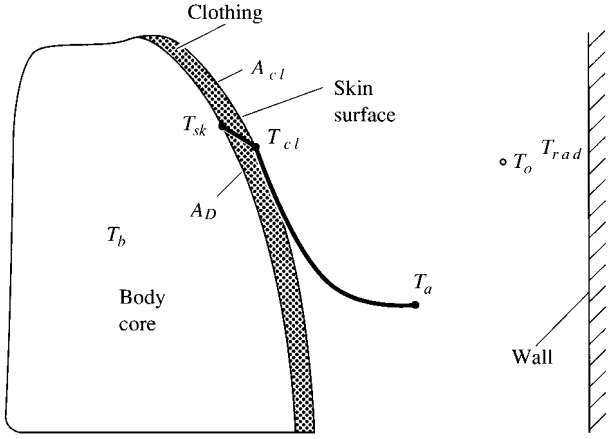


FIGURE 4.1 Sensible heat exchange between the human body and the indoor environment.

When the air velocity $v < 30$ fpm (0.15 m/s), $h_c = 0.7$ Btu/h·ft²·°F (4 W/m²·°C). For typical indoor temperatures and a clothing emissivity nearly equal to unity, the linearized radiative heat-transfer coefficient $h_r = 0.83$ Btu/h·ft²·°F (4.7 W/m²·°C). Let R_{cl} be the R value of clothing, in h·ft²·°F/Btu (m²·°C/W). Then

$$C + R = \frac{T_{sk} - T_{cl}}{R_{cl}} \tag{4.10}$$

where T_{sk} = mean skin surface temperature, °F (°C). If the human body is able to maintain a thermal equilibrium with very little evaporative loss from the skin surface, then the skin temperature $T_{sk.n}$ will be around 93°F (33.9°C).

Combining Eqs. (4.5) and (4.10), and eliminating T_{cl}, we find

$$\begin{aligned} C + R &= \frac{f_{cl}h(T_{sk} - T_o)}{R_{cl}f_{cl}h + 1} \\ &= F_{cl}f_{cl}h(T_{sk} - T_o) \end{aligned} \tag{4.11}$$

In this equation, the dimensionless clothing efficiency F_{cl} is defined as

$$F_{cl} = \frac{1}{R_{cl}f_{cl}h + 1} = \frac{T_{cl} - T_o}{T_{sk} - T_o} \tag{4.12}$$

In Eq. (4.11), if $T_o > T_{sk}$, then $C + R$ could be negative, i.e., there could be a sensible heat gain.

TABLE 4.2 Insulation Values for Clothing Ensembles*

Ensemble description†	R_{cl}, clo	R_{cl}, h·ft²·°F / Btu	f_{cl}
Walking shorts, short-sleeve shirt	0.41	0.36	1.11
Fitted trousers, short-sleeve shirt	0.50	0.44	1.14
Fitted trousers, long-sleeve shirt	0.62	0.55	1.19
Same as above, plus suit jacket	0.96	0.85	1.23
Loose trousers, long-sleeve shirt, long-sleeve sweater, T-shirt	1.01	0.89	1.28
Sweat pants, sweatshirt	0.77	0.68	1.19
Knee-length skirt, short-sleeve shirt, pantyhose (no socks), sandals	0.54	0.48	1.26
Knee-length skirt, long-sleeve shirt, full slip, pantyhose (no socks)	0.67	0.59	1.29
Long-sleeve coveralls, T-shirt	0.72	0.63	1.23
Overalls, long-sleeve shirts, long underwear tops and bottoms, flannel long-sleeve shirt	1.00	0.88	1.28

*For mean radiant temperature equal to an air temperature and air velocity less than 40 fpm.

†Unless otherwise noted, all ensembles included briefs or panties, shoes, and socks.

Source: Adapted from McCullough and Jones (1984). Reprinted with permission.

Clothing Insulation

Clothing insulation R_{cl} can be determined through measurements on a heated manikin, a model of the human body for laboratory experiments. After $C + R$ is measured from the thermal manikin in a controlled indoor environment, R_{cl} can be calculated from Eq. (4.11) since f_{cl}, T_{sk}, T_o, and h are also known values. Clothing insulation R_{cl} can be expressed either in $h \cdot ft^2 \cdot °F/Btu$ ($m^2 \cdot °C/W$) or in a new unit called *clo*, where 1 clo = 0.88 $h \cdot ft^2 \cdot °F/Btu$ (0.16 $m^2 \cdot °C/W$).

Clothing insulation R_{cl} values and area ratios f_{cl} for typical clothing ensembles, taken from McCullough and Jones (1984), are listed in Table 4.2.

4.4 EVAPORATIVE HEAT LOSSES

Evaporative heat loss E is heat loss due to the evaporation of sweat from the skin surface E_{sk} and respiration losses E_{res}. Actually, metabolic heat is mainly dissipated to the indoor air through the evaporation of sweat when the indoor air temperature is nearly equal to the skin temperature.

Respiration Losses

During respiration, there is convective heat loss C_{res} that results from the temperature of the inhaled air being increased to the exhaled air temperature, or about 93°F (33.9°C). There is also a latent heat loss L_{res} due to evaporation of liquid water into water vapor inside the body core. The amount of respiration loss depends mainly on the metabolic rate. In summer, at an indoor temperature of 75°F (23.9°C) and a relative humidity of 50 percent, respiratory losses $E_{res} = C_{res} + L_{res}$ are approximately equal to $0.09M$. In winter, E_{res} is slightly greater. For simplicity, let $E_{res} = 0.1M$.

Evaporative Heat Loss from Skin Surface

Evaporative heat loss from the skin surface E_{sk} consists of (1) the evaporation of sweat as a result of thermoregulatory mechanisms of the human body E_{rsw} and (2) the direct diffusion of liquid water from the skin surface E_{dif}. Evaporative heat loss due to regulatory sweating E_{rsw}, $Btu/h \cdot ft^2$ (W/m^2), is directly proportional to the mass of the sweat produced, i.e.,

$$E_{rsw} \propto \dot{m}_{rsw} h_{fg} \tag{4.13}$$

where $\dot{m}_{rsw}$ = mass flow rate of sweat produced, $lb/h \cdot ft^2$ ($kg/s \cdot m^2$)
h_{fg} = latent heat of vaporization at 93°F (33.9°C), Btu/lb (J/kg)

Maximum Evaporative Heat Loss due to Regulatory Sweating

The wetted portion of the human body needed for the evaporation of a given quantity of sweat w_{rsw} is

$$w_{rsw} = \frac{E_{rsw}}{E_{max}} \tag{4.14}$$

In Eq. (4.14), E_{max} represents the maximum evaporative heat loss due to regulatory sweating when the skin surface of the human body is entirely wet. Its magnitude is directly proportional to the vapor pressure difference between the wetted skin surface and the indoor ambient air, and it can be

TABLE 4.3 Moisture Permeability of Clothing Ensembles

Ensemble description	i_m
Cotton/polyester long-sleeve shirt, long trousers, street shoes, socks, briefs	0.385
Cotton short-sleeve shirt, long trousers, work boots, socks, briefs, cotton gloves	0.41
Cotton/nylon long-sleeve shirt, cotton/nylon trousers, combat boots, socks, helmet liner (army battle dress uniform)	0.36

*Measured with $T_{rad} = T_a$, and air velocity = 40 fpm.

Source: Adapted with permission from *ASHRAE Handbook 1989, Fundamentals*.

calculated as

$$E_{max} = h_{e,c}(p_{sk,s} - p_a) \tag{4.15}$$

where $p_{sk,s}$ = saturated water vapor pressure at skin surface temperature, psia (kPa)
p_a = water vapor pressure of ambient air, psia (kPa)
$h_{e,c}$ = overall evaporative heat-transfer coefficient of clothed body, in Btu/h · ft²·psi (W/m² · kPa)

Woodcock (1962) proposed the following relationship between $h_{e,c}$ and h_s, the overall sensible heat transfer coefficient, in Btu/h · ft² · psi (W/m² · kPa):

$$i_m \text{LR} = \frac{h_{e,c}}{h_s} \tag{4.16}$$

The moisture permeability index i_m denotes the moisture permeability of the clothing and is dimensionless. Clothing ensembles worn indoors usually have an i_m = 0.3 to 0.5. The moisture permeability indexes i_m of some clothing ensembles are presented in Table 4.3.

The Lewis relation LR in Eq. (4.16) relates the evaporative heat-transfer coefficient h_e and the convective heat-transfer coefficient h_c, both in Btu /h · ft² · °F · (W/m² · °C). LR = $f(h_e/h_c)$ has a magnitude of 205°F/psi (16.5°C/kPa). In Eq. (4.16), the overall sensible heat-transfer coefficient h_s can be calculated as

$$h_s = \frac{1}{R_t} = \frac{f_{cl}h}{f_{cl}hR_{cl} + 1} \tag{4.17}$$

where R_t = total resistance to sensible heat transfer between the skin and the indoor environment, h · ft² · °F/Btu (m² · °C/W).

Diffusion Evaporative Heat Loss and Total Skin Wetness

The minimum level of evaporative heat loss from the skin surface occurs when there is no regulatory sweating and the skin wetness due to direct diffusion $E_{df,min}$, Btu/h · ft² (W/m²), is approximately equal to $0.06E_{max}$ under normal conditions, or

$$E_{df,min} = 0.06E_{max} \tag{4.18}$$

When there is a heat loss from regulatory sweating E_{rsw}, the diffusion evaporative heat loss E_{dif}, Btu/h · ft² (W/m²), for the portion of skin surface that is not covered with sweat can be

calculated as

$$E_{\text{dif}} = (1 - w_{\text{rsw}})0.06E_{\max} \tag{4.19}$$

Therefore, the total evaporative heat loss from the skin surface is

$$\begin{aligned} E_{\text{sk}} &= E_{\text{rsw}} + E_{\text{dif}} \\ &= w_{\text{rsw}}E_{\max} + (1 - w_{\text{rsw}})0.06E_{\max} = (0.06 + 0.94w_{\text{rsw}})E_{\max} \\ &= w_{\text{sk}}E_{\max} \end{aligned} \tag{4.20}$$

In Eq. (4.20), w_{sk} is called the total skin wetness; it is dimensionless, and it can be calculated as

$$w_{\text{sk}} = \frac{E_{\text{sk}}}{E_{\max}} \tag{4.21}$$

4.5 MEAN RADIANT TEMPERATURE AND EFFECTIVE TEMPERATURE

Mean Radiant Temperature

Mean radiant temperature T_{Rad} is defined as the temperature of a uniform black enclosure in which an occupant would have the same amount of radiative heat exchange as in an actual indoor environment. Mean radiant temperature T_{Rad}, °R(K), can be calculated by the expression

$$T_{\text{Rad}}^4 = T_{R1}^4 F_{0-1} + T_{R2}^4 F_{0-2} + \cdots + T_{Rn}^4 F_{0-n} \tag{4.22}$$

where $T_{R1}, T_{R2}, \cdots, T_{Rn}$ = absolute temperature of surrounding surfaces of indoor environment, °R(K)
F_{0-1} = shape factor denoting fraction of total radiant energy leaving surface of occupant's clothing 0 and arriving on the surface 1
F_{0-2} = fraction of total radiant energy leaving surface 0 and arriving on surface 2, etc.

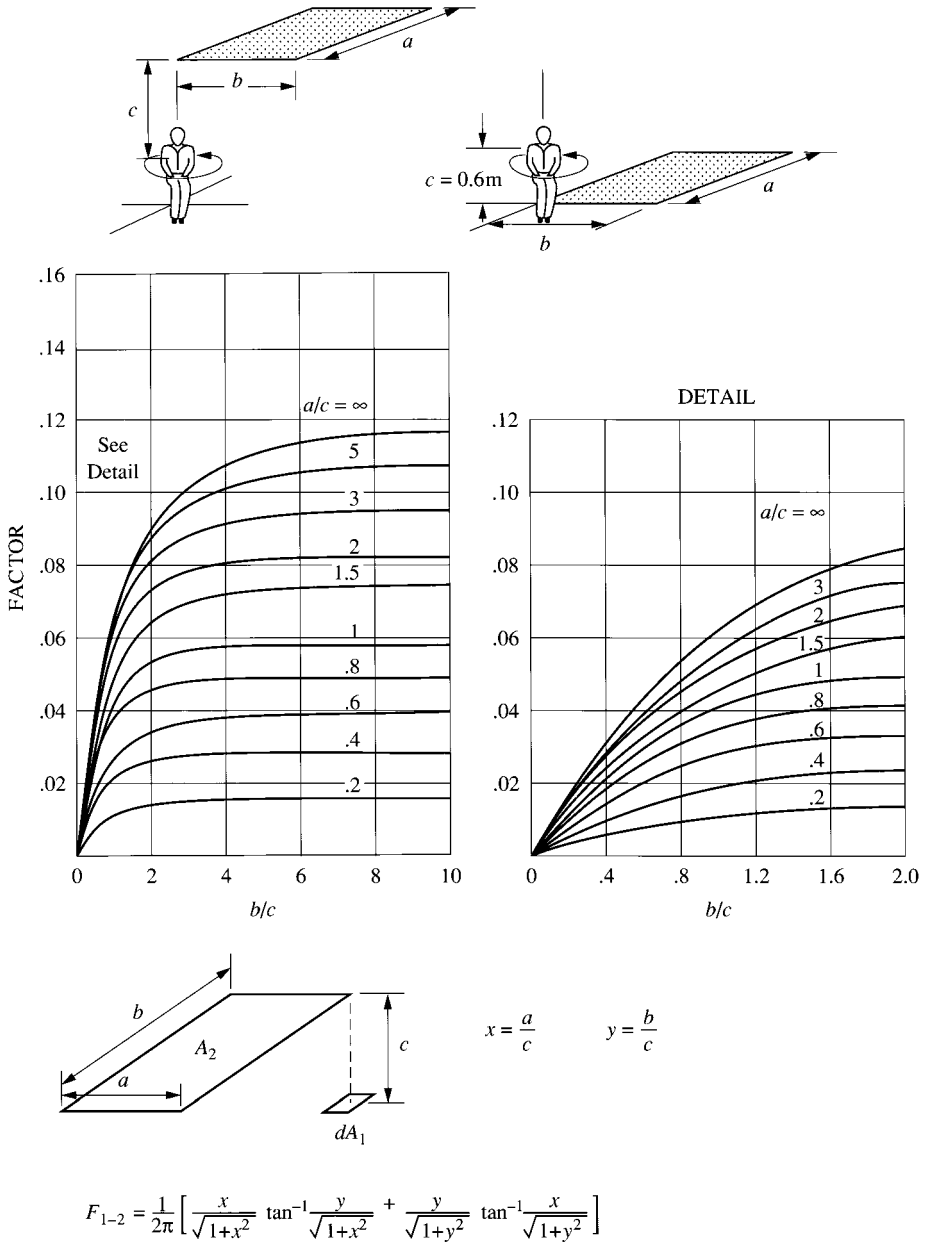
Shape factors F_{0-n} depend on the position and orientation of the occupant as well as the dimensions of the enclosure. One can use Figs. 4.2 and 4.3 to estimate the mean value of the shape factor between a seated person and rectangular surfaces. The sum of the shape factors of all the surfaces with respect to the seated occupant in an enclosure is unity.

The temperature measured by a globe thermometer, called the *globe temperature*, is often used to estimate the mean radiant temperature. The globe thermometer consists of a copper hollow sphere of 6-in. (152-mm) diameter that is coated with black paint on the outer surface. A precision thermometer or thermocouple is inserted inside the globe with the sensing bulb or the thermojunction located at the center of the sphere. Because the net radiant heat received at the globe surface is balanced by the convective heat transfer from the globe surface in reaching a thermal equilibrium, according to Bedford and Warmer (1935), such a relationship gives

$$T_{\text{Rad}}^4 = T_{Rg}^4 + 0.247 \times 10^9 v^{0.5}(T_{Rg} - T_{Ra}) \tag{4.22a}$$

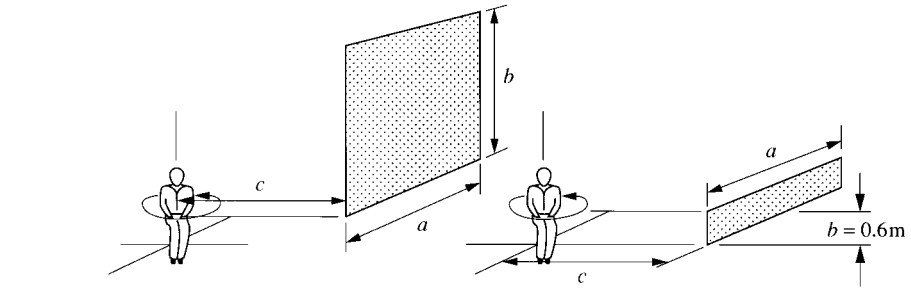
where T_{Rad} = absolute mean radiant temperature, °R (K)
T_{Rg} = absolute globe temperature, °R (K)
v = ambient air velocity, fpm (m/s)
T_{Ra} = absolute air temperature, °R (K)

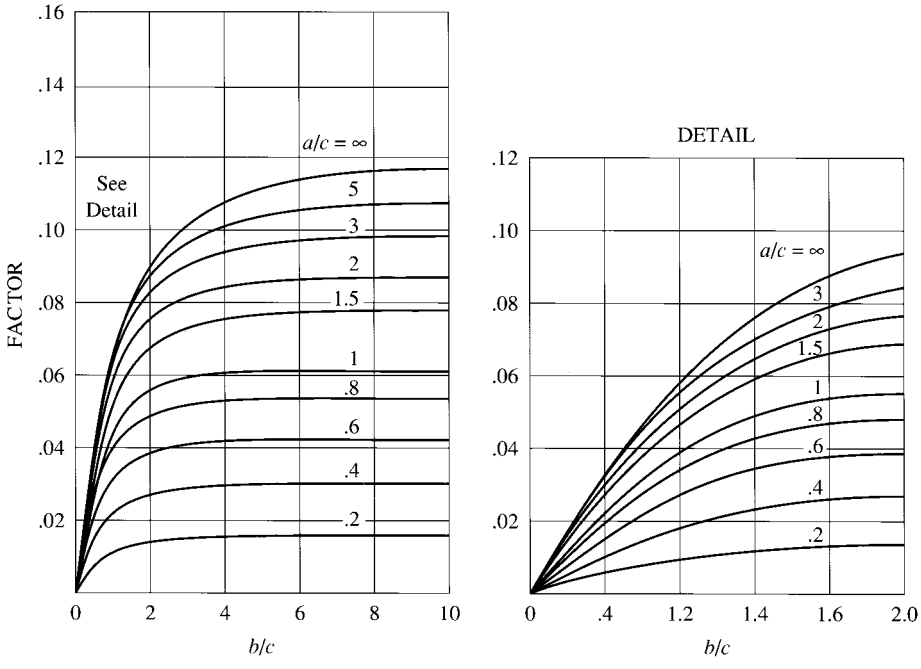
After T_{Rg}, v, and T_{Ra} are measured, the mean radiant temperature T_{Rad} can be calculated from Eq. (4.22). The mean radiant temperature indicates the effect, due to the radiant energy from the

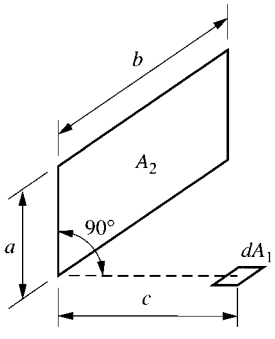


$$F_{1-2} = \frac{1}{2\pi}\left[\frac{x}{\sqrt{1+x^2}}\tan^{-1}\frac{y}{\sqrt{1+x^2}} + \frac{y}{\sqrt{1+y^2}}\tan^{-1}\frac{x}{\sqrt{1+y^2}}\right]$$

FIGURE 4.2 Mean value of shape factor between a sedentary person and a horizontal plane. (*Source: P. O. Fanger, Thermal Comfort Analysis and Applications in Environmental Engineering, 1972. Reprinted with permission.)*







$$x = \frac{a}{b} \qquad y = \frac{c}{b}$$

$$F_{1-2} = \frac{1}{2\pi}\left[\tan^{-1}\frac{1}{Y} - \frac{Y}{\sqrt{x^2+y^2}}\tan^{-1}\frac{1}{\sqrt{x^2+y^2}}\right]$$

FIGURE 4.3 Mean value of shape factor between a seated person and a vertical plane. (*Source: P. O. Fanger, Thermal Comfort Analysis and Applications in Environmental Engineering, 1970. Reprinted with permission.*)

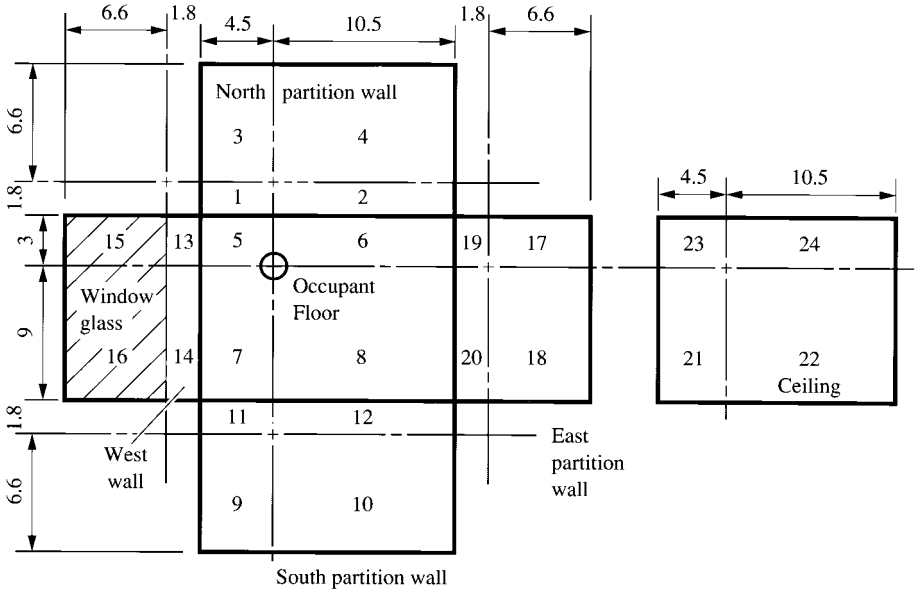


FIGURE 4.4 Dimension, in feet, of a private office.

surroundings, on radiant exchange between an occupant or any substance and the enclosure. Such an influence may be significant if the mean radiant temperature is several degrees higher than the temperature of the indoor air.

Example 5.1. The dimensions of a private office and the location of a person seated within it are shown in Fig. 4.4. The surface temperatures of the enclosure are as follows:

West window	88°F (31.1°C)
West wall	80°F (26.7°C)
North partition wall	75°F (23.9°C)
East partition wall	75°F (23.9°C)
South partition wall	75°F (23.9°C)
Floor	78°F (25.6°C)
Ceiling	77°F (25°C)

Calculate the mean radiant temperature of the enclosure that surrounds this office. The orientation of the seated occupant is unknown.

Solution

1. Regarding the north partition wall, the shape factor denotes the fraction of the total radiant energy that leaves the outer surface of the clothing of the occupant (surface 0) and arrives directly on the north partition wall (surfaces 1, 2, 3, and 4) and is given by

$$F_{0-1,2,3,4} = F_{0-1} + F_{0-2} + F_{0-3} + F_{0-4}$$

Here F_{0-1} is the shape factor for the radiation from surface 0 to surface 1 of the north partition wall. Based on the curves in Fig. 4.3, for a ratio of $b/L = 1.8/3 = 0.6$ and a ratio of $a/L = 4.5/3 = 1.5$,

TABLE 4.4 Values of F_{0-n} and $T_{Rn}^4 F_{0-n}$ in Example 4.1

Surface	Surface temperature, °R	Shape factor	b/L	a/L	F_{0-n}	$T_{Rn}^4 F_{0-n} \times 10^8$
North partition wall	535	F_{0-1}	0.6	1.5	0.04	32.77
		F_{0-2}	0.6	3.5	0.045	36.87
		F_{0-3}	2.2	1.5	0.07	57.35
		F_{0-4}	2.2	3.5	0.087	71.27
East partition wall	535	F_{0-17}	0.63	0.29	0.014	11.47
		F_{0-18}	0.63	0.86	0.03	24.58
		F_{0-19}	0.17	0.29	0.004	3.28
		F_{0-20}	0.17	0.86	0.009	7.37
South partition wall	535	F_{0-9}	0.73	0.5	0.023	18.84
		F_{0-10}	0.73	1.17	0.038	31.13
		F_{0-11}	0.2	0.5	0.008	6.55
		F_{0-12}	0.2	1.17	0.013	10.65
West wall	540	F_{0-13}	0.4	0.67	0.018	15.31
	540	F_{0-14}	0.4	2	0.03	25.51
West window	548	F_{0-15}	1.6	0.67	0.04	36.07
	548	F_{0-16}	1.6	2	0.07	63.13
Floor	538	F_{0-5}	1.67	2.5	0.068	56.97
		F_{0-6}	1.67	5.8	0.073	61.16
		F_{0-7}	5	2.5	0.087	72.89
		F_{0-8}	5	5.8	0.102	85.45
Ceiling	537	F_{0-21}	1.36	0.68	0.033	27.44
		F_{0-22}	1.36	1.59	0.052	43.24
		F_{0-23}	0.45	0.68	0.015	12.47
		F_{0-24}	0.45	1.59	0.025	20.79
				Σ	0.994	$8.32.56 \times 10^8$

the shape factor $F_{0-1} = 0.04$. Here L is the horizontal distance from the occupant to the north partition wall. The shape factors F_{0-2}, F_{0-3}, and F_{0-4} can be calculated in the same manner.

2. To determine the shape factor F_{0-5} for the radiation from the occupant (surface 0) to the floor (surface 5), we note that the vertical distance L from the center of the seated occupant to the floor is 1.8 ft. From the curves in Fig. 4.2, the ratio $b/L = 3/1.8 = 1.67$, and the ratio $a/L = 4.5/1.8 = 2.5$; thus the shape factor $F_{0-5} = 0.068$. All the remaining shape factors can be determined in the same manner as listed in Table 4.4.

3. For the north partition wall (surface 1),

$$T_{R1}^4 F_{0-1} = (75 + 460)^4 \times 0.04 = 32.77 \times 10^8$$

Other products $T_{Rn}^4 F_{0-n}$ can be similarly calculated, as listed in Table 4.4. The sum of the products $\Sigma T_{Rn}^4 F_{0-n} = 832.56 \times 10^8$. Therefore

$$T_{\text{Rad}}^4 = 832.56 \times 10^8$$

That is,

$$T_{\text{Rad}} = 537.2°\text{R} \qquad \text{or} \qquad T_{\text{rad}} = 77.2°\text{F}(25.1°\text{C})$$

4. The sum of the shape factors $\Sigma F_{0-n} = 0.994$ is nearly equal to 1.

Effective Temperature

The *effective temperature* ET* is the temperature of an environment that causes the same total heat loss from the skin surface as in an actual environment of an operative temperature equal to ET* and at a relative humidity of 50 percent. And ET* can be calculated as

$$\text{ET}^* = T_o + w_{\text{sk}} i_m \,\text{LR}\,(0.5 p_{\text{ET},s}) \tag{4.23}$$

where $p_{\text{ET},s}$ = saturated water vapor pressure at ET*, psia (kPa abs.). The right-hand side of Eq. (4.23) describes the conditions of the indoor air regarding the total heat loss from the human body. The same value of the combination $T_o + w_{\text{sk}} i_m \text{LR}(0.5 p_{\text{ET},s})$ results in the same amount of total heat loss from the skin surface, if other parameters remain the same. Theoretically, total skin wetness w_{sk} and clothing permeability index i_m are constants for a specific ET* line.

Because the effective temperature is based on the operative temperature T_o, it is a combined index of T_a, T_{rad}, and p_a. In an indoor air temperature below 77°F (25°C), the constant-ET* lines are nearly parallel to the skin temperature lines for sedentary occupants with a clothing insulation of 0.6 clo; therefore, ET* values are reliable indexes to indicate thermal sensations at normal indoor air temperature during low activity levels. The term *effective temperature* was originally proposed by Houghton and Yaglou in 1923. A new definition of ET* and its mathematical expression were developed by Gagge et al. in 1971. It is the environmental index commonly used in specifying and assessing thermal comfort requirements.

4.6 FACTORS AFFECTING THERMAL COMFORT

Daily experience and many laboratory experiments all show that thermal comfort occurs only under these conditions:

1. There is a steady-state thermal equilibrium between the human body and the environment; i.e., heat storage of the body core S_{cr} and the skin surface S_{sk} are both equal to zero.
2. Regulatory sweating is maintained at a low level.

From the heat balance equation at steady-state thermal equilibrium Eq. (4.1) we have

$$M = W + C + R + E_{\text{sk}} + E_{\text{res}}$$

Let $E_{\text{res}} = 0.1M$ and the mechanical efficiency $\mu = 0.05M$. From Eq. (4.11), $C + R$ can be determined. Also, from Eqs. (4.15), (4.16), and (4.20), E_{sk} is a known value. If we substitute into Eq. (4.1), the heat balance equation at steady-state thermal equilibrium can be expressed as

$$M(1 - 0.05 - 0.1) = F_{\text{cl}} f_{\text{cl}} h(T_{\text{sk}} - T_o) + w_{\text{sk}} i_m \,\text{LR}\, h_s(p_{\text{sk},s} - p_a)$$

or

$$0.85M = F_{\text{cl}} f_{\text{cl}} h(T_{\text{sk}} - T_o) + w_{\text{sk}} i_m \text{LR} h_s(p_{\text{sk},s} - p_a) \tag{4.24}$$

In Eq. (4.24), the physiological and environmental factors that affect the balance—the metabolic rate and the heat losses on the two sides of the equation—are as follows:

1. Metabolic rate M determines the magnitude of the heat energy that must be released from the human body, i.e., the left-hand side of the equation.
2. Indoor air temperature T_a is a weighted component of the operating temperature T_o. It also affects the sensible heat loss and the vapor pressure of indoor air p_a in the calculation of the evaporative loss from the skin surface.

3. Mean radiant temperature T_{rad} is another weighted component of the operating temperature T_o. It affects the sensible heat loss from the human body.
4. Relative humidity of the ambient air φ_a is the dominating factor that determines the difference $p_{sk,s} - p_a$ in the evaporative loss from the skin surface. Air relative humidity becomes important when the evaporative heat loss due to regulatory sweating is the dominating heat loss from the human body.
5. Air velocity v_a influences the heat transfer coefficient h and the clothing efficiency F_{cl} in the term in Eq. (4.24) for the sensible heat loss from the human body. It also affects the overall sensible heat transfer coefficient h_s in the evaporative heat loss term and the clothing permeability i_m term in Eq. (4.24).
6. Clothing insulation R_{cl} affects the clothing efficiency F_{cl}, the area ratio f_{cl}, the heat transfer coefficient h, the clothing permeability index i_m, and the overall sensible heat-transfer coefficient h_s.

4.7 THERMAL COMFORT

Thermal comfort is defined as the state of mind in which one acknowledges satisfaction with regard to the thermal environment. In terms of sensations, thermal comfort is described as a thermal sensation of being neither too warm nor too cold, defined by the following seven-point thermal sensation scale proposed by ASHRAE:

-3 = cold
-2 = cool
-1 = slightly cool
0 = neutral
$+1$ = slightly warm
$+2$ = warm
$+3$ = hot

Fanger's Comfort Equation

A steady-state energy balance is a necessary condition for thermal comfort, but is not sufficient by itself to establish thermal comfort. Fanger (1970) calculated the heat losses for a comfortable person, experiencing a neutral sensation, with corresponding skin temperature T_{sk} and regulatory sweating E_{rsw}. The calculated heat losses L are then compared with the metabolic rate M. If $L = M$, the occupant feels comfortable. If $L > M$, then this person feels cool; and if $L < M$, then this person feels warm.

Using the responses of 1396 persons during laboratory experiments at Kansas State University of the United States and Technical University of Denmark, Fanger developed the following equation to calculate the *predicted mean vote* (PMV) in the seven-point thermal sensation scale:

$$\text{PMV} = (0.303e^{-0.036M} + 0.276)(M - L) \tag{4.25}$$

In Eq. (4.25), the metabolic rate M and heat losses L are both in $\text{Btu/h} \cdot \text{ft}^2$ (W/m^2). According to Fanger's analysis, the *predicted percentage of dissatisfied* (PPD) vote for thermal comfort at a PMV = 0 is 5 percent, and at a PMV = ± 1 is about 27 percent.

Tables of PMV and comfort charts including various combinations of operating temperature T_o, air velocity v, metabolic rate M, and clothing insulation R_{cl} have been prepared to determine comfortable conditions conveniently. Fanger's comfort charts also include relative humidity. Six of his

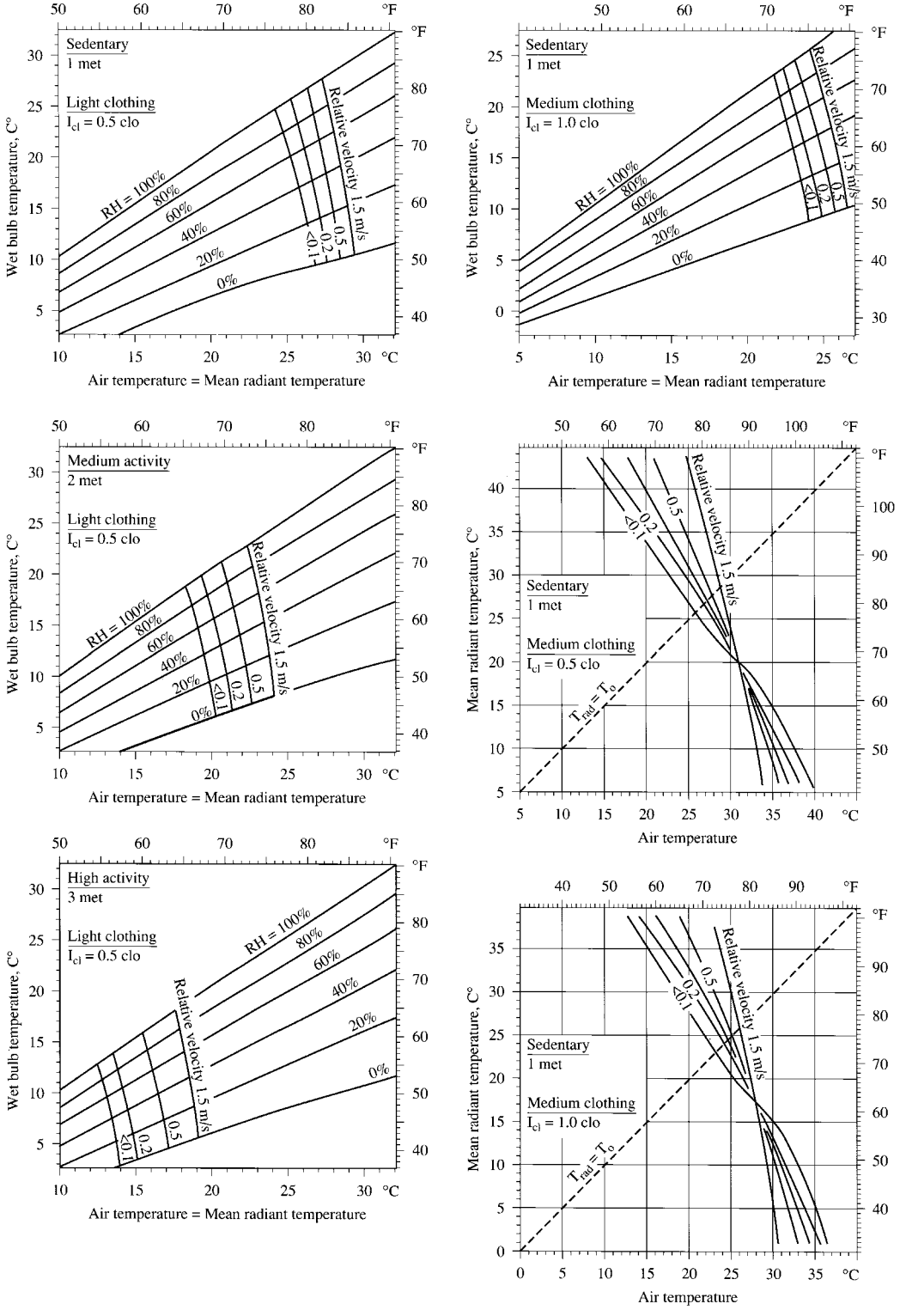


FIGURE 4.5 Fanger's comfort charts. *(Abridged with permission from ASHRAE Handbook 1981, Fundamentals.)*

comfort charts at various activity levels, wet-bulb temperatures, relative humidities, and air velocities are shown in Fig. 4.5. In the first four charts, the air temperatures are equal to the mean radiant temperatures. Three of these four have a clothing insulation of 0.5 clo. The other is for 1-met activity level and 1 clo, because one rarely finds an occupant with such heavy outwear at an activity level of 2 or 3 met. In the fifth and sixth charts, the air temperature could be different from the mean radiant temperature with a constant relative humidity of 50 percent. These comfort variations clearly show that all six factors—air temperature T_a, mean radiant temperature T_{rad}, relative humidity φ, air velocity v, metabolic rate M, and clothing insulation R_{cl}—seriously affect the thermal comfort.

For example, from Fanger's comfort chart, a sedentary occupant at an activity level of 1.0 met, with a clothing insulation of 0.5 clo, in an air conditioned space at a relative humidity of 50 percent and an air velocity less than 20 fpm (0.1 m/s), feels comfortable with an air temperature equal to the mean radiant temperature of 78°F (25.6°C). If all values were identical except for a 2-met activity level, the temperature would need to be 67°F (19.4°C) for the same level of comfort.

Another factor, the duration of exposure to the indoor thermal environment, should be discussed here. If an indoor environment can provide thermal comfort for the occupant, the duration of the exposure has no significant influence upon the physiological responses of the person's thermal regulatory mechanism. If the indoor environment is uncomfortable, subjecting the subject to a certain degree of heat or cold stress, the time exposure will influence the person's physiological response.

ASHRAE Comfort Zones

Based on results of research conducted at Kansas State University and at other institutions, ANSI/ASHRAE Standard 55-1992 specified winter and summer comfort zones to provide for the selection of the indoor parameters for thermal comfort (see Fig. 4.6). This chart is based upon an occupant activity level of 1.2 met (69.8 W/m^2). For summer, typical clothing insulation is 0.5 clo, that is, light slacks and short-sleeve shirt or comparable ensemble; there is no minimum air speed that is necessary for thermal comfort. Standard 55-1992 recommended a summer comfort zone with an effective boundary temperature ET* = 73 to 79°F (22.5 to 26°C) at 68°F (20°C) wet-bulb as its upper-slanting boundary and dew-point temperature 36°F (2.2°C) as its bottom flat boundary. If the clothing insulation is 0.1 clo higher, the boundary temperatures both should be shifted 1°F (0.6°C) lower. Rohles et al. (1974) and Spain (1986) suggested that the upper boundary of the summer comfort zone can be extended to 85 or 86°F (29.4 or 30°C) ET* if the air velocity of the indoor air can be increased to 200 fpm (1 m/s) by a ceiling fan or other means.

The winter comfort zone is based upon a 0.9-clo insulation including heavy slacks, long-sleeve shirt, and sweater or jacket at an air velocity of less than 30 fpm (0.15 m/s). Standard 55-1992 recommended a winter comfort zone with an effective boundary temperature ET^* = 68 to 74°F (20 to 23.3°C) at 64°F (17.8°C) wet-bulb as its slanting upper boundary and at dew-point 36°F (2.2°C) as its bottom flat boundary.

Indoor air parameters should be fairly uniform in order to avoid local discomfort. According to Holzle et al., 75 to 89 percent of the subjects tested found the environment within this summer comfort zone to be thermally acceptable.

ASHRAE comfort zones recommend only the optimal and boundary ET* for the determination of the winter and summer indoor parameters. For clothing insulation, activity levels, and indoor air velocities close to the values specified in Standard 55-1992, a wide range of indoor design conditions are available.

Comfort-Discomfort Diagrams

A comfort diagram provides a graphical presentation of the total heat loss from the human body at various operative or air temperatures and indoor relative humidities when the activity level, level of clothing insulation, and air velocity are specified. The abscissa of the comfort diagram is the opera-

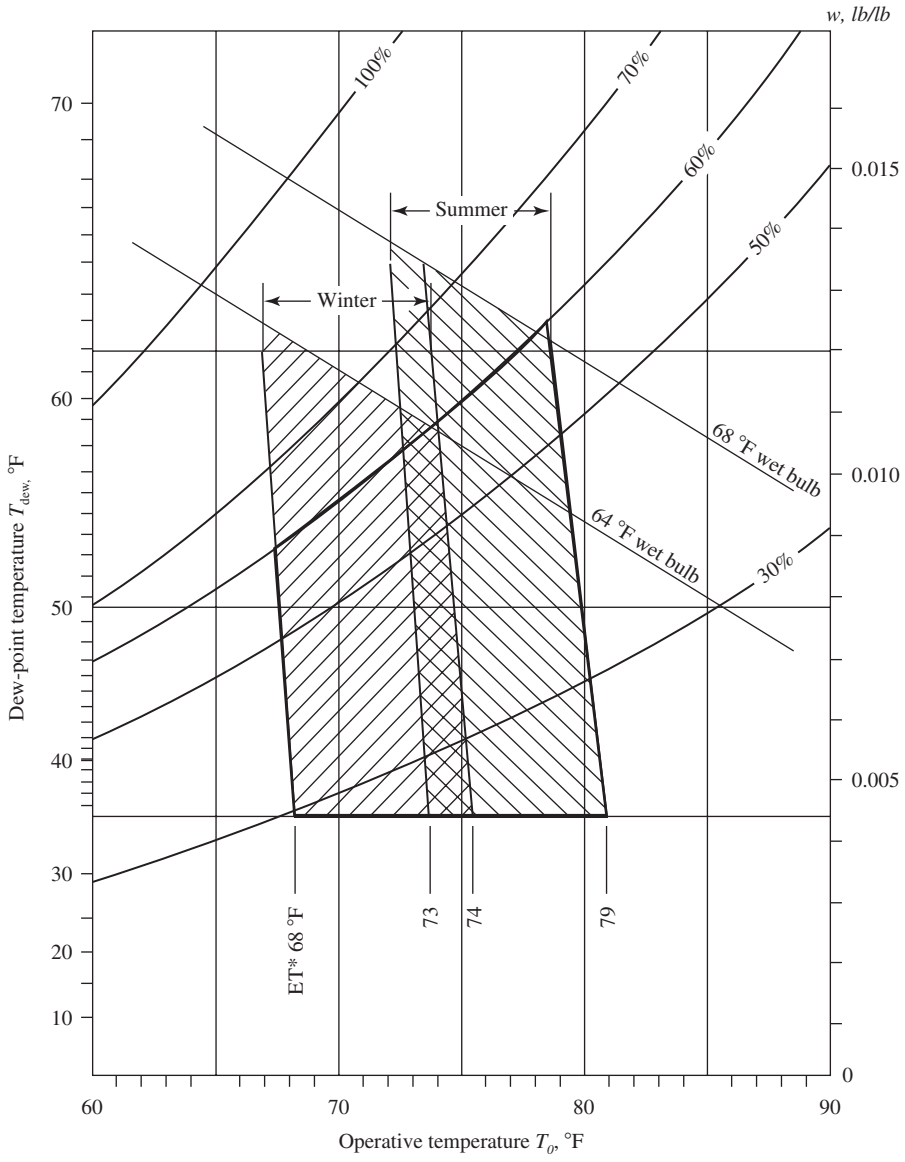


FIGURE 4.6 ASHRAE comfort zones. *(Adapted with permission from ANSI/ASHRAE Standard, 55–1992.)*

tive temperature T_o or ambient air temperature T_a, whereas the ordinate can be either water vapor pressure p or humidity ratio w. Figure 4.7 shows a comfort diagram with a sedentary activity level, a clothing insulation value of 0.6 clo, and still-air conditions, i.e., an air velocity $v < 20$ fpm (0.1 m/s). The curved lines represent relative humidity, and the straight lines represent effective temperature ET*. The short dash curves diverging from the ET* lines are total skin wetness w_{sk} lines. The figure is based upon $T_o = T_a$.

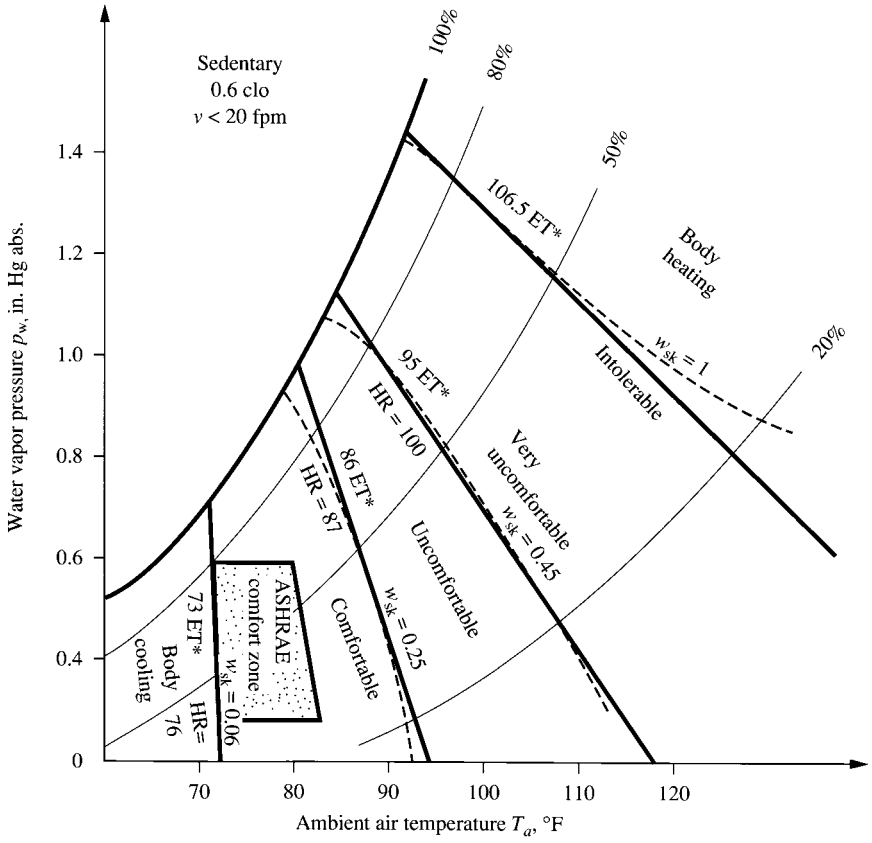


FIGURE 4.7 Comfort-discomfort diagram.

Effective temperature ET* lines are calculated according to Eq. (4.23). Because the total skin wetness w_{sk} is a constant for a specific ET* below 79°F ET*, the ET* lines and w_{sk} lines coincide with one other. At higher ET* values, w_{sk} lines curve to the left at high relative humidities. For low ambient air temperatures, evaporative heat loss from the skin surface E_{sk} is small; therefore, ET* and w_{sk} lines are nearly vertical. As E_{sk} becomes greater and greater, the slopes of the ET* and w_{sk} lines decrease accordingly.

The comfort diagram is divided into five zones by the ET* and w_{sk} lines:

1. *Body cooling zone.* For the condition given in Fig. 4.7, if the effective temperature ET* $<$ 73°F (22.8°C), the occupant will feel cold in this zone. Because the heat losses exceed the net metabolic rate, the skin and body core temperatures tend to drop gradually.

2. *Comfort zone.* This is the zone between the lower boundary ET* = 73°F (22.8°C), and w_{sk} = 0.06, and the higher boundary ET* = 86°F (30°C), and w_{sk} = 0.25. Steady-state thermal equilibrium is maintained between the occupant and the environment, and regulatory sweating is at a rather low level. The occupant will feel comfortable in this zone, and the heart rate (HR) is between 76 and 87 beats per minute.

ASHRAE's winter and summer comfort zones are a part of this zone. The lower boundary of the ASHRAE winter comfort zone forms the lower boundary of this comfort zone. The reason that the

lower boundary in this diagram is ET* = 73°F (22.8°C) whereas ET* = 68°F (20°C) in ASHRAE's winter comfort zone is that a lower clothing insulation of 0.6 clo is used here.

3. *Uncomfortable zone.* In this zone, 86°F < ET* ≤ 95°F (30°C ≤ ET* ≤ 35°C) and $0.25 < w_{sk} \leq 0.45$. Thermal equilibrium also exists between the occupant and the environment, and the evaporative heat loss due to regulatory sweating dominates. Heart rate shows a range between 87 and 100. The occupant feels uncomfortable, i.e., warm or hot, when his or her physiological parameters are in these ranges.

4. *Very uncomfortable zone.* In this zone, 95°F < ET* < 106.5°F (35°C < ET* < 41.4°C) and $0.45 < w_{sk} < 1$. Although thermal equilibrium is still maintained with zero heat storage at the skin and the body core, there is a danger of a heat stroke when ET* > 95°F (35°C). Toward the upper boundary of this zone, the skin surface is nearly entirely wet, and the heart rate exceeds 120. Under these conditions, the occupant will feel very hot and very uncomfortable.

5. *Body heating zone.* When ET* = 106.5°F (41.4°C) and $w_{sk} = 1$, thermal regulation by evaporation fails. At a higher ET* or w_{sk}, the environment is intolerable, and the temperatures of the body core and skin tend to rise gradually.

For an air conditioned space with an occupant at low activity levels ($M < 2$ met), the indoor environment is usually maintained within the comfort zone, and the physiological and thermal responses of the occupant are also in the comfort zone. Only at higher activity levels do the thermal responses occasionally fall into the discomfort zone.

4.8 INDOOR AIR TEMPERATURE AND AIR MOVEMENTS

Comfort Air Conditioning Systems

For comfort air conditioning systems, most occupants have a metabolic rate of 1.0 to 1.5 met. The indoor clothing insulation in summer is usually 0.35 to 0.6 clo, and in winter it is 0.8 to 1.2 clo. Relative humidity has a lesser influence on thermal comfort, and will be discussed in the next section, but indoor air temperature and air velocity are discussed here.

Many researchers have conducted tests to determine the effects of airspeed on the preferred indoor air temperature and the thermal comfort of occupants. The relationship between the preferred indoor air temperatures and various airspeeds is presented in Fig. 4.8. Most of the data were taken under these conditions: metabolic rate $M = 400$ Btu/h (117 W), clothing insulation $R_{cl} = 0.6$ clo, $T_{rad} = T_a$, and relative humidity of the indoor air $\varphi = 50$ percent. The one exception is the students in Holzle's experiments, who had 0.54 clo for summer and 0.95 clo for winter. Examination of Fig. 4.8 shows the following:

- Higher indoor air temperature requires greater indoor air velocity to provide thermal comfort.
- Variation of airspeed has a greater influence on preferred indoor air temperature at lower air temperatures.

ANSI/ASHRAE Standard 55-1992 recommended that within the thermally acceptable temperature ranges in the ASHRAE summer and winter comfort zones discussed in Sec. 4.7, there be no minimum airspeed (nondirectional) that is necessary for thermal comfort.

If temperature is increased above the level allowed for the comfort zone, means must be provided to elevate the airspeed. For instance, when indoor air temperature $T_a = T_{rad}$ (mean radiant temperature), given that the airspeed for a summer comfort zone of 79°F (26°C) is 40 fpm (0.2 m/s), if T_a has an increase of 2°F (1.1°C) from 79 to 81°F (26 to 27.2°C), according to ANSI/ASHRAE Standard 55-1992, there must be a relevant increase of airspeed of about 70 fpm (0.35 m/s).

Before specifying T_a for summer conditions, one needs to determine whether occupants are likely to wear suit jackets, such as members of a church congregation or guests in a multipurpose

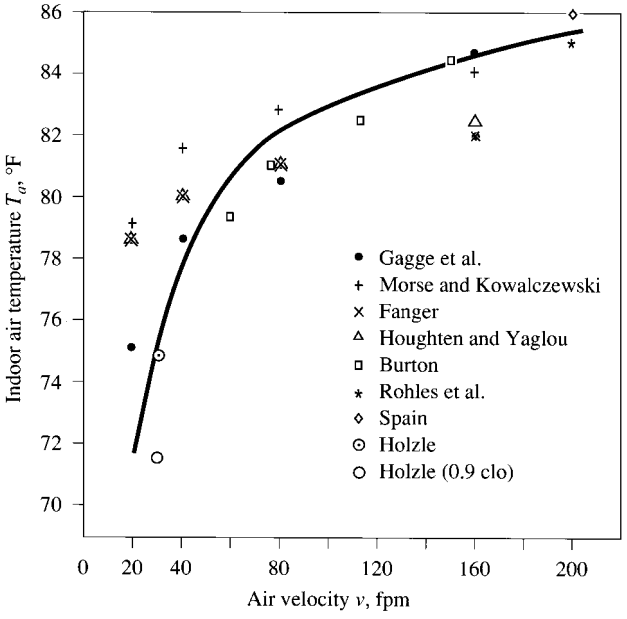


FIGURE 4.8 Preferred indoor air temperatures at various air velocities.

hall. In such cases, a reduction of 4°F of summer optimal ET* may be necessary because of the increase in clothing insulation of about 0.4 clo.

Design Considerations

When one is specifying indoor design conditions, thermal comfort must be provided at optimum cost while using energy efficiently. These principles should be considered:

1. To determine the optimum summer and winter indoor design temperatures, consider the local clothing habits and the upper and lower acceptable limits on clothing insulation at various operative temperatures T_o, as shown in Fig. 4.9.
2. It is always more energy-efficient to use different indoor design temperatures for summer and winter than a year-round constant value. An unoccupied-period setback during winter always saves energy. There are also buildings in which a constant indoor temperature is required for the health and comfort of the occupants, such as in many health care facilities.
3. For short-term occupancies, or when the metabolic rate is higher than 1.2 met, a strategy of using a lower energy-use ceiling fan or a wall-mounted fan to provide higher air velocity may be considered. Thus a higher indoor design temperature within the extended summer comfort zone may be acceptable for occupants, especially in industrial settings.

Indoor Design Temperatures for Comfort Air Conditioning

According to ANSI/ASHRAE Standard 55-1992, *Thermal Environmental Conditions for Human Occupancy*, and ASHRAE/IES 90.1-1999, *Energy-Standard for Buildings Except Low-Rise*

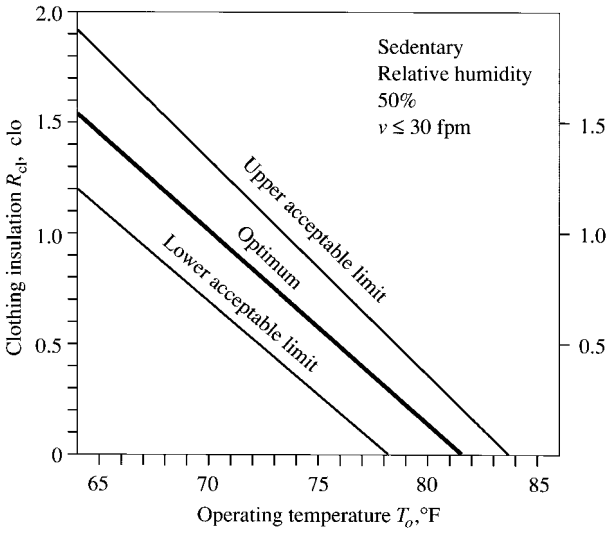


FIGURE 4.9 Relationship between clothing insulation R_{cl} and operating temperature T_o. *(Source: ASHRAE Transactions 1983, Part I B. Reptinted with permission.)*

Residential Buildings, the following indoor design temperature and air speed apply for comfort air conditioning systems when the activity level is 1.2 met, there is a relative humidity of 50 percent in summer, mean airspeed $\leq$ 30 fpm (0.15 m/s), and $T_a = T_{rad}$:

	Typical clothing insulation, clo	Optimum operative temperature	Indoor design temperature range
Winter	0.9	71°F (22°C)	69–74°F (20.5–23.5°C)
Summer	0.5	76°F (24.5°C)	74–79°F (23.5–26°C)

According to ANSI/ASHRAE Standard 55-1992, "within the thermally acceptable temperature ranges, there is no minimum air speed that is necessary for thermal comfort." If the summer indoor temperature is 79°F (26°C), an airspeed of 40 fpm (0.2 m/s) is recommended. If $T_a > 79$°F (26°C), a relevant increase of airspeed in the indoor occupied zone should be considered. Refer to ANSI/ASHRAE Standard 55-1992.

If the space relative humidity can be lowered to 35 to 40 percent in summer, then the higher limit 78°F (25.5°C) is often specified.

Indoor badminton and table tennis tournament arenas should have air velocities below 30 fpm (0.15 m/s). To avoid nonuniformity and to prevent local discomfort, the air temperature difference between 4 in. from the floor and 67 in. above the floor should not exceed 5°F (3°C). The radiant temperature asymmetry in the vertical direction should be less than 9°F (5°C), and in the horizontal direction less than 18°F (10°C).

To determine whether the specified air temperature and airspeed are met, they should be measured at 4-, 24-, and 43-in. (0.1-, 0.6-, and 1.1-m) levels for sedentary occupants, and 4-, 43-, and 67-in. (0.1-, 1.1-, and 1.7-m) levels for standing activity. The duration for determining the mean value of the air movement should be 3 min. or 30 times the 90 percent response time of the measuring instrument, whichever is greater.

Process Air Conditioning Systems

For process air conditioning systems, indoor design temperature is usually based on previous experiences. For precision manufacturing projects, a basic temperature plus a tolerance, such as 72 ± 2°F (22.2 ± 1.1°C) for a precision machinery assembling workshop, is often specified. First, the tolerance should be neither too tight nor too loose. Second, either the temperature fluctuation at various times within the working period or the temperature variation within the working space, or both, should be included in this tolerance.

In unidirectional-flow clean rooms, the velocity of the airstream in the working area is often specified as 90 ± 20 fpm (0.45 ± 0.1 m/s) to prevent contamination of the products.

4.9 HUMIDITY

Comfort Air Conditioning Systems

According to ANSI/ASHRAE Standard 55-1992, for the zone occupied by people engaged in light, primarily sedentary activity (≤ 1.2 met), the relative humidity should conform with the limits of ASHRAE winter and summer comfort zones, as shown in Fig. 4.6. These limits are intended to maintain acceptable thermal conditions for the occupants' comfort.

ASHRAE/IESNA Standard 90.1-1999 mandates that where a zone is served by system(s) with both humidification or dehumidification capacity, means shall be provided to prevent simultaneous operation of humidification and dehumidification equipment. Standard 90.1-1999 also specifies that where humidistatic controls are provided, such controls shall be capable of preventing reheating, mixing of hot and cold air streams, and simultaneous heating and cooling. Refer to Section 29.12 for more details.

The following results should be considered during the design and evaluation of the performance of comfort air conditioning systems:

- Maintaining an indoor space relative humidity φ_r between 20 and 30 percent in winter prevents or reduces the condensation at the inner side of the window glass.
- In high-occupancy applications, it may be economical and is still comfortable to specify the summer indoor relative humidity at 55 to 60 percent if the indoor temperature is within the summer comfort zone.
- The indoor relative humidity at part load may be considerably higher than at full load in some air conditioning systems in summer.
- When the indoor relative humidity is below 25 percent, the incidence of respiratory infections increases significantly. If, simultaneously, indoor temperatures are low, such as below 70°F (21°C), the induced static electricity in carpeted rooms may cause uncomfortable shocks to occupants contacting metal furniture or decorations.
- Because the increase in the outdoor ventilation rates in ASHRAE Standard 62-1999 for the air conditioning system serves the building located in areas where the humidity ratio of outdoor air is very low during winter, the system has greater difficulty maintaining an indoor relative humidty of 20 to 30 percent without winter humidification.
- If a humidifier is installed, its humidifying capacity should not exceed the actual humidifying requirements so that wet surfaces do not occur inside the air-handling unit, packaged unit, and supply ducts. Wet surface and dirt cause the growth of microorganisms and poor indoor air quality. Therefore, for comfort air conditioning systems, the recommended indoor relative humidity

levels are as follows:

	Relative humidity, %
Summer	30–65
Winter	
Commercial and public buildings	20–60
Health care buildings	30–60

Process Air Conditioning Systems

Humidity affects the physical properties of many materials and, therefore, their manufacturing processes.

Moisture Content. Relative humidity has a marked influence on the moisture content of hygroscopic materials such as natural textile fibers, paper, wood, leather, and foodstuffs. Moisture content affects the weight of the products and sometimes their strength, appearance, and quality.

Dimensional Variation. Hygroscopic materials often extend at higher relative humidity and contract at lower humidity. A 2 percent increase in moisture content may result in a 0.2 percent increase in dimension of paper. That is why lithographic printing requires a relative humidity of 45 ± 2 percent.

Corrosion and Rust. Corrosion is an electrochemical process. Moisture encourages the formation of electrolytes and therefore the corrosion process. A relative humidity greater than 50 percent may affect the smooth operation of bearings in precision instruments. When indoor relative humidity exceeds 70 percent, rust may be visible on the surface of the machinery and on parts made of steel and iron.

Static Electricity. Static electricity may cause minute particles to repel or attract one another, which is detrimental to many manufacturing processes. Static electricity charges minute dust particles, in the air, causing them to cling to equipment and work surfaces. Static electricity exists in an indoor environment at normal air temperatures when relative humidity is less than about 40 percent.

Loss of Water. Vegetables and fruits lose water vapor through evaporation from their surfaces during storage. Low temperatures and high relative humidities, such as $\varphi = 90$ to 98 percent, may reduce water loss and delay desiccation.

It is important to specify the exact relative humidity required for product quality and cost control. For process air conditioning systems, the specified relative humidity is either a year-round single value or a range. A strict relative-humidity requirement always includes a basic value and a tolerance, such as the relative humidity for lithographic printing mentioned before. When temperature and relative humidity controls are both required, they should be specified as a combination. Consider this example:

	Temperature, °F (°C)	Relative humidity, %
Clean room	72 ± 2 (22.2 ± 1.1)	45 ± 5

Case Study 4.1. A factory workshop has the following environmental parameters during summer:

Indoor air temperature	79°F (26.1°C)
Indoor air relative humidity	50 percent

Space air velocity	20 fpm (0.1 m/s)
Clothing insulation of the workers	0.5 clo
Activity level	3 met

As a result of comfort complaints by personnnel, you are asked to recommend effective and economical corrective measures to improve the indoor environment. The following table includes the information required during analysis:

Area ratio of the clothed body f_{cl}	1.2
Permeability index of clothing i_m	0.4
Skin surface temperature, T_{sk}	92.7°F (33.7°C)
Saturation water vapor pressure at 92.7°F	0.764 psia (5.27 kPa abs.)
at 79°F	0.491 psia (3.39 kPa abs.)

Solution

1. When the space air velocity $v = 20$ fpm, the convective heat-transfer coefficient can be calculated, from Eq. (4.9), as

$$h_c = 0.0681v^{0.69} = 0.0681 \times 20^{0.69} = 0.538 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

Because at normal indoor conditions the radiative heat-transfer coefficient $h_r = 0.83$ Btu/h · ft^2 · °F, the surface heat-transfer coefficient h is

$$h = h_c + h_r = 0.538 + 0.83 = 1.368 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

The clothing insulation $R_{cl} = 0.5 \times 0.88 = 0.44$ h · ft^2 · °F/Btu. From Eq. (4.12), the clothing efficiency is found to be

$$F_{cl} = \frac{1}{R_{cl} f_{cl} h + 1} = \frac{1}{0.44 \times 1.2 \times 1.368 + 1} = 0.5806$$

Therefore, from Eq. (4.11), we find the sensible heat loss from the skin surface of the worker

$$\begin{aligned} C + R &= F_{cl} f_{cl} h(T_{sk} - T_o) = 0.5806 \times 1.2 \times 1.368(92.7 - 79) \\ &= 13.06 \text{ Btu/h} \cdot \text{ft}^2 \end{aligned}$$

From Eq. (4.17), the overall sensible heat-transfer coefficient h_s is

$$\begin{aligned} h_s &= \frac{f_{cl} h}{f_{cl} h R_{cl} + 1} \\ &= \frac{1.2 \times 1.368}{1.2 \times 1.368 \times 0.44 + 1} = 0.9531 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F} \end{aligned}$$

From Eq. (4.16), the overall evaporative heat-transfer coefficient $h_{e,c}$ is

$$h_{e,c} = h_s i_m \text{ LR} = 0.9531 \times 0.4 \times 205 = 78.16 \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{psi}$$

The maximum evaporative heat loss E_{max} due to regulatory sweating can be calculated from Eq. (4.15) as

$$\begin{aligned} E_{max} &= h_{e,c}(p_{sk,s} - p_a) = 78.16(0.764 - 0.5 \times 0.491) \\ &= 40.24 \text{ Btu/h} \cdot \text{ft}^2 \end{aligned}$$

In Eq. (4.24),

$$0.85M = 0.85 \times 3 \times 18.46 = 47.07 \text{ Btu/h} \cdot \text{ft}^2$$

When $\mu = 0.05$ and $E_{res} = 0.1M$, the total evaporative loss from the skin surface of the worker, from Eq. (4.1), is

$$E_{sk} = 0.85M - (C + R) = 47.07 - 13.06 = 34.01 \text{ Btu/h} \cdot \text{ft}^2 \ (107.3 \text{ W/m}^2)$$

Therefore, the total skin wetness is

$$w_{sk} = \frac{E_{sk}}{E_{max}} = \frac{34.01}{40.24} = 0.845$$

In Fig. 4.7, when $w_{sk} = 0.845$, a person is very uncomfortable.

2. From Fanger's comfort chart, shown in Fig. 4.5, for a person with an activity level of 3 met and a clothing insulation of 0.5 clo, at a relative humidity of 50 percent and an air velocity of 20 fpm and with $T_a = T_{rad}$ for neutral thermal sensation, the indoor air temperature should be 56°F (13.3°C). Obviously, this is not economical because too much refrigeration is required.

3. Let us analyze the results if ceiling fans or wall fans are used to increase the space air velocity v to 300 fpm (1.5 m/s). Then the convective heat-transfer coefficient h_c is

$$h_c = 0.681 \times 300^{0.69} = 3.486 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

The surface heat-transfer coefficient is

$$h = 3.486 + 0.83 = 4.316 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Also the clothing efficiency is calculated as

$$F_{cl} = \frac{1}{0.44 \times 1.2 \times 4.316 + 1} = 0.305$$

Then the sensible heat loss is equal to

$$C + R = 0.305 \times 1.2 \times 4.316(92.7 - 79) = 21.64 \text{ Btu/h} \cdot \text{ft}^2$$

The overall sensible heat-transfer coefficient is

$$h_s = \frac{1.2 \times 4.316}{1.2 \times 4.136 \times 0.44 + 1} = 1.58 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

The overall evaporative heat-transfer coefficient $h_{e,c}$ can be shown to be

$$h_{e,c} = 1.58 \times 0.4 \times 205 = 129.5 \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{psi}$$

Then the maximum evaporative heat loss due to regulatory sweating is

$$E_{max} = 129.5(0.7604 - 0.5 \times 0.491) = 66.68 \text{ Btu/h} \cdot \text{ft}^2 \ (210.3 \text{ W/m}^2)$$

The total skin wetness is

$$w_{sk} = 47.07 - 21.64/66.68 = 0.381$$

That is, w_{sk} has been greatly reduced compared with the value at $v = 20$ fpm (0.1 m/s). In Fig. 4.7, $w_{sk} = 0.381$ is in the uncomfortable zone. Workers will feel warm, but the indoor environment has been considerably improved. This may be the most cost-effective solution.

4.10 SICK BUILDING SYNDROME AND INDOOR AIR QUALITY

Sick building syndrome is a kind of building-related illness that has received public attention since the 1970s. ASHRAE (1987) defined the *sick building* as ". . . a building in which a significant number (more than 20 percent) of building occupants report illness perceived as being building related. This phenomenon, also known as 'sick building syndrome' is characterized by a range of symptoms including, but not limited to, eye, nose, and throat irritation, dryness of mucous membranes and skin, nose bleeds, skin rash, mental fatigue, headache, cough, hoarseness, wheezing, nausea, and dizziness. Within a given building, there usually will be some commonality among the symptoms manifested as well as temporal association between occupancy in the building and appearance of symptoms." If there are signs of actual illnesses, these illness are classified as building-related illnesses.

Poor *indoor air quality* (IAQ) is the dominant factor that causes sick building syndrome. Indoor air quality is defined as an indication of harmful concentrations of the indoor air contaminants that affect the health of the occupants or the degree of satisfaction of a substantial majority (80 percent or more) of occupants exposed to such an indoor environment. Poor control of the indoor air temperature and relative humidity are causes of discomfort symptoms. They may also increase the indoor air contaminants. However, unsatisfactory indoor temperature and indoor relative humidity are only indirect causes of poor indoor air quality.

National Institute for Occupational Safety and Health (NIOSH) of the United States (1989), according to the results of 529 building investigations between 1971 and 1988, and Health and Welfare Canada (HWC), according to the results of 1362 building investigations between 1984 and 1989, classified the reasons for sick building syndrome as follows:

	NIOSH, 529 Buildings		HWC, 1362 Buildings	
	No. of buildings	Percent	No. of buildings	Percent
Inadequate ventilation	280	53	710	52
Indoor contaminants	80	15	165	12
Outdoor contaminants	53	10	125	9
Biological contaminant	27	5	6	0.4
Building fabric contamination	21	4	27	2
Unknown sources	68	13	329	24

Inadequate ventilation includes lack of outdoor air, poor air distribution, poor thermal control, and inadequate maintenance; and it is the primary cause of indoor air quality. The survey found 70 to 80 percent of the investigated buildings had no known problems. Effective operation and control of the HVAC&R system will be discussed in later chapters.

In the United States, most people spend about 90 percent of their time indoors. The purpose of specifying the indoor design conditions in the design documents is to provide the occupants with a satisfactory indoor environment at optimum cost.

After the energy crisis in 1973, a lower outdoor ventilation rate, a tighter building shell, and the use of variable-air-volume (VAV) systems at part-load operation may reduce the amount of outdoor air intake significantly. Indoor air quality therefore has become one of the critical HVAC&R problems especially in commercial buildings since the 1980s.

Indoor Air Contaminants

Based on the results of the field investigations of three office buildings by Bayer and Black in 1988, the indoor air contaminants that relate to indoor air quality and the symptoms of the sick building syndrome are mainly the following:

1. *Total particulate concentration*. This parameter includes particulates from building materials, combustion products, and mineral and synthetic fibers. In February 1989, the U.S. Environmental Protection Agency (EPA) specified the allowable indoor concentration level of particulates of 10 μm and less in diameter (which can penetrate deeply into the lungs, becoming hazardous to health) as follows:

$$50\ \mu g/m^3\ (0.000022\ gr/ft^3): \quad 1\ year$$

$$150\ \mu g/m^3\ (0.000066\ gr/ft^3): \quad 24\ h$$

According to *ASHRAE Handbook 1997, Fundamentals*, particles less than 2 μm in diameter are most likely retained in the lungs, and particles less than 0.1 to 0.5 μm in diameter may leave the lungs with the exhaled breath. Particles larger than 8 to 10 μm in diameter are separated and retained in the upper respiratory tract. Particles between 2 and 8 μm in diameter are deposited mainly in the conducting airways of the lungs and are swallowed or coughed out quickly.

2. *Combustion products*. Carbon monoxide (CO) is a colorless, odorless gas, a product of incomplete combustion. CO interferes with the delivery of oxygen throughout the body. NO_2 is a combustion product from gas stoves and other sources. There is growing evidence that NO_2 may cause respiratory disease. Indoor concentrations for CO and NO_2 are the same as specified in the *National Primary Ambient-Air Quality Standard* by the EPA later in this section.

3. *Volatile organic compounds (VOCs)*. These include formaldehyde and a variety of aliphatic, aromatic, oxygenated, and chlorinated compounds. Mucous membrane irritation caused by formaldehyde is well established. U.S. Department of Housing and Urban Development (HUD) specifies a target level of indoor concentration of formaldehyde for manufacturing homes of 0.4 ppm.

4. *Nicotine*. Environmental tobacco smoke is clearly a discomfort factor to many adults who do not smoke. Nicotine and other components of tobacco smoke are also a health risk for human beings.

5. *Radon*. Radon is a colorless, odorless, inert radiative gas widely found in soil, rocks, and water, created by the decay of the radium and uranium. It travels through the pores of rock and soil and infiltrates into a house along cracks and other openings in the basement slab or walls; pressure-driven radon containing soil gas is caused by thermal stack, wind, and the mechanical ventilation system.

The annual average concentration of radon in residential buildings in the United States is about 1.25 pCi/L. Only about 6 percent of U.S. homes have an annual average radon concentration exceeding the EPA recommended annual average indoor concentration of 4 pCi/L for residential and school occupancies. Pressure-driven flow of radon containing soil gas is the primary source for elevated concentrations. At various locations in the United States, the indoor radon concentrations may vary hourly, daily, and seasonally, sometimes by as much as a factor of 10 to 20 on a daily basis. The radiative decay of radon produces a series of radioactive isotopes called *progeny*. These progeny are chemically active. They can deposit directly into the lung, or attach to airborne particles and then deposit into the lung. Some of the progeny are alpha particle emitters and may lead to cellular changes and initiate lung cancer.

6. *Occupant-generated contaminants and odors*. These include odors and emissions (bioeffluents) from the human body, particulates, and other contaminants.

7. *Bioaerosols.* These contaminants include bacteria, mold and midew, viruses, and pollens. Bacteria and viruses are airborne, carried by dust or transmitted by people and animals; standing water (wet surface) and dirt (nutrients) can become the breeding ground for mold, mildew, and other biological contaminants. Pollens originate from plants.

In addition to the preceding indoor air contaminants, others such as sulfur dioxide and ozone can have a significant effect on occupants. Carbon dioxide (CO_2) is a kind of gas released from human beings and is not an indoor contaminant at the concentrations found in most buildings. ASHRAE Standard 62-1999 specifies guidelines for indoor concentration for ozone during continuous exposure time as 100 $\mu g/m^3$.

Basic Strategies to Improve Indoor Air Quality

There are three basic strategies to improve indoor air quality: control the contaminated source, remove air contaminants from the indoor air by air cleaner, and use outdoor ventilation air to dilute the concentrations of indoor air contaminants.

To eliminate or to reduce the emissions of air contaminants from the contaminated source is often the most effective way to improve the IAQ as well as periodically cleaning the duct's interior surface and coil's condensate pan and using building materials and carpets that do not release or release only negligible volatile organic compounds and dust. However, emissions and odor released from occupants are difficult to eliminate or to reduce, and smoking is now prohibited in many public places and limited to specified areas in many commercial buildings.

The volatile organic compounds and combustion products contain numerous minute particles of size between 0.003 and 1 μm. Only high-efficiency air filters and activated carbon filters can remove these minute particles and odors from the airstream effectively. High efficiency air filters and carbon filters are expensive to install, operate, and maintain.

Adequate outdoor ventilation air to dilute the air contaminations in practice has been proved an essential, practical, and cost-effective means to improve the indoor air quality. ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality*, specifies two alternative procedures to obtain acceptable IAQ: the ventilation rate procedure and indoor air quality procedure.

- In the ventilation rate procedure, acceptable indoor air quality is achieved by providing ventilation air of specified quality and quantity to the space.
- In the IAQ procedure, acceptable air quality is achieved within the space by controlling known and specifiable contaminants.

The ventilation rate and IAQ procedurs are discused again in Sec. 23.2.

ASHRAE Standard 62-1999 defined *acceptable indoor air quality* as air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80 percent or more) of the people exposed do not express dissatisfaction.

ASHRAE Standard 62-1999 defines *ventilation air* as that portion of supply air that is outdoor air plus any recirculated air that has been treated for the purpose of maintaining acceptable indoor air quality. Ventilation is the process of supplying and removing ventilation by natural and mechanical means, and the ventilation rate means the rate of ventilation air supplied to the conditioned space through the air system.

If outdoor air is used to dilute the concentration of indoor contaminants, its quality must meet the *National Primary Ambient-Air Quality Standard* provided by the EPA. Part of the time average concentrations are shown below:

	Long-term concentration			Short-term concentration		
Pollutants	$\mu g/m^3$	ppm	Average period of exposure	$\mu g/m^3$	ppm	Average period of exposure
Particulate matter	50		1 year	150		24 hours
Sulfur oxides	80	0.03	1 year	365	0.14	24 hours
Carbon monoxide				40,000	35	1 hour
				10,000	9	8 hours
Nitrogen dioxide	100	0.055	1 year			
Oxidants (ozone)				235	0.12	1 hour
Lead	1.5		3 months			

Only particulate matter is expressed in annual geometric means; the other two are annual arithmetic means. For carbon monoxide and ozone, both values are not to be exceeded more than once a year.

Outdoor Air Requirements for Occupants

For both comfort and process air conditioning systems, outdoor air is required to do the following:

- To meet metabolic requirements of the occupants
- To dilute the indoor air contaminants, odors, and pollutants to maintain an acceptable indoor air quality
- To support any combustion process or replace the amount of exhaust air required in laboratories, manufacturing processes, or restrooms
- To provide makeup for the amount of exfiltrated air required when a positive pressure is to be maintained in a conditioned space

The amount of outdoor air required for metabolic oxidation processes for occupants is actually rather small. ASHRAE Standard 62-1999 noted that where only dilution ventilation is used to control indoor air quality, a CO_2 indoor-to-outdoor differential concentration ($C_{i,CO_2} - C_o$) is not greater than about 700 ppm, and the CO_2 production $\dot{V}_{CO_2}$ of a sedentary occupant who is eating a normal diet is 0.0106 cfm (0.3 L/min). The amount of outdoor air required for each indoor occupant $\dot{V}_{o,oc}$ can be calculated as

$$\dot{V}_{o,oc} = \frac{\dot{V}_{CO_2}}{C_{i,CO_2} - C_o} = \frac{0.0106}{0.0007}$$

$$\approx 15 \text{ cfm } (7 \text{ L/s}) \tag{4.26}$$

Thayer (1982), using data from different authors, developed a dilution index that indicates 15 cfm (7 L/s) of outdoor air per person will satisfy more than 80 percent of the occupants in the space.

Usually, for comfort air conditioning systems, the same outdoor air used for the dilution of the concentration of air contaminants including human bioeffluents is sufficient for the metabolic oxygen requirement, for exhausting air from restrooms, and for replacing exfiltrated air lost from the conditioned space as a result of positive pressure. If the outdoor air supply is used to dilute the concentration of a specific indoor air contaminant, the rate of outdoor air supply $\dot{V}_o$, in cfm (L/min), can be calculated as

$$\dot{V}_o = \frac{2118\dot{m}_{par}}{C_i - C_o} \tag{4.27}$$

where $\dot{m}_{par}$ = rate of generation of contaminants in space, mg/s
C_i, C_o = concentrations of air contaminants indoors and outdoors, respectively, mg/m^3

Values of C_o can be found from the EPA *National Primary Ambient-Air Quality Standards*. The indoor concentration of CO_2 and other contaminants should meet the specified value as stated before. Some of the outdoor air requirements for ventilation, often called the ventilation rate, specified in ASHRAE Standard 62-1999 are indicated in Table 4.5. For clean rooms, Federal Standard 209B specifies the rate of outdoor air, or makeup air, to be 5 to 20 percent of the supply air.

When only dilution ventilation is used to control indoor air quality and CO_2 is used as an indicator of human bioeffluents, ASHRAE Standard 62-1999 noted that an indoor-outdoor differential concentration not greater than about 700 ppm of CO_2 indicates that comfort (odor) criteria related to bioeffluents are likely to be satisfied. The CO_2 concentrations in outdoor air typically are between 300 to 350 ppm. Using CO_2 as an indicator of bioeffluents does not elimnate the need for consideration of other contaminants.

The refrigeration capacity required to cool and dehumidify the outdoor air can be a major por-

TABLE 4.5 Outdoor Air Requirements Recommended by ASHRAE Standard 62–1999

Application	cfm/person
Dining room	20
Bar and cocktail lounges	30
Conference rooms	20
Office spaces	20
Office conference rooms	20
Retail stores	0.2–0.3*
Beauty shops	25
Ballrooms and discos	25
Spectator areas	15
Theater auditoriums	15
Transportation waiting rooms	15
Classrooms	15
Hospital patient rooms	25
Residences	0.35†
Smoking lounges	60

*cfm/ft^2 floor area
†Air changes/h
Source: Abridged with permission from ASHRAE Standard 62-1999.

tion of the total refrigeration requirement during summer, depending upon the occupant density and the amount of exhaust air. The amount of infiltration depends on the wind speed and direction, as well as the outdoor and indoor temperatures and pressure differences, which are variable. Therefore, infiltration is not a reliable source of outdoor air supply. Infiltration cannot replace specified outdoor air ventilation requirements.

4.11 AIR CLEANLINESS

The manufacturing process of semiconductors, pharmaceutical, aerospace, and operating rooms in health care facilities need clean indoor environment, clean spaces, and clean rooms. A clean room is a constructed enclosed area in which air cleanliness is expressed in terms of particle count of air contaminants and in which the associated temperature, humidity, air pressure, and lighting are controlled within specific limits. A clean space is a defined area in which air cleanliness and environmental conditions are controlled within specific limits.The quality of their manufactured products is closely related to the size and number of particulates contained in the space air. An indoor air quality of allowable total particulate annual-average concentration of 50 $\mu g/m^3$ cannot meet the air cleanliness requirements. Therefore, Federal Standard (FS) 209E specifies the following classes for clean spaces and clean rooms:

Class 1. Particle count not to exceed 1 particle/ft^3 (35 particles/m^3) of a size of 0.5 μm and larger, with no particle exceeding 5 μm.

Class 10. Particle count not to exceed 10 particles/ft^3 (353 particles/m^3) of a size of 0.5 μm and larger, with no particle exceeding 5 μm.

Class 100. Particle count not to exceed 100 particles/ft^3 (3531 particles/m^3) of a size of 0.5 μm and larger.

Class 1000. Particle count not to exceed 1000 particles/ft^3 (35,315 particles/m^3) of a size of 0.5 μm and larger.

Class 10,000. Particle count not to exceed 10,000 particles/ft^3 (353,150 particles/m^3) of a size

of 0.5 μm and larger or 65 particles/ft^3 (2295 particles/ft^3) of a size 5.0 μm and larger.

Class 100,000. Particle count not to exceed 100,000 particles/ft^3 (3,531,500 particles/m^3) of a size of 0.5 μm and larger or 700 particles/ft^3 (24,720 particles/ft^3) of a size 5.0 μm and larger.

Refer to Federal Standard 209E for more details.

Since workers in these clean rooms wear protective gowns and hats, in order to provide these air cleanliness classes, a year-round constant temperature and associated relative humidity and a specified unidirectional airflow should be maintained. High-efficiency particulate air (HEPA) filters and ultra-low-penetration air (ULPA) filters should be installed in the air conditioning systems. Special building materials with hard and clean surfaces should be used as the building envelope. More details are covered in Chap. 30.

4.12 SOUND LEVEL

Sound and Sound Level

Sound can be defined as a variation in pressure due to vibration in an elastic medium such as air. A vibrating body generates pressure waves, which spread by alternate compression and rarefaction of the molecules within the transmitting medium. Airborne sound is a variation of air pressure, with atmospheric pressure as the mean value. Because sound is transmitted by compression and expansion of molecules, it cannot travel in a vacuum. The denser the material, the faster the traveling speed of a sound wave. The velocity of a sound wave in air is approximately 1130 fps. In water, it is about 4500 fps and in steel 15,000 fps.

Noise is any unwanted sound. In air systems, noise should be compensated for, either by attenuation (the process of reducing the amount of sound that reaches the space) or by masking it with other, less objectionable sounds.

Sound Power Level and Sound Pressure Level

Sound power is the ability to radiate power from a sound source excited by an energy input. The intensity of sound power is the power output from a sound source expressed in watts (W). Because of the wide variation of sound output—from the threshold hearing level of 10^{-12} W to a level of 10^8 W, generated by the launching of a Saturn rocket, a ratio of 10^{20} to 1—it is more appropriate and convenient to use a logarithmic expression to define sound power level, i.e.,

$$L_w = 10 \log \frac{w}{10^{-12}\,\text{W}} \text{ re 1 pW} \tag{4.28}$$

where L_w = sound power level, dB
w = sound source power output, W

Here 10^{-12}W, or 1 pW (picowatt), is the international reference base, and re indicates the reference base.

The human ear and microphones are pressure-sensitive. Analogous to the sound power level, the sound pressure level is defined as

$$L_p = 20 \log \frac{p}{2 \times 10^{-5}\,\text{Pa}} \text{ re 20 μPa} \tag{4.29}$$

where L_p = sound pressure level, dB
p = sound pressure, Pa

TABLE 4.6 Typical Sound Pressure Levels

Source	Sound pressure, Pa	Sound pressure level dB re 20 μPa	Subjective reaction
Military jet takeoff at 100 ft	200	140	Extreme
Passenger's ramp at jet airliner (peak)	20	120	Threshold of pain
Platform of subway station (steel wheels)	2	100	
Computer printout room*	0.2	80	
Conversational speech at 3 ft	0.02	60	
Window air conditioner*	0.006	50	Moderate
Quiet residential area*	0.002	40	
Whispered conversation at 6 ft	0.0006	30	
Buzzing insect at 3 ft	0.0002	20	
Threshold of good hearing	0.00006	10	Faint
Threshold of excellent youthful hearing	0.00002	0	Threshold of hearing

*Ambient.
Source: Abridged from *ASHRAE Handbook 1989, Fundamentals*. Reprinted by permission.

Here the reference sound pressure level is 2×10^{-5} Pa (pascal), or 20 μPa, corresponding to the hearing threshold. Because sound power is proportional to the square of the sound pressure, 10 log $p^2 = 20 \log p$. Sound pressure levels of various sources are listed in Table 4.6.

The sound power level of a specific source is a fixed output. It cannot be measured directly and can only be calculated through the measurement of the sound pressure level. On the other hand, sound pressure level is the sound level measured at any one point and is a function of distance from the source and characteristics of the surroundings.

Octave Bands

Sound waves, like other waves, are characterized by the relationship between wavelength, speed, and frequency:

$$\text{Wavelength} = \frac{\text{speed}}{\text{frequency}} \tag{4.30}$$

People can hear frequencies from 20 Hz to 20 kHz. To study and analyze sound, we must break it down into components. A convenient way is to subdivide the audible range into eight octave bands or sometimes 24 $\frac{1}{3}$- octave bands. An octave is a frequency band in which the frequency of the upper band limit is double the frequency of the lower band limit. The center frequency of an octave or a $\frac{1}{3}$-octave band is the geometric mean of its upper and lower band limits. An octave or $\frac{1}{3}$-octave band is represented by its center frequency. The eight octave bands and their center frequencies are listed in Table 4.7.

Addition of Sound Levels

Because sound levels, in dB, are in logarithmic units, two sound levels cannot be added arithmetically. If sound levels A, B, C, . . ., in dB, are added, the combined overall sound level ΣL can be calculated as

$$\Sigma L = 10 \log (10^{0.1A} + 10^{0.1B} + 10^{0.1C} + \cdot \cdot \cdot) \tag{4.31}$$

TABLE 4.7 Octave Bands and Their Center Frequencies

Band number	Band frequency, H2		
	Lower	Center	Upper
	22.4	31.5	45
1	45	63	90
2	90	125	180
3	180	250	355
4	355	500	710
5	710	1,000	1,400
6	1,400	2,000	2,800
7	2,800	4,000	5,600
8	5,600	8,000	11,200

Source: Abridged with permission from *ASHRAE Handbook 1989, Fundamentals* with permission.

Human Response and Design Criteria

The human brain does not respond in the same way to lower frequencies as to higher frequencies. At lower sound pressure levels, it judges a 20-dB sound at 1000 Hz to have the same loudness as a 52-dB sound at 50 Hz. However, at high sound pressure levels, a 100-dB sound at 1000 Hz seems as loud as 110 dB at 50 Hz.

The purpose of noise control in an air conditioned space is to provide a background sound low enough to avoid interference with the acoustical requirements of the occupants. The distribution of the background sound should be balanced over a wide range of frequencies, without whistle, hum, rumble, or audible beats. Three types of criteria for sound control are currently used in indoor system design:

1. *A-weighted sound level dBA.* The A-weighted sound level dBA tries to simulate the response of the human ear to sound at low sound pressure levels. An electronic weighting network automatically simulates the lower sensitivity of the human ear to lower-frequency sounds by subtracting a certain number of decibels at various octave bands, such as approximately 27 dB in the first octave band, 16 dB in the second, 8 dB in the third, and 4 dB in the fourth. The A-weighted sound level gives a single value. It is simple and also takes into consideration the human judgment of relative loudness at low sound pressure levels. Its main drawback is its failure to consider the frequency spectrum or the subjective quality of sound.

2. *Noise criteria, or NC, curves.* NC curves represent actual human reactions during tests. The shape of NC curves is similar to the equal loudness contour representing the response of the human ear, as shown in Fig. 4.10. NC curves are intended to indicate the permissible sound pressure level of a broadband noise at various octave bands by a single NC curve sound level rating. NC curves are practical. They also consider the frequency spectrum of the broadband noise. The main problem with NC curves is that the shape of the curve does not approach a balanced, bland-sounding spectrum that is neither rumbly nor hissy.

3. *Room criteria, or RC, curves.* RC curves, as shown in Fig. 4.11, are similar to NC curves except that the shape of an RC curve is a close approximation of a balanced, bland-sounding spectrum.

ASHRAE recommends the indoor design RC or NC ranges presented in Table 4.8. For sounds containing significant pure tones or impulsive sounds, a 5- to 10-dB lower value should be specified. Noise is always an annoying element and source of complaints in indoor environments. Attenuation

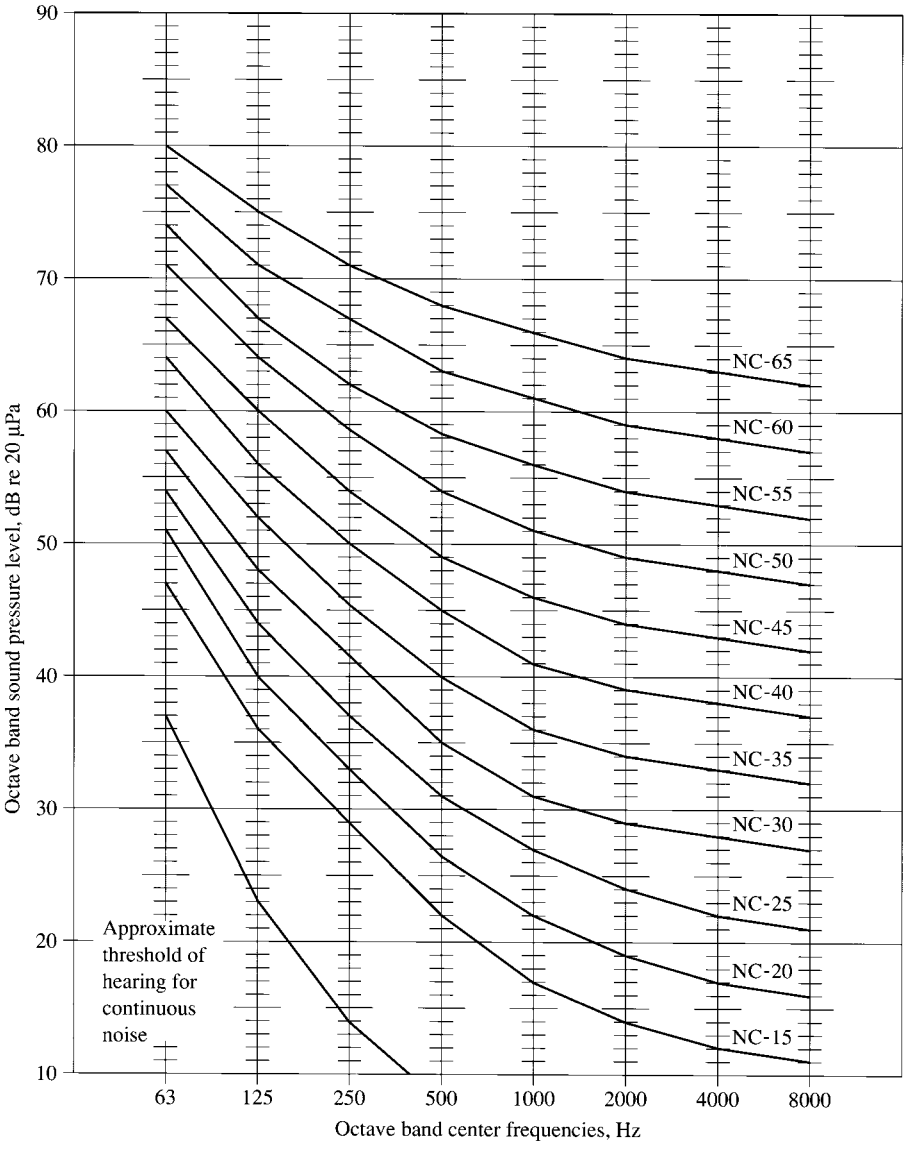


FIGURE 4.10 Noise criteria (NC) curves. *(Source: ASHRAE Handbook 1989, Fundamentals. Reprinted with permission.)*

to achieve an NC or RC goal less than 30 dB for a central air conditioning system or a packaged system is very expensive. In actual practice, the NC or RC criteria range for private residences and apartments varies greatly because of personal requirements and economic considerations.

To meet the listed design criteria range, the measured sound pressure levels of at least three of the four octave bands between 250 and 2000 Hz should be within the listed NC or RC range. In industrial workshops with machinery and equipment, Occupational Safety and Health Administration

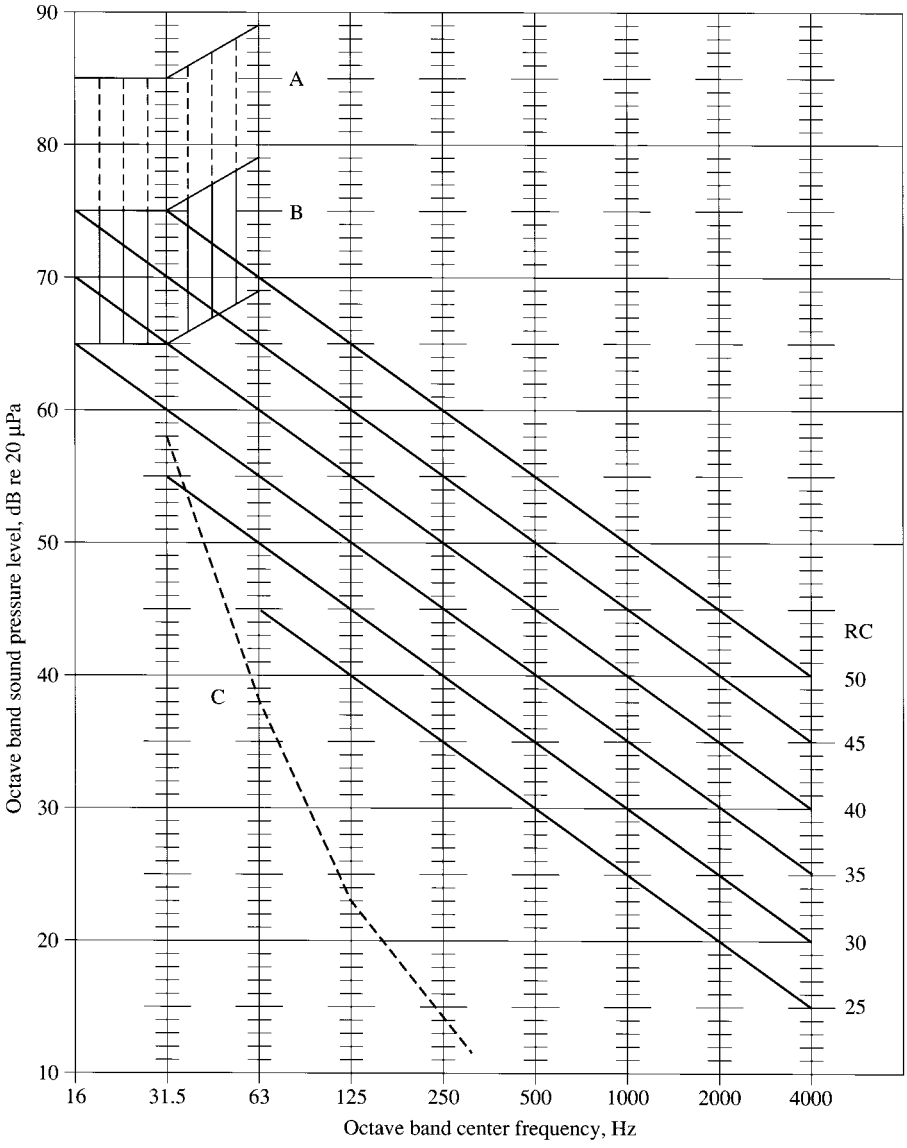


FIGURE 4.11 Room criteria (RC) curves. *(Source: ASHRAE Handbook 1989, Fundamentals. Reprinted with permission.)*

(OSHA) Standard Part 1910.95, published by the U.S. Department of Labor, specifies: "Feasible administrative or engineering controls shall be utilized when employees are subjected to sound levels exceeding those in Table G-16. If such controls fail to reduce sound levels to below those in Table G-16, personal protective equipment shall be provided and used. Exposure to impulsive and impact noise should not exceed 140 dB peak sound pressure level." Table G-16 is reproduced in this text as Table 4.9.

TABLE 4.8 Recommended Criteria for Indoor Design RC or NC Range

Type of area	Recommended criteria for RC or NC range
Private residence	25–30
Apartments	25–30
Hotels/motels	
Individual rooms or suites	30–35
Meeting/banquet rooms	25–30
Halls, corridors, lobbies	35–40
Service/support areas	40–45
Offices	
Executive	25–30
Conference	25–30
Private	30–35
Open-plan	30–35
Computer equipment rooms	40–45
Public circulation	40–45
Hospitals and clinics	
Private rooms	25–30
Wards	30–35
Operating rooms	35–40
Corridors	35–40
Public areas	35–40
Churches	25–30
Schools	
Lecture and classrooms	25–30
Open-plan classrooms	30–35
Libraries	35–40
Concert halls	†
Legitimate theaters	†
Recording studios	†
Movie theaters	30–35

Note: These are for unoccupied spaces, with all systems operating.

*Design goals can be increased by 5 dB when dictated by budget constraints or when intrusion from other sources represents a limiting condition.

†An acoustical expert should be consulted for guidance on these critical spaces.

Source: *ASHRAE Handbook 1987, HVAC Systems and Applications*. Reprinted with permission.

TABLE 4.9 Occupational Noise Exposure (Occupational Health and Safety Administration Table G-16)

Duration per day, h	Sound level dBA slow response
8	90
6	92
4	95
3	97
2	100
$1\frac{1}{2}$	102
1	105
$\frac{1}{2}$	110
$\frac{1}{4}$ (or less)	115

Note: If the variations in noise level involve maxima at intervals of 1 s or less, it is to be considered continuous.

In all cases where the sound levels exceed the values shown herein, a continuing, effective hearing conservation program shall be administered.

Source: *Occupational Safety and Health Standard, Part 1910.95*. Reprinted with permission.

4.13 SPACE PRESSURE DIFFERENTIAL

Most air conditioning systems strive to maintain a slightly higher indoor pressure than that of the surroundings. This positive pressure tends to eliminate or reduce infiltration, the entry of untreated air to the space. Negative space pressure may cause space high humidity levels, mold and mildew growth, combustion equipment backdraft, and entry of sewer gas. For rooms where toxic,

hazardous, contaminated, or objectionable gases or substances are produced, a slightly lower pressure, or an appropriate negative room pressure, should be maintained to prevent the diffusion of these substances to the surroundings and at the same time prevent and reduce the damage of the uncontrolled outdoor airflow.

The magnitude of the positive or negative pressure to be maintained in the space must be carefully determined. A higher positive pressure always means a greater amount of exfiltrated air, which creates requirements for greater volumes of outdoor air intake. A higher negative pressure means a greater infiltration. Both result in higher initial and operating costs. Space pressurization is closely related to the amount of effective leakage area, or tightness of the building. Cummings et al. (1996) reported that the space pressure differentials between indoor and outdoor air and building tightness vary widely for seven field-surveyed restaurants. For comfort systems in low-rise buildings, the recommended pressure differential is +0.005 to +0.03 in. WC (+1.25 to +7.5 Pa), which is often adopted for average building tightness. For a 44-story high-rise building, the pressure differential between the entrance lobby outdoors and rooftop floor because of the stack effect during winter may exceed 0.3 in. WC (75 Pa). The phrase *in. WC* represents the pressure at the bottom of a water column that has a height of the specified number of inches.

Another type of space pressurization control for the health and comfort of passengers in an aircraft is cabin pressurization control. If an airplane is flying at an altitude of 28,000 ft (8540 m) with an ambient pressure of 4.8 psia (33.1 kPa abs.), the minimum cabin pressure required is 10.8 psia (75 kPa abs.); and a cabin pressurization control system must be installed to maintain a pressure differential of $10.8 - 4.8 = 6$ psi (41 kPa) between the cabin and the ambient air.

Process air conditioning systems, such as for clean rooms, need properly specified space pressurization to prevent contaminated air from entering the clean, uncontaminated area from surrounding contaminated or semicontaminated areas. According to Federal Standard 209B, *Clean Rooms and Work Station Requirements, Controlled Environment*, published by the U.S. government in 1973, the minimum positive pressure differential between the clean room and any adjacent area of less clean requirements should be 0.05 in. WC (12.5 Pa) with all entryways closed. When the entryways are open, an outward flow of air is to be maintained to minimize migration of contaminants into the room.

Space pressurization is discussed in Chaps. 20 and 23.

4.14 OUTDOOR DESIGN CONDITIONS

In principle, the capacity of air conditioning equipment is selected so that indoor design conditions can be maintained when the outdoor weather does not exceed the design values. The outdoor weather affects the space cooling load and the capacity of the air system to condition the required amount of outdoor air. It is not economical to choose the annual maximum or annual minimum values as the design data. Outdoor design conditions are usually determined according to statistical analysis of the weather data of the previous 30 years so that either 99.6 or 99 percent of the winter indoor design conditions can then be attained annually, or only 0.4, 1.0, or 2.0 percent of the time annually the summer indoor design conditions are exceeded. The greater the need for strict control of indoor environmental parameters, the greater the percentage coverage of the winter outdoor design hours or the smaller the percentage occurrence of the total hours that exceed the outdoor design values.

The recommended design values are based on data from the National Climatic Data Center (NCDC) from 1961 to 1991, or from 1982 to 1993, and Canadian Weather Energy and Engineering Data Sets (CWEEDS) from 1953 through 1993. Per *ASHRAE Handbook 1997, Fundamentals*, and ASHRAE/IESNA Standard 90.1-1999, the design conditions recommended for large cities in the United States are listed in Table 4.10.

Summer and Winter Outdoor Design Conditions

In Table 4.10, note the following:

- Summer outdoor design dry-bulb temperature for a specific locality $T_{o,s}$, in °F (°C), is the rounded higher integral number of the statistically determined summer design temperature $T_{o,ss}$ such that the average number of hours of occurrence of outdoor temperatures T_o higher than $T_{o,ss}$ annually is on average 0.4 percent (35 h), 1 percent (88 h), or 2 percent (175 h).
- The summer outdoor mean coincident design wet-bulb temperature $T'_{o,s}$, in °F (°C), is the mean of all wet-bulb temperatures occurring at the specific summer outdoor design dry-bulb temperature $T_{o,s}$ during the summer.
- The 1.0 percent summer design wet-bulb temperature T'_{o1} is the design value having an average annual occurrence of $T'_o \leq T'_{o1}$ of 88 h. Variable T'_{o1} is used for evaporative cooling systems, cooling towers, and evaporative condensers, which are covered in later chapters.
- The mean daily range, in °F (°C), indicates the mean of the difference between the daily maximum and minimum temperatures for the warmest month.
- The winter outdoor design dry-bulb temperature $T_{o,w}$, in °F, is the rounded lower integral value of the statistically determined winter outdoor design temperature $T_{o,ws}$, such that the annual average number of hours of occurrence of outdoor temperature at values $T_o \geq T_{o,ws}$ should be equaled or exceeded 99.6 or 99 percent of the total number of annual hours. The annual average number of hours in winter when $T_o < 0.99T_{o,ws}$ is 35 h, and $T_o < 0.99T_{o,ws}$ is 88 h.
- The number of degree-days is the difference between a base temperature and the mean daily outdoor air temperature for any one day $T_{\text{base}} - T_{o,m}$, both in °F. Annually, the total number of heating degree-days with a base temperature of 65°F, or HDD65, is

$$\text{HDD65} = \sum_{n=1} (65 - T_{o,m}) \tag{4.32}$$

  where n = number of days whose $T_{o,m} < 65$°F per annum. The total number of cooling degree-days with a base temperature of 50°F, or CDD50, is

$$\text{CDD50} = \sum_{m=1} (T_{o,m} - 50) \tag{4.33}$$

  where m = number of days whose $T_{o,m} > 50$°F per annum. Heating and cooling degree-days with different base temperatures have been used as climatic parameters to calculate the energy flux through building envelope, or to determine the U value and the configuration of the building envelope.

- In the last two columns, MWS/MWD to DB 99.6% indicates mean coincident wind speed (MWS)/mean coincident wind direction (MWD), i.e., most frequently occurring with the heating dry-bulb 99.6 percent. In MWD, wind direction is expressed in degrees; 270° represents west, and 180° south.

The Use of Outdoor Weather Data in Design

During the design of air conditioning systems, the following parameters are often adopted:

1. Indoor and outdoor design conditions are used to calculate the space cooling and heating loads.
2. Summer outdoor dry-bulb and coincident wet-bulb temperatures are necessary to evaluate the coil load. The summer outdoor wet-bulb temperature is used to determine the capacity of the evaporative coolers, cooling towers, and evaporative condensers.

TABLE 4.10 Climatic Conditions for the United States and Canada

City	State	Lat, degree	Elevation, ft	Winter: Design dry-bulb 99.6% °F	Winter: Design dry-bulb 99% °F	Summer: Design dry-bulb/mean coincident wet-bulb 0.4% °F	Summer: Design dry-bulb/mean coincident wet-bulb 1% °F	Summer: Design dry-bulb/mean coincident wet-bulb 2% °F	Mean daily range, °F	Design wet-bulb, 1% °F	Annual average daily incident solar energy on east or west, Btu/ft²· day	Annual cooling degree-days base 50°F, CDD50	Annual heating degree-days base 65°F, HDD65	MWS/MWD to DB 99.6% MWS	MWS/MWD to DB 99.6% MWD
Albuquerque	New Mexico	35.05	5,315	13	18	96/60	93/60	91/60	25.4	64	1,105	3,908	4,425	8	360
Anchorage	Alaska	61.17	131	−14	−9	71/59	68/57	65/56	12.6	58	538		10,570	8	290
Atlanta	Georgia	33.65	1,033	18	23	93/75	91/74	88/73	17.3	76	807	5,038	2,991	12	320
Atlantic City CO	New Jersey	39.45	66	8	13	91/74	88/73	86/72	18.1	76				9	310
Baltimore	Maryland	39.18	154	11	15	93/75	91/74	88/73	18.8	76	739	3,709	4,707	10	290
Billings	Montana	45.80	3,570	−13	−7	93/63	90/62	87/61	25.8	64	814	2,466	7,164	10	230
Birmingham	Alabama	33.57	630	18	23	94/75	92/75	90/74	18.7	77	789	5,206	2,918	7	340
Bismarck	North Dakota	46.77	1,660	−21	−16	93/68	90/67	86/66	26.5	70	766	2,144	8,968	7	290
Boise	Idaho	43.57	2,867	2	9	96/63	94/63	91/62	30.3	64	916	2,807	5,861	6	130
Boston	Massachusetts	42.37	30	7	12	91/73	87/71	84/70	15.3	74	659	2,897	5,641	17	320
Bridgeport	Connecticut	41.17	16	8	12	86/73	84/72	82/71	14.1	74		2,997	5,537	14	320
Buffalo	New York	42.93	705	2	5	86/70	84/69	81/68	17.7	72	609	2,468	6,747	12	270
Burlington	Vermont	44.47	341	−11	−6	87/71	84/69	82/68	20.4	72	698	2,228	7,771	6	70
Caribou	Maine	46.87	623	−14	−10	85/69	82/67	79/66	19.5	70	649	1,470	9,651	10	270
Casper	Wyoming	42.92	5,289	−13	−5	92/59	89/58	86/58	30.4	61	961	2,082	7,682	9	260
Charleston	South Carolina	32.90	49	25	28	94/78	92/77	90/77	16.2	79	796	6,188	2,013	7	20
Charleston	West Virginia	38.37	981	6	11	91/73	88/73	86/71	19.1	75	667	3,655	4,646	7	250
Charlotte	North Carolina	35.22	768	18	23	94/74	91/74	89/73	17.8	76	809	4,704		6	50
Chicago,	Illinois	41.98	673	−6	−1	91/74	88/73	86/71	19.6	75	729	2,941	6,536	10	270
Cincinnati CO	Ohio	39.10	482	5	12	93/74	90/75	88/73	20.0	76		3,733	4,988	9	260
Cleveland	Ohio	41.42	804	1	6	89/73	86/72	84/71	18.6	74		2,755	6,201	12	230
Concord	New Hampshire	43.20	344	−8	−2	90/71	87/70	84/68	24.1	73	630	2,087	7,554	4	320
Dallas/ Fort Worth	Texas	32.90	597	17	24	100/74	98/74	96/74	20.3	77	875	6,587	2,259	13	350
Denver	Colorado	39.75	5,331	−3	3	93/60	90/59	87/59	26.9	63	971	2,732	6,020	6	180
Des Moines	Iowa	41.53	965	−9	−4	93/76	90/74	87/73	18.5	76	788	3,371	6,497	11	320
Detroit	Michigan	42.23	663	0	5	90/73	87/72	84/70	20.4	74	676	3,046	6,167	11	240
Honolulu	Hawaii	21.35	16	61	63	89/73	88/73	87/73	12.2	75	953	9,949	0	5	320
Houston	Texas	29.97	108	27	31	96/77	94/77	92/77	18.2	79	805	6,876	1,599	8	340
Indianapolis	Indiana	39.73	807	−3	3	91/75	88/74	86/73	18.9	77	692	3,453	5,615	8	230
Jackson	Mississippi	32.32	331	21	25	95/77	93/76	92/76	19.2	79	833	5,900	2,467	7	340
Kansas City	Missouri	39.32	1,024	−1	4	96/75	93/75	90/74	18.8	77		3,852	5,393	10	320
Lansing	Michigan	42.77	873	−3	2	89/73	86/72	84/70	21.7	74		2,449	7,101	8	290
Las Vegas	Nevada	36.08	2,178	27	30	108/66	106/66	103/65	24.8	70	1,136	6,745	2,407	7	250

Lincoln CO	Nebraska	40.85	1,188	−7	−2	97/74	94/74	91/73	22.3	76		3,455	6,278	9	350
Little Rock	Arkansas	34.92	312	16	21	97/77	95/77	92/76	19.5	79	831	5,299	3,155	9	360
Los Angeles CO	California	33.93	105	43	45	85/64	81/64	78/64	10.9	69	962	4,777	1,458	6	70
Louisville	Kentucky	38.18	489	6	12	93/76	90/75	88/74	18.2	77	727	4,000	4,514	10	290
Memphis	Tennessee	35.05	285	16	21	96/78	94/77	92/77	16.8	79	806	5,467	3,082	10	20
Miami	Florida	25.82	13	46	50	91/77	90/77	89/77	11.4	79	874	9,474	200	10	340
Milwaukee	Wisconsin	42.95	692	−7	−2	89/74	86/72	83/70	16.6	74	724	2,388	7,324	13	290
Minneapolis/ St. Paul	Minnesota	44.88	837	−16	−11	91/73	88/71	85/70	19.1	74	709	2,680	7,981		
New Orleans	Louisiana	29.98	30	30	34	93/79	92/78	90/78	15.5	80	838	6,910	1,513	7	340
New York	New York	40.65	23	11	15	91/74	88/72	85/71	13.9	75	650	3,342	5,027	17	320
Norfolk	Virginia	36.90	30	20	24	93/77	91/76	88/75	15.3	77	792	1,586	3,609	12	340
Phoenix	Arizona	33.43	1,106	34	37	110/70	108/70	106/70	23	75	1,116	8,425	1,350	5	90
Pittsburgh	Pennsylvania	40.50	1,224	2	7	89/72	86/70	84/69	19.5	73	642	2,836	5,968	10	260
Portland CO	Oregon	45.60	39	22	27	90/67	86/66	83/64	21.6	67	647	2,517	4,522	13	120
Providence	Rhode Island	41.73	62	5	10	89/73	86/71	83/70	17.4	74	677	2,743	5,884	12	340
Rapid City	South Dakota	44.05	3,169	−11	−5	95/65	91/65	88/64	25.3	68	819	2,412	7,301	9	350
St. Louis	Missouri	38.75	564	2	8	95/76	93/75	90/74	18.3	78	797	4,283	4,758	12	290
Salt Lake City	Utah	40.78	4,226	6	11	96/62	94/62	92/61	27.7	65	975	3,276	5,765	7	160
San Antonio	Texas	29.53	794	26	30	98/73	96/73	94/74	19.1	77	878	7,142	1,644	10	350
San Diego	California	32.73	30	44	46	85/67	81/67	79/67	8.9	71	950	5,223	1,256	3	70
San Francisco CO	California	37.62	16	37	39	83/63	78/62	74/61	16.7	63	941	2,883	3,016	5	160
Seattle CO	Washington	47.45	449	23	28	85/65	81/64	78/62	18.3	65	621	2,120	4,611	10	10
Shreveport	Louisiana	32.47	259	22	26	97/77	95/77	93/76	19.1	79	843	6,166	2,264	9	360
Spokane	Washington	47.62	2,461	1	7	92/62	89/61	85/60	26.1	63	758	2,032	6,842	7	50
Syracuse	New York	43.12	407	−3	2	88/72	85/71	83/70	20.3	73	611	2,399	6,834	7	90
Tucson	Arizona	32.12	2,556	31	34	104/65	102/65	100/65	29.4	71	1,112	6,921	1,678	7	140
Tulsa	Oklahoma	36.20	676	9	14	100/76	97/76	94/75	19.5	78	820	5,150	3,691	11	360
Washington	DC	38.85	66	15	20	95/76	92/76	89/74	16.6	78	724			11	340
Wichita	Kansas	37.65	1,339	2	8	100/73	97/73	94/73	22.2	76		4,351	4,791	13	360
Wilmington	Delaware	39.68	79	10	14	91/75	89/74	86/73	17.0	76	726	3,557	4,937	11	290
Canada															
Calgary	Alberta	51.12	3,556	−22	−17	83/60	80/59	77/57	22.0	61				7	0
Montreal	Quebec	45.47	118	−12	−7	85/71	83/70	80/68	17.6	72				7	250
Regina	Saskatchewan	50.43	1,893	−29	−24	89/64	85/64	82/62	23.6	66				9	270
Toronto	Ontario	43.67	568	−4	1	87/71	84/70	81/68	20.2	72				9	340
Vancouver	British Columbia	49.18	7	18	24	76/65	74/64	71/62	14.0	64				6	90
Winnipeg	Manitoba	49.90	784	−27	−23	87/68	84/67	81/66	20.5	70				7	320

Note: CO designates locations within an urban area. Most of the data are taken from city airport temperature observations. Some semirural data are comparable to airport data.
Source: Abridged with permission from *ASHRAE Handbook 1997, Fundamentals* and ASHRAE/IES Standard 90.1-1999.

3. Outdoor weather data presented consecutively for a whole year of 8760 h, or other simplified form, are sometimes used for year-round energy estimations.
4. Outdoor climate has a significant influence on the selection of an air conditioning system and its components.

Outdoor Weather Characteristics and Their Influence

The following are characteristics of outdoor climate in the United States:

1. For clear days during summer, the daily maximum temperature occurs between 2 and 4 P.M. In winter, the daily minimum temperature usually occurs before sunrise, between 6 and 8 A.M.
2. The maximum combined influence of outdoor temperature and solar radiation on load calculations usually occurs in July or August. January is often the coldest month in many locations.
3. The mean daily temperature range of the warmest month varies widely between different locations. The smallest mean daily range, 10°F (5.6°C), occurs at Galveston, Texas. The greatest, 45°F (25°C), occurs at Reno, Nevada. Many cities have a mean daily range of 15 to 25°F (8.3 to 13.9°C). Usually, coastal areas have smaller mean daily ranges, and continental areas and areas of high elevation have greater values.
4. Extremes in the difference between 1 percent design dry-bulb temperature in summer and the 99 percent design temperature in winter in the United States are
   - A maximum of 119°F (66°C) at Bettles, Alaska
   - A minimum of 18°F (10°C) at Kaneohe, MCAS, Hawaii
5. Snelling (1985) studied outdoor design temperatures for different locations all over the country. He found that extremely cold temperatures may last for 3 to 5 days. Extremely hot temperatures seldom last more than 24 h.
6. Among the 538 locations listed in Table 4.10, 138 locations have a summer mean coincident wet-bulb temperature corresponding to 1 percent dry-bulb, $T'_{os} \leq 68$°F (20°C). There it is advantageous to use evaporative cooling systems to replace part of or all the refrigeration capacity.
7. When commercial buildings are only occupied after 10 A.M. or even after 12 noon, a winter outdoor design temperature higher than the 99.0 percent design dry-bulb temperature should be considered.

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CHAPTER 5
ENERGY MANAGEMENT AND CONTROL SYSTEMS

5.1 AUTOMATIC CONTROL SYSTEM AND HISTORICAL DEVELOPMENT 5.2
- Air Conditioning Automatic Control System 5.2
- Historical Development 5.2
- Distribution of Energy Management and Control Systems 5.3

5.2 CONTROL LOOP AND CONTROL METHODS 5.5
- Control Loop 5.5
- Sequence of Operations 5.5
- Control Methods 5.7
- Comparison of Control Methods

5.3 CONTROL MODES 5.9
- Two-Position Control 5.9
- Step Control and Modulating Control 5.10
- Floating Control 5.11
- Proportional Control 5.11
- Proportional plus Integral (PI) Control 5.13
- Proportional-Integral-Derivative (PID) Control 5.14
- Compensation Control, or Reset 5.15
- Applications of Various Control Modes 5.15

5.4 SENSORS AND TRANSDUCERS 5.16
- Temperature Sensors 5.18
- Humidity Sensors 5.18
- Pressure Sensors 5.19
- Flow Sensors 5.19
- Carbon Dioxide and Air Quality Sensors 5.20
- Occupancy Sensors 5.20
- Wireless Zone Sensors and Intelligent Network Sensors 5.21
- Transducers or Transmitters 5.21

5.5 CONTROLLERS 5.21
- Direct-Acting or Reverse-Acting 5.21
- Normally Closed or Normally Open 5.22
- Pneumatic Controllers 5.22
- Electric and Electronic Controllers 5.23
- Direct Digital Controllers 5.23

5.6 WATER CONTROL VALVES AND VALVE ACTUATORS 5.26
- Valve Actuators 5.26
- Types of Control Valves 5.27
- Valve Characteristics and Ratings 5.27
- Valve Selection 5.29
- Valve Sizing 5.30

5.7 DAMPERS AND DAMPER ACTUATORS 5.32
- Types of Volume Control Dampers 5.32
- Damper Actuators (Motors) 5.33
- Volume Flow Control between Various Airflow Paths 5.33
- Flow Characteristics of Opposed- and Parallel-Blade Dampers 5.35
- Damper Selection 5.37
- Damper Sizing 5.37

5.8 SYSTEM ARCHITECTURE 5.38
- Architecture of a Typical EMCS with DDC 5.38
- System Characteristics 5.40
- Future Development 5.41

5.9 INTEROPERABILIITY AND OPEN PROTOCOL BACnet 5.41
- Interoperability 5.41
- BACnet—Open Data Communication Protocol 5.41
- Application Layer 5.42
- Network Layer 5.43
- Data Link/Physical Layer—Network Technology 5.43
- Connection between BACnet and Proprietary Network 5.44
- LonTalk Protocol 5.44

5.10 CONTROL LOGIC AND ARTIFICIAL INTELLIGENCE 5.45
- Fuzzy Logic 5.45
- Knowledge-Based Systems and Expert Systems 5.47
- Artificial Neural Networks 5.51

5.11 PROGRAMMING FOR DDC SYSTEMS 5.53
- Evolution of DDC Programming 5.53
- Graphical Programming 5.53
- Templates 5.54
- Graphical Programming for Mechanical Cooling Control 5.55

5.12 TUNING DDC UNITS 5.55
- Tuning PI Controllers 5.55
- Self-Tuning PI and PID Controllers 5.55
- Adaptive Control 5.56

5.13 FACTORS AFFECTING CONTROL PROCESSES 5.56
- Load 5.56
- Climate Change 5.56
- System Capacity 5.57
- Performance of Control Processes 5.57
- Thermal Capacitance 5.58

5.14 FUNCTIONAL CONTROLS 5.58
- Generic Controls 5.59

Specific Controls 5.60
Commissioning and Maintenance 5.61
5.15 FAULT DETECTION AND DIAGNOSTICS 5.61
Basics 5.61
Expert System Rule-Based Diagnostics 5.62
ARX and ANN Model-Based Diagnosis 5.63
5.16 CONTROLS IN ASHRAE/IESNA STANDARD 90.1-1999 5.66
General 5.66
Off-Hour Controls 5.66
REFERENCES 5.67

5.1 AUTOMATIC CONTROL SYSTEM AND HISTORICAL DEVELOPMENT

Air Conditioning Automatic Control System

An air conditioning automatic control system or simply a control system, primarily modulates the capacity of the air conditioning equipment to maintain predetermined parameter(s) within an enclosure or for the fluid entering or leaving the equipment to meet the load and climate changes at optimum energy consumption and safe operation. The predetermined parameter or variable to be controlled is called the *controlled variable*. In air conditioning or in HVAC&R, the controlled variable can be temperature, relative humidity, pressure, enthalpy, fluid flow, contaminant concentration, etc.

Because of the variation of the space load and the outdoor climate, a control system is one of the decisive factors for an air conditioning, or an HVAC&R, system to achieve its goal: to effectively control the indoor environmental parameters, provide an adequate amount of outdoor air, be energy-efficient, and provide better security and safety. Today nearly all HVAC&R systems are installed with a control system to provide effective operation and energy conservation.

Historical Development

Early controls for comfort air conditioning systems used in the Capitol building since 1928 were pneumatic controls that included high- and low-limit thermostats in the supply air discharge of the air-handling unit. These controls were used to maintain the desired supply air temperature. Dewpoint control of the supply air maintained room humidity within a desirable range. A thermostat in the return-air passage was used to control the air volume flow rate passing through the cooling coil, and a 16-point recorder connected with resistance thermometers was used to record the room temperature at various locations.

Shavit (1995) stated that the use of thermocouples offers the possibility of remote monitoring of the space temperature. In coordination with pneumatic local control systems, the first centralized monitoring system with a central panel and remote set point change was installed in the White House in 1950. The concept of a *building automation system* (BAS) which centralizes the monitoring management of building services was devised and developed in 1950s. The primary improvement of the pneumatic control system occurred in 1963. The vapor and bulb-based sensors were replaced by a remote pneumatic sensor using a bimetal strip and a nozzle flapper to regulate the compressed air pressure between 3 and 15 psi (20.7 and 103.4 kPa).

Shavit also indicated that the first on-line computer was introduced in 1967 in One Main Place in Dallas, Texas. This system had the first set energy conservation software including enthalpy control, demand limit, optimum start/stop, reset according to the zone of highest demand, chiller optimization, and night purge. In 1970, the introduction of all-electronic, solid-state centralized control system was another significant milestone. Solid-state components improved the scanning process and serial transmission as well as reduced many wires in the trunk wiring to a single pair.

The 1973 energy crisis greatly boosted the control industry. Many energy management systems (EMSs) were installed for the purpose of saving energy. The microprocessor-based direct digital control (DDC) was first introduced in 1981 and the DDC unit controller in 1986–1987. By means of the incorporated hardware and software, it tremendously increased the control functions, developed the control logic, and accelerated the data processing and analysis.

Distribution of Energy Management and Control Systems

A building automation system is a centralized monitoring, operation, and management system of the building services in a building including air conditioning (HVAC&R), lighting, fire protection, and security. Only air conditioning control systems and smoke control systems are covered in this handbook.

In the United States, a single-zone, two-position electric control system, as shown in Fig. 5.1, is now widely used in residential and light commercial buildings that adopt DX packaged systems. *Single-zone* means that the load characteristics of the whole conditioned space are similar, it is controlled by one controller, and *two-position* means on-off control. This kind of control system is simple, easy to manage, and low in cost. For a large high-rise building constructed in the 1990s, an energy management and control system (EMCS), as shown in Fig. 5.2, is often the choice. *Variable air volume* indicates that airflow is varied to match the variation of the system load. Today, an EMCS is a microprocessor-based system with direct digital control which optimizes the operation and the indoor environmental parameters of an HVAC&R system in order to maintain a healthy and comfortable environment at optimum energy use. Today, an EMCS is an advanced-generation energy management and control system and is also a part of a building automation system.

According to the surveyed results in *Commercial Buildings Characteristics 1992* by EIA, the percentages of the floor area served by the installed EMCS among the total conditioned area in commercial buildings are as follows:

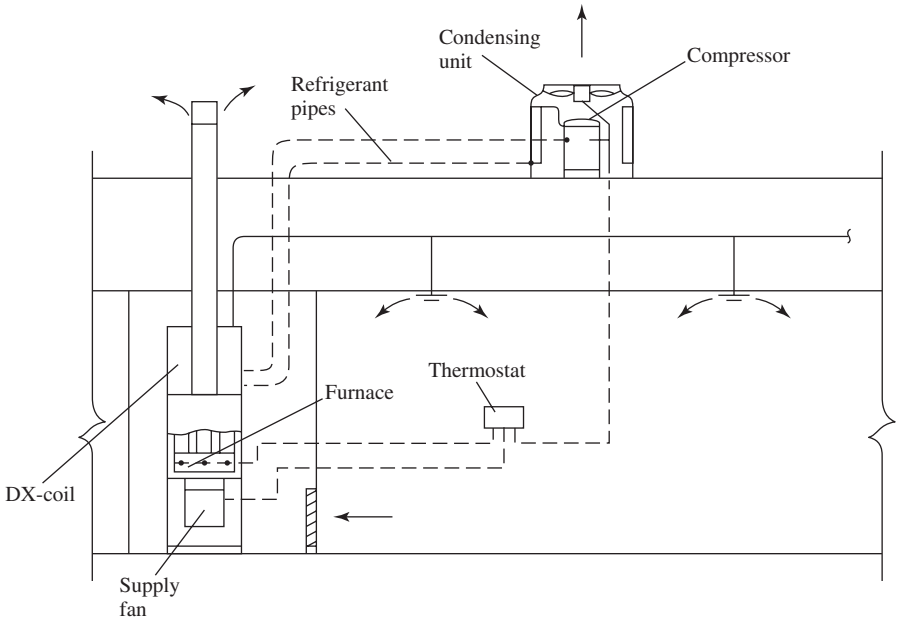


FIGURE 5.1 A single-zone, two-position electric control system.

Year built	Building with EMCS, million ft^2	Percentage of floor space
Before 1970	5431	18
1970–1979	3313	29
1980–1989	4343	36
1990–1992	1236	58

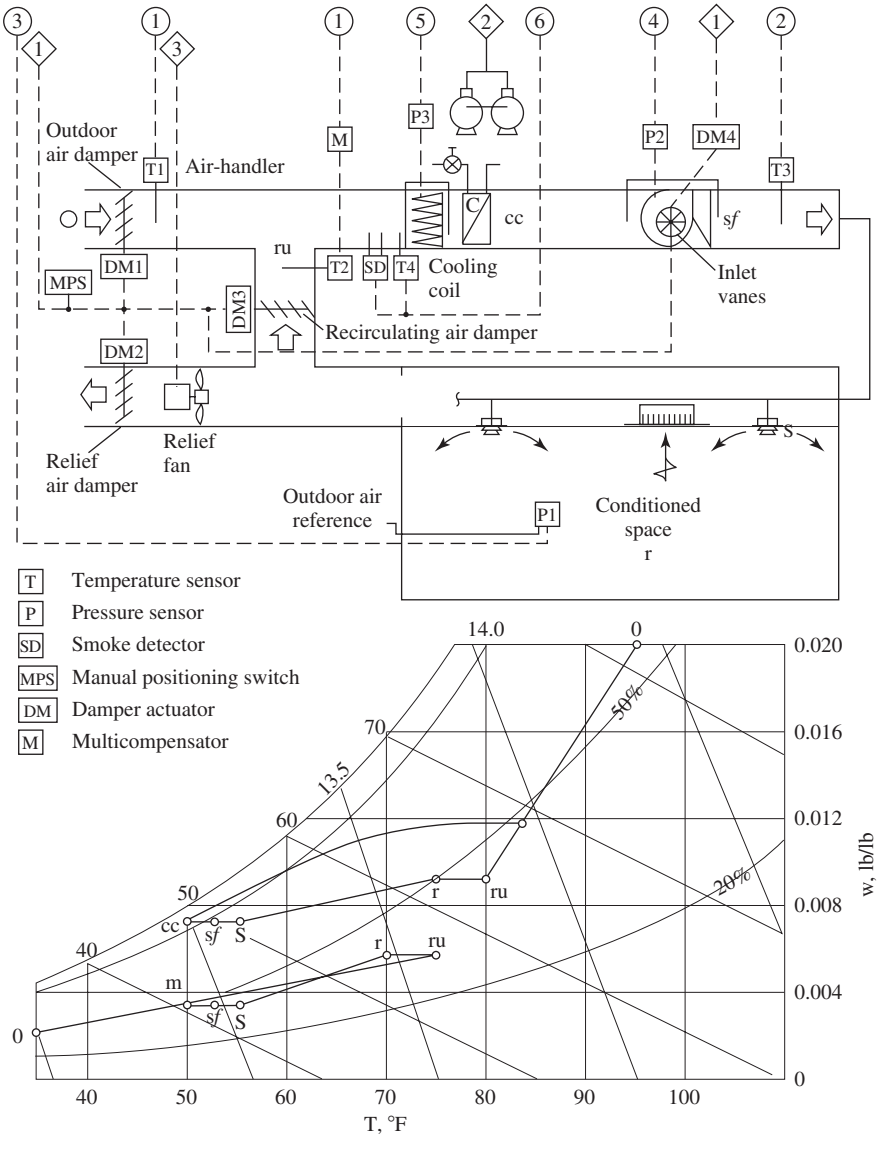


FIGURE 5.2 Control diagram and psychrometric analysis of an energy management and control system for a single-zone, VAV cooling packaged system.

The later the building is constructed and the larger the conditioned area of the commercial building, the greater is the chance to install an energy management and control system (EMS).

5.2 CONTROL LOOP AND CONTROL METHODS

Control Loop

The basic element of a control system is a control loop. A control loop often consists of a sensor (such as T2 in Fig. 5.2), which senses and measures the controlled variable of recirculating air; a DDC unit controller which compares the sensed input signal with the predetermined condition (the set point) and sends an output signal to actuate the third element, a controlled device or control element (such as a damper or a valve in Fig. 5.2). Modulation, minute adjustments, or on-off control of dampers and valves will change the controlled devices position or operating status, which affects the controlled variable by changing the airflow and water flow, electric power supply, and so on. The controlled variable is thus varied toward the predetermined value, the set point. Air and water are control media or control agents. Airflow and water flow are the manipulated variables. The equipment that varies the output capacity by changing the opening position of the dampers and valves is called the process plant.

There are two types of control loops: open and closed. An open-loop system assumes a fixed relationship between the controlled variable and the input signal being received. The sensed variable is not the controlled variable, so there is no feedback. An example of an open-loop system would be a ventilating fan that turns on when the outdoor temperature exceeds a specified set point. The sensed variable is the outdoor air temperature, and the controlled variable is the state of the fan (on or off).

A closed-loop system depends on sensing the controlled variable to vary the controller output and modulate the controlled device. In Fig. 5.2, temperature sensor T2 senses the controlled variable, the recirculating air temperature entering the air handler T_{ru}; the DDC unit controller receives this sensed signal input and produces an output according to the software stored to modulate the fan speed, or the position of the inlet vanes of the supply fan. As the supply fan speed changes, or inlet vanes open and close, the supply air volume flow rate varies and the space temperature T_r and recirculating temperature T_{ru} change accordingly. This change in T_{ru} is sensed again by T2 and fed back to the controller for further modulation of fan speed or inlet vanes to maintain values of T_{ru} that approach the set points. These components form a closed-loop system.

Figure 5.3 shows a block diagram of this closed-loop system. It shows a secondary input to the controller, such as the outdoor air temperature T_o, which may reset the set point in the controller to provide better and more economical control. The disturbances that affect the controlled variable are load variations and changes in the outdoor weather. After the controller senses the signal feedback, it sends a corrective signal to the controlled device based on the difference between the sensed controlled variable and the set point. Thus, T_{ru} is under continuous comparison and correction.

A control system or its component, control subsystem, used to control the controlled variable(s) in a conditioned space, or within a mechanical device or equipment, may contain only one control loop; or it may contain two or more loops.

Sequence of Operations

The sequence of operations is a description of the sequential order of the functional operations that a control system is supposed to perform, which plans and guides the operation and control of an air conditioning system. For a single-zone variable-air-volume (VAV) cooling packaged system shown in Fig. 5.2, when cold air supply is required during full occupancy, the sequence of operations of cooling mode is as follows:

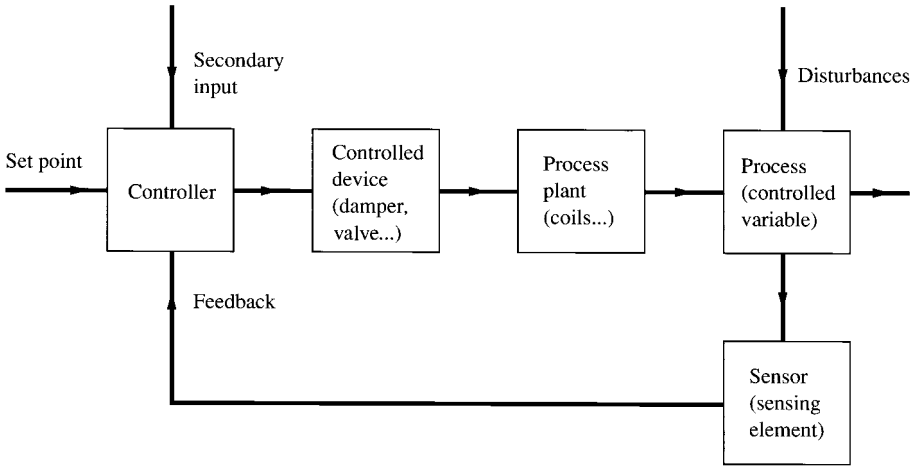


FIGURE 5.3 Block diagram for a closed-loop feedback control system.

- The supply fan is started and stopped by the scheduling software stored in the DDC unit controller. Manual override is possible. When the time schedule puts this system in cooling occupy mode, the microprocessor-based controller goes through a short initiation period, such as a 2-min period. During this period, dampers are driven to fully open, minimum open, and fully closed positions. These determine the effective range of the economizer potentiometer range. The smoke detector in the return air or the low-temperature limit sensor of the mixed air will stop the supply fan if necessary. The supply fan status (on or off) is determined by the pressure differential switch across the fan. When the initiation period is completed, the supply fan is turned on.
- The control system tends to maintain the recirculating temperature T_{ru} at around the cold set point, and it uses the 100 percent all outdoor air free cooling economizer cycle as the first-stage cooling. If the outdoor temperature $T_o \leq T_{ru}$, the outdoor air damper is fully opened and the recirculating air damper closed. If the outdoor air temperature $T_o > T_{ru}$, the outdoor air damper is closed to a minimum position to provide required outdoor ventilation air, and the recirculating damper is fully opened.
- When the initiation period is completed, the supply fan is turned on from its zero speed. If T_{ru}, sensed by the temperature sensor T2, is at a value above the set point, $T_{ru} > T_{c,\text{set}}$, also $T_o \leq T_{ru}$, the speed of the supply fan is gradually increased by the variable-speed drive inverter, resulting in a higher supply volume flow rate. When the fan speed is raised to its upper limit, the supply volume flow rate is then at its maximum value. A still higher space load further raises T_{ru}, and it exceeds the cold set point $T_{c,\text{set}}$, and if $T_o \leq T_{ru}$, chilled water starts to flow to the cooling coil to cool the air simultaneously with the outdoor air free cooling, in order to make $T_{ru} \cong T_{c,\text{set}}$.
- Only when the outdoor air damper is fully open and the static pressure difference between the space air and outdoor air is greater than a preset value, such as $\Delta p_{s-o} > 0.03$ in. WC (7.5 Pa), will the relief fan be energized. Its speed is modulated to maintain $\Delta p_{s-o} \approx 0.03$ in. WC (7.5 Pa).
- When T_{ru} drops, the chilled water flow to the cooling coil is reduced first, and then the outdoor air free cooling, outdoor air volume flow rate.

Refer to Sec. 21.2 for the details of the sequence of operations of single-zone VAV cooling systems. In the design and operation of an EMCS system, the necessary documentation includes the sequence of operation, control diagrams, specifications, operation, and maintenance manual.

Control Methods

According to the types of control signal and the different kinds of energy used to transmit the signals, as well as whether a software is used during control operation, control methods can be classified as direct digital, pneumatic, electric, and electronic.

Analog and Digital. There are two types of control signals: analog and digital. An analog signal is in the form of a continuous variable. It often uses the magnitude of electric voltage or pneumatic pressure to represent the air temperature. A digital signal is a series of on and off pulses used to transmit information.

A conventional analog controller receives a continuous analog signal, such as a voltage or a pneumatic signal, that is proportional to the magnitude of the sensed variable. The controller compares the signal received from the sensor to the desired value (i.e., the set point) and sends a signal to the actuator in proportion to the difference between the sensed value and the set point.

A digital controller, or microprocessor-based controller, receives an electric signal from sensor(s). It converts the electric signal to digital pulses of different time intervals to represent the signals values. The microprocessor of the digital controller performs the mathematical operations and knowledge processing on these values. The output from the microprocessor can be either in digital form to actuate relays or converted to an analog signal (say, a voltage or a pneumatic pressure) to operate the actuator(s).

Direct Digital Control (DDC). A control system using DDC involves adopting a microprocessor-based digital controller to perform mathematical operations and knowledge processing according to the predetermined control algorithms or computer programs. The key element of DDC compared to analog control is the software and hardware contained in the direct digital controller which expands the control functions tremendously and adopts recently developed control logic. ADDC unit usually has more precise sensors and uses the same type of controlled devices as other control methods.

Figure 1.2 shows an energy management and control system using DDC for an air-handling unit in a typical floor of the NBC Tower, and Fig. 5.2 shows an EMS with DDC for a single-zone VAV system.

Pneumatic Control. In a control system using pneumatic control, compressed air is used to operate the sensors, controllers, and actuators and to transmit the signals. It consists of: a compressed air supply and distribution system, sensors, controllers, and actuators. Figure 5.4 shows a typical pneumatic control system. In Fig. 5.4, a filter is used to remove the dust particles, including submicrometer-size particles, contained in the air. The function of the pressure-reducing valve is to reduce the pressure of compressed air discharged from the air compressor to the required value in the main supply line. The discharged compressed air is usually at a gauge pressure of 18 to 25 psig (124 to 172 kPa·g), and the pressure signal required to actuate the valve or damper actuator is 3 to 13 psig (20.7 to 89.6 kPa · g).

The advantages of pneumatic control are as follows:

- The compressed air itself is inherently a proportional control signal.
- The cost of modulating actuators is low, especially for large valves and dampers.
- Pneumatic controls require less maintenance and have fewer problems.
- Pneumatic controls are explosionproof.

The disadvantages stem mainly from comparatively fewer control functions, the high cost of sophisticated pneumatic controllers, and the comparatively higher first cost of a clean and dry, compressed air supply for small projects.

Electric or Electronic Control. Both types of control use electric energy as the energy source for the sensors and controllers. A control system using electric control often offers two-position on-off control, as shown in Fig. 5.1. Switches, relays, contactors, and electromechanical devices are

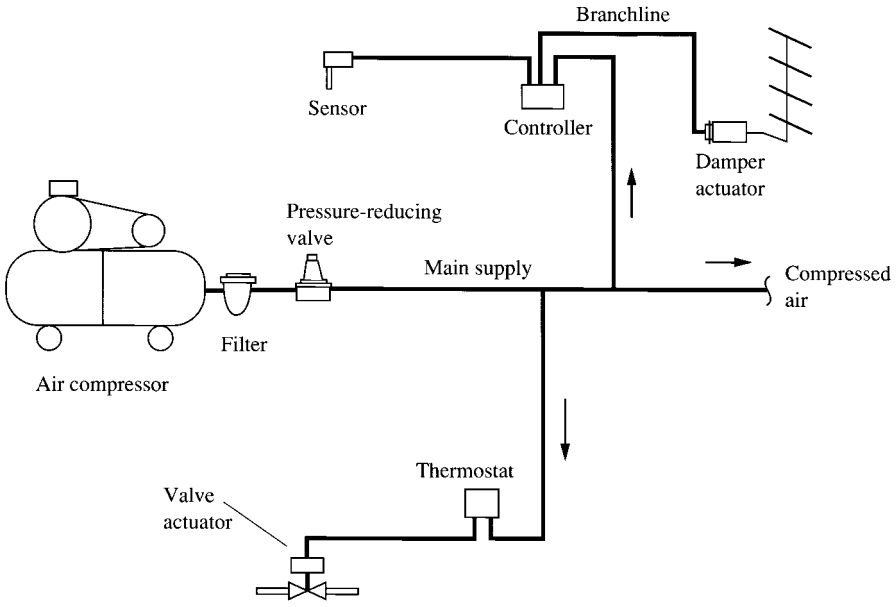


FIGURE 5.4 A typical pneumatic control system.

system components for electric control systems. They are generally used for low-cost, small, and simpler control systems. In addition to the switches and relays, a control system using electronic control has transistors, diodes, capacitors, and printed-circuit boards as system components. Electronic control systems always have more accurate sensors and solid-state controllers with sophisticated functions and can be easily interfaced with the building automation system. Electronic control has a faster response and more accurate processing of data than electric control systems. Electronic control systems cost more and need skilled personnel for maintenance and troubleshooting.

Comparison of Control Methods

Because of the increasing demand for more complicated controllers to satisfy the needs of better indoor environmental control, satisfactory indoor air quality, improved energy saving, lower cost, and greater reliability, the recent trend is to use EMCS with direct digital control for more demanding and large projects. A modern DDC system consists of electronic sensors, microprocessor-based controllers incorporated with electronic components, and electronic or pneumatic actuators. It is often more effective and cheaper to operate a large pneumatic actuator than to use a large electronic actuator in a DDC system.

An EMCS using DDC offers the following advantages:

1. Flexibility of providing required control functions and the ability to coordinate multiple functions directly from complicated software programs (it can even mimic a human expert within a certain knowledge domain and offer artificial intelligence)
2. More precise and faster-response control actions provided by the microprocessor-based controller
3. The possibility of using high-level self-checking and self-tuning system components, which increases the system reliability

One of the main disadvantages of DDC is a still-higher first cost. However, the cost per control action is comparatively lower. As more and more DDC systems are installed, DDC will become familiar to us. The cost of the microprocessor-based controllers and other DDC system components will drop further in the future. In the 1990s, the trend in HVAC&R control for new and retrofit projects is to use EMCS with DDC systems except for small projects.

5.3 CONTROL MODES

Control modes describe how the corrective action of the controller takes place as well as its effect on the controlled variable. For applications in HVAC&R, control modes can be classified as two-position, step, floating, proportional, proportional-integral, and proportional-integral-derivative.

Before we discuss the control modes in detail, the term *lag*, or *time lag*, should be introduced. According to ASHRAE terminology, lag is (1) the time delay required for the sensing element to reach equilibrium with the controlled variable; or (2) any retardation of an output with respect to the causal input, including the delay, because of the transport of material or the propagation of a signal.

Two-Position Control

In two-position control, the controller controls the final control element at one of two positions: maximum or minimum (except during the short period when it changes position). Examples of two-position control include starting and stopping the motor of a fan, pump, or compressor and turning on or off an electric heater. Sometimes it is called on-off control.

Figure 5.5 shows a two-position control mode for an electric heater installed in a branch duct. In the middle diagram, the ordinate indicates the controlled variable, the space temperature T_r; in the lower diagram, the ordinate denotes the output capacity of the final control element, the electric heater. If the controller turns on the electric heater when the sensor senses a space temperature 69.5°F (20.8°C), and turns off at 70.5°F (21.4°C), the result is a cyclic operation of the electric heater and the rise and fall of T_r toward the two positions of 69.5°F (20.8°C) and 70.5°F (21.4°C). The thermal storage effect of the electric heater, the branch duct, the building envelope that surrounds the space, and the sensor itself will have a time lag effect on T_r. The rise of T_r is a convex curve with an overshoot higher than 70.5°F (21.4°C), and the drop of T_r is a concave curve with an undershoot lower than 69.5°F (20.8°C). The difference between the two points, on and off, is called the *differential*.

If the heating capacity of the electric heater that results from an on-and-off cyclic operation is represented by $Q_{e,t}$, and if the actual space heating load $Q_{rh} < \frac{1}{2}Q_{e,t}$, the slope of the rising T_r curve will then be greater than the slope of the falling T_r curve. As a consequence, the "on" period will be shorter than the "off" period. Such a cyclic T_r curve is shown in the upper left corner of Fig. 5.5. If $Q_{rh} > \frac{1}{2}Q_{e,t}$, the condition will be reversed (see the upper right corner of Fig. 5.5). The two-position control varies the ratio of on and off periods to meet any variation in the space heating load.

To reduce the overshoot and undershoot of the controlled variable in a two-position control mode, a modification called *timed two-position control* has been developed. A small heating element attached to the temperature sensor is energized during on periods. This additional heating effect on the sensor shortens the on timing. During off periods, the heating element is deenergized.

The differential of two-position control, the overshoot, and the undershoot all result in a greater fluctuation of the controlled variable. A suitable differential is always desirable in two-position control in order to prevent very short cycling, which causes hunting, a phenomenon of short cycling of the controlled variable. Two-position control is not suitable for precise control of the controlled variable, but it is often used for status control, such as opening or closing a damper, turning a small single piece of equipment on or off for capacity control, etc., and for lower-cost control systems.

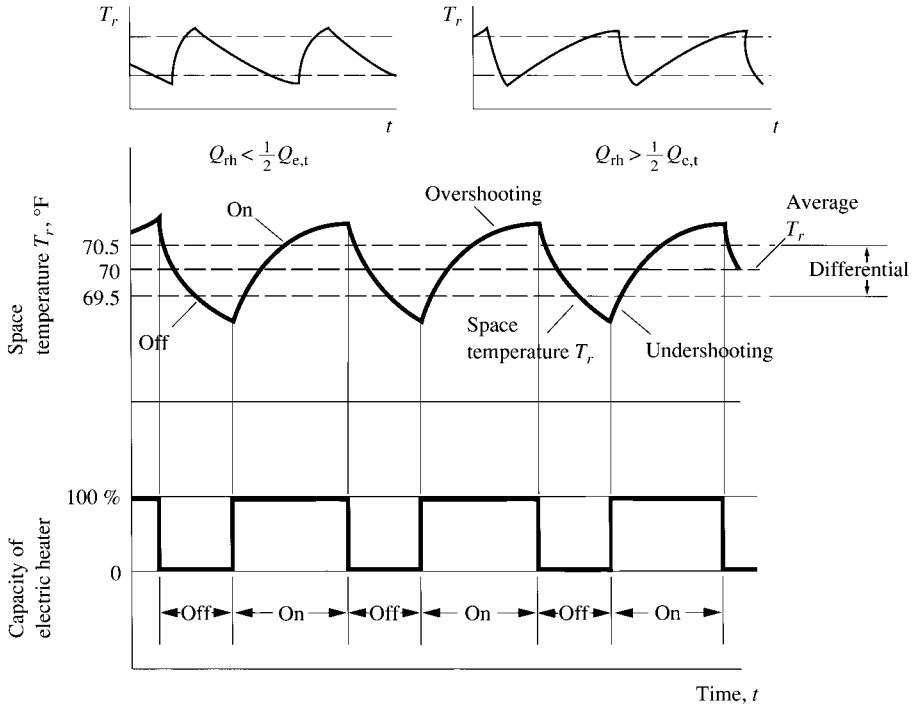


FIGURE 5.5 A typical two-position control mode.

Step Control and Modulating Control

In step control, the controller operates the relays or switches in sequence to vary the output capacity of the process plant in steps or stages. The greater the deviation of the controlled variable from the set point and the faster the rate of change of the controlled variable, the higher will be the output capacity of the process plant. In modulating control, the controller activates the control device continuously and thus the change of the output capacity gradually.

During the earlier stage of step control, there must be a differential of controlled variable between two on or two off points of a particular piece of equipment of the process plant. The result is a greater fluctuation of the controlled variable. In the DDC controller-activated new generation of step control, the software of the DDC unit determines how fast each capacity step is added or subtracted according to the deviation from the set point and the rate of change of the controlled variable.

Figure 5.6 shows step control of the cooling capacity of refrigeration compressors using DDC to maintain the discharge air temperature T_{dis} within predetermined limits in a packaged unit. When the air economizer alone can no longer balance the DX coils load, and T_{dis} floats to the upper limit of the control band, point 1, the first stage of cooling is energized. If the cycling and the continuously energized first-stage cooling capacity still cannot balance the coils load, then T_{dis} continues to rise until it reaches point 2, which is 1°F (0.56°C) higher than the upper limit of the control band. The first-stage cooling is then locked on, and the second stage is energized to cycling. When T_{dis} rises to point 2, all the present cooling stages will be locked on, and an additional cooling stage is added for cycling. When T_{dis} drops to point 7, the current cycling cooling stage will be deenergized, and the one cooling stage next to the deenergized stage will be cycling. It always needs a time delay, say at least 4 min, to turn on or off again in order to prevent hunting and other damage.

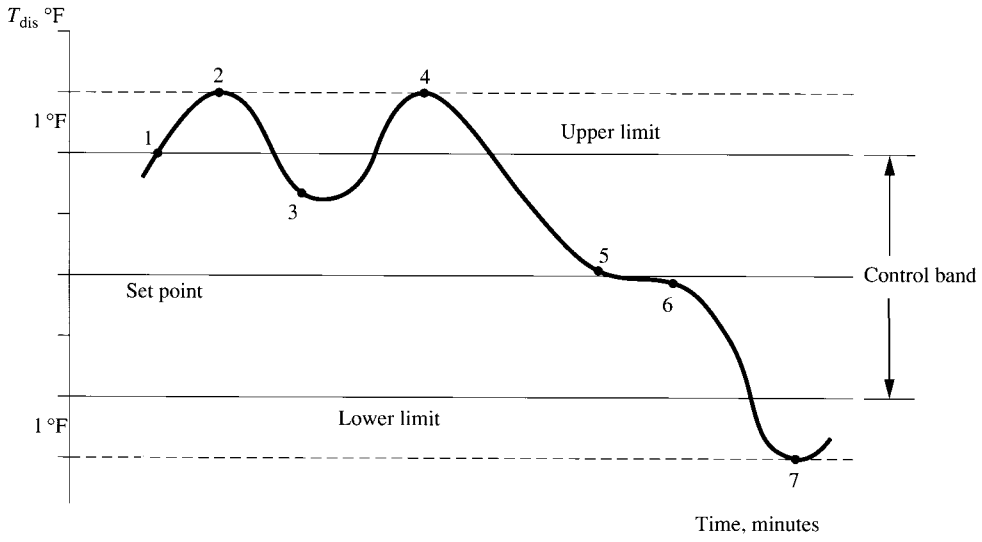


FIGURE 5.6 DDC-activated discharge air temperature control for a VAV rooftop packaged system.

Step control is another kind of on-and-off control having smaller varying capacity and thus a fluctuating controlled variable. It has been widely used for the capacity control of refrigeration compressors and the electric heaters.

Floating Control

In a floating control mode, the controller moves the control device by means of the actuator toward the set point only when the control point is out of the differential, or dead band, as shown in Fig. 5.7. A control point is the actual value of the controlled variable at a certain time instant. Figure 5.7 represents a duct static pressure control system using a floating control mode by opening and closing the inlet vanes of a supply fan. The control device—inlet vanes—can be moved in either an opening or a closing direction depending on whether the control point is over the upper limit of the differential or under the lower limit. In Fig. 5.7, when the controlled variable, the duct static pressure Δp_s, is above the upper limit of the differential, the controller then closes the inlet vanes. If Δp_s is below the lower limit of the differential, the controller opens the inlet vanes.

A floating control mode is more suitable for control systems with a minimal lag between the sensor and the control medium. A control medium is the medium in which the controlled variable exists.

Proportional Control

In a proportional control mode, the controller moves the controlled device to a position such that the change in its output capacity is proportional to the deviation of the controlled variable from the set point. The position of the controlled device is linearly proportional to the magnitude of the controlled variable. Figure 5.8 represents the control of the space temperature through the modulation of a two-way valve of a cooling coil using proportional control. In Fig. 5.8, the controlled variable is the space temperature T_r, and the controlled device is the valve.

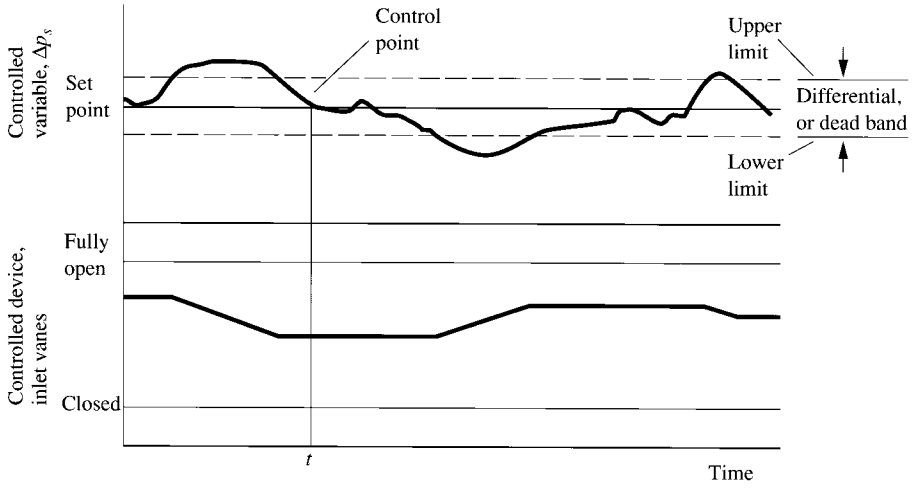


FIGURE 5.7 Floating control mode.

In a proportional control mode, the throttling range is the change in the controlled variable when the controller moves the controlled device from the position of maximum output to the position of minimum output. The controlled variable's range of values that will move the proportional controller through its operating range is called the *proportional band*. In a proportional control system, the throttling range is equal to the proportional band.

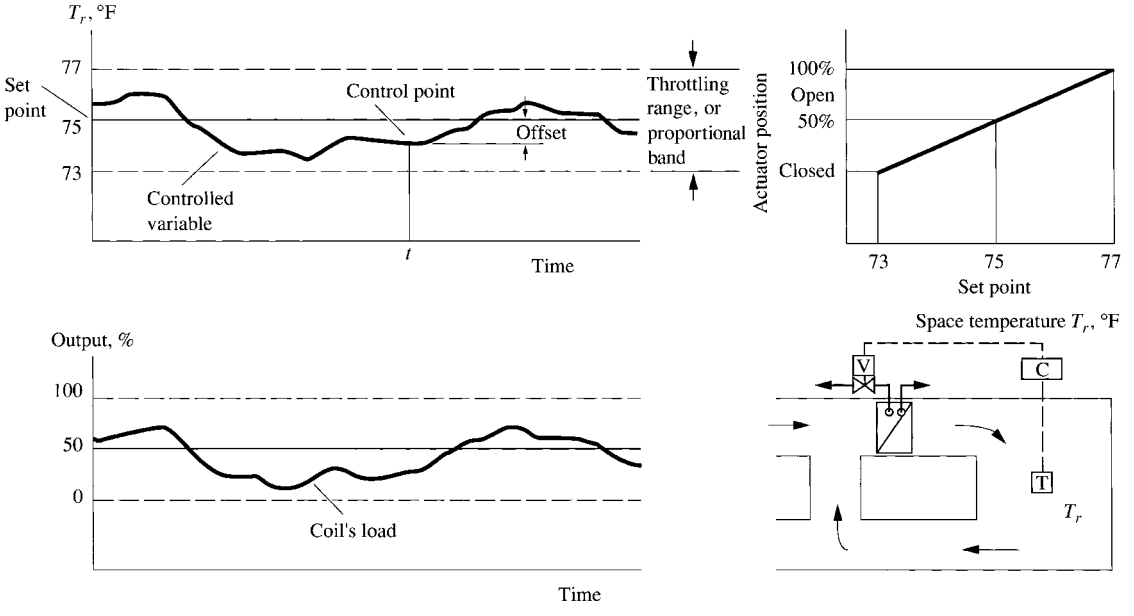


FIGURE 5.8 Proportional control mode.

The set point is the desired value of the controlled variable, or the desired control point that the controller seeks to achieve. The difference between the control point and the set point is called the *offset*, or *deviation*. In proportional control, when the controlled variable is at the bottom line of the throttling range, the controller will position the actuator at the closed position. At the set point, the actuator will be at 50 percent of the open position. When the controlled variable is at the top of the throttling range, the actuator will be at 100 percent of the open position.

In a proportional control mode, since the output signal V of the controller is proportional to the deviation of the control point from the set point, their relationship can be expressed as:

$$V = K_p e + M \tag{5.1}$$

where K_p = proportional gain, proportional to 1/(throttling range)
e = error signal, i.e., deviation or offset
M = output value when deviation is zero (usually, output value at middle of output range of controller)

For space and discharge air temperature control using proportional control, the offset is directly proportional to the space load and the coil load. The space or discharge temperature T, °F (°C), can be calculated as:

$$T = R_{\text{load}} T_{t,r} + T_{\min} \tag{5.2}$$

where $T_{t,r}$ = throttling range, expressed in terms of space or discharge temperature, °F (°C)
$T_{\min}$ = space and discharge temperature when space load or coil load is zero, °F (°C)

In Eq. (5.2), R_{load} represents the load ratio of the space load or coil load, which is dimensionless and can be evaluated as

$$R_{\text{load}} = \frac{\text{actual load}}{\text{design load}} \tag{5.3}$$

The actual load and design load must be in the same units, such as Btu/h (W).

For a proportional control mode, a certain degree of offset, or deviation, is inherent. The position of the controlled device is a function of the offset. Only when offset exists will the controller position the actuator and the valve or damper at a position greater or smaller than 50 percent open. The throttling range is primarily determined by the HVAC&R system characteristics and cannot be changed after the system is designed. A proportional control mode is suitable for an HVAC&R system that has a large thermal capacitance, resulting in a slow response and stable system and allowing comparatively narrower throttling range and thus a smaller offset. Fast-reacting systems need a large throttling range to avoid instability and short cycling, or hunting.

Proportional plus Integral (PI) Control

In a proportional plus integral control mode, a second component, the integral term, is added to the proportional action to eliminate the offset. The output of the controller can thus be expressed as

$$V = K_p e + K_i \int e \, dt + M \tag{5.4}$$

where K_i = integral gain and t = time.

In Eq. (5.4), the second term on the right-hand side (the integral term) indicates that (1) the error, or offset, is measured at regular time intervals and (2) the product of the sum of these measurements and K_i is added to the output of the controller to eliminate the offset. The longer the offset exists, the greater the response of the controller. Such a control action is equivalent to resetting the set point in order to increase the controller output to eliminate the offset. As such, PI control is sometimes called proportional plus reset control. Figure 5.9*a* shows the variations of a controlled

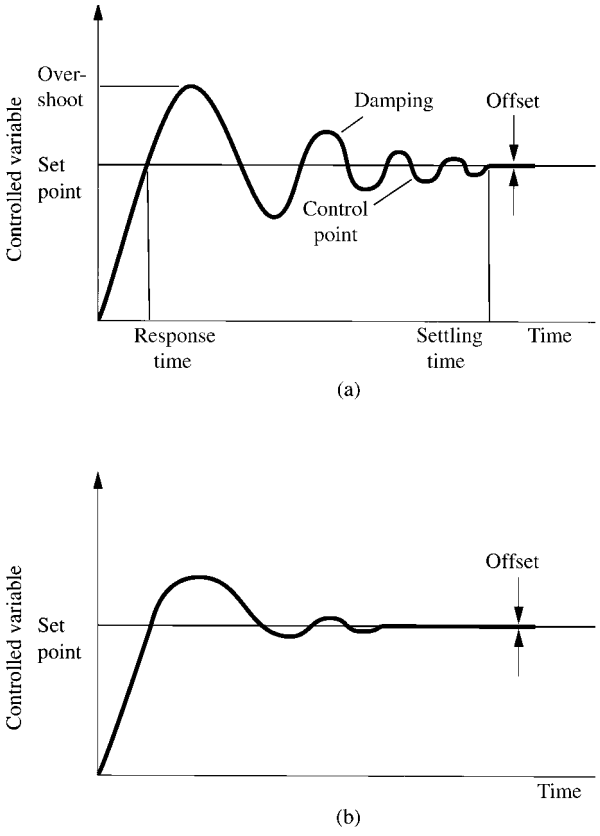


FIGURE 5.9 Proportional-integral and proportional-integral-derivative control modes. (a) Proportional-integral; (b) proportional-integral-derivative.

variable for a proportional-integral control mode. In PI control, proportional-integral control signals are additive.

For PI control, the controlled variable does not have any offset once it has achieved a stable condition, except due to any inaccuracy of the instrumentation and measurement. Proper selection of the proportional gain K_p and integral gain K_i as well as proper tuning is important for system stability and control accuracy. PI control may be applied to fast-acting control systems with a greater throttling range setting at the controller for better system stability, e.g., discharge air temperature, discharge chilled water temperature, and duct static pressure control systems.

Proportional-Integral-Derivative (PID) Control

A PID control mode has additional control action added to the PI controller: a derivative function that opposes to any change and is proportional to the rate of change. The output of such a controller can be described by the following equation:

$$V = K_p e + K_i \int e\, dt + K \frac{de}{dt} + M \tag{5.5}$$

where K is the derivative gain. The effect of adding the derivative function $K\,de/dt$ is that the quicker the control point changes, the greater the corrective action provided by the derivative function. Figure 5.9*b* shows the variation of the controlled variable for a PID control mode. As with a PI control mode, PID control mode also has no offset once the controlled variable has reached a stable condition, except due to instrument inaccuracy.

Compared to the PI control mode, the PID control mode, which is a combination of proportional, integral, and derivative actions, exhibits faster corrective action and a smaller overshoot and undershoot following an offset and a change of the controlled variable. The controlled variable is brought to the required set point in a shorter time. However, it is more difficult to determine properly three constants, or gains (K_p, K_i, and K).

Compensation Control, or Reset

Compensation control, or reset, is a type of control mode in which a compensation sensor is generally used to reset a main sensor to compensate for a variable change sensed by the compensation sensor. The purpose is to achieve operation that is more effective, energy-efficient, or both.

In the design of a reset mode, the first things to decide are the control point at which the main sensor will be reset and the variable to be sensed by the compensation sensor. The main sensor, which senses the mixed air temperature in the air-handling unit, is usually reset by a compensation sensor that senses the outdoor temperature, as shown in Fig. 5.10.

Another task to decide on is the relationship between the variables sensed by the main and compensation sensors—the reset schedule. Many reset schedules have a different relationship between these two sensors at various stages. For example, in Fig. 5.10, when the outdoor temperature T_o is less than 30°F (− 1.1°C), it is within the range of stage I. No matter what the magnitude of T_o is, the set point of the mixed temperature T_m is 65°F (18.3°C). When $30°F \le T_o \le 95°F$ ($-1.1°C \le T_o \le 35°C$), it is in stage II. The linear relationship between the temperatures sensed by the main and compensation sensors in Fig. 5.10 can be expressed by the following equation:

$$T_m = 65 - \frac{65 - 55}{95 - 30}(T_o - 30) \tag{5.6}$$

When $T_o > 95°F$ (35°C), the set point of the mixed temperature is always 55°F (12.8°C).

Compensation control modes have been widely adopted to reset space temperature, discharged air temperatures from air-handling units or packaged units, or water discharge temperatures from central plants.

Applications of Various Control Modes

Selection of a suitable control mode depends on

- Operating characteristics
- Process or system characteristics, such as whether the thermal capacitance should be taken into consideration
- Characteristics of load changes

If a simpler control mode can meet the requirements (say, two-position control versus PID control), the simpler control mode is always the first choice. Except for two-position and step control, all the other control modes are modulation controls. Modulation control is a control mode that is capable of increasing or decreasing a variable according to the deviation from the required value in small increments continuously. Also note that there is a significant difference between the two-position control, step control, floating control, and proportional control modes with an offset and the offset free PI and PID controls.

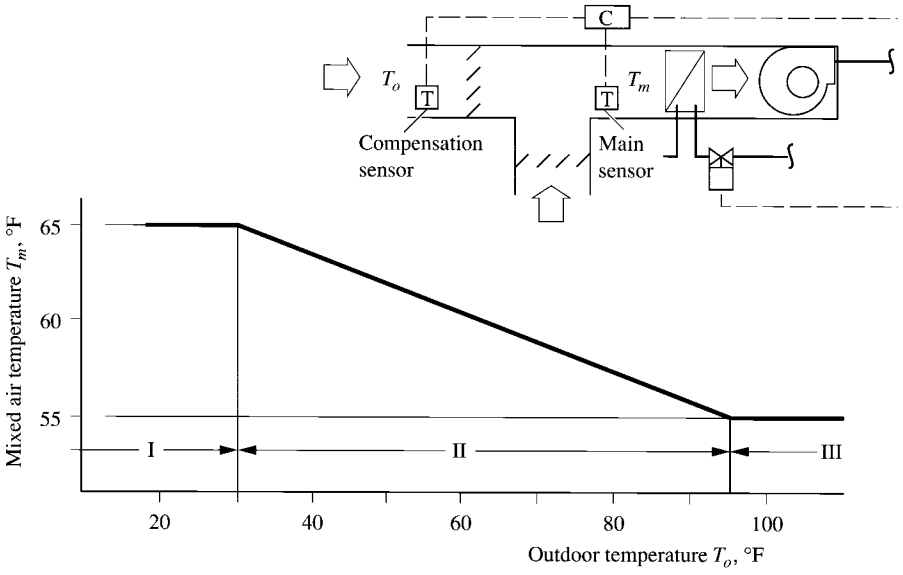


FIGURE 5.10 Reset control and schedule.

For HVAC&R processes, PI control can satisfy most of the requirements, and it is most widely used. Microprocessor-based PID control is a very powerful tool. PID control mode is more appropriate for fast-acting duct static pressure control and airflow control, and it is recommended to set the controller with a large proportional band for control system stability, a slow reset to eliminate deviation, and a derivative action to provide a quick response.

5.4 SENSORS AND TRANSDUCERS

A sensor is a device that acts as a component in a control system to detect and measure the controlled variable and to send a signal to the controller. A sensor consists of a sensing element and accessories, as shown in Fig. 5.11. The term *sensing element* often refers to that part of the sensor that actually senses the controlled variable. In HVAC&R systems, the most widely used sensors are temperature sensors, humidity sensors, pressure sensors, and flow sensors. Recently, CO_2 sensors, air quality sensors, and occupancy sensors are being used in many new and retrofit projects. Electronic sensors that send electric signals to electronic controllers, can also be used for the DDC units, and the current trend is to use solid-state miniaturizing sensing elements. Electric output from the sensors is usually expressed in 0 to 10 V dc or 4 to 20 mA. In the selection of sensors, accuracy, sensitivity of response, reliability, long-term stability or drift, required calibration intervals, maintainability, and especially the possibility of contamination by dust particles due to contact with the control medium should be considered. Drift is a kind of offset. ASHRAE's *Terminology* defines *drift* as "change in an output-input relationship over time with the change unrelated to input, environment, or load." A calibration period of 1 year and longer is often considered acceptable.

Location of the sensor affects the sensor output directly. A sensor should be located in the critical area where the controlled variable needs to be maintained within required limits. The sensing element of an air sensor should be well exposed to the air, so that air can flow through the sensor without obstruction. Air sensors should not be affected directly by the supply airstream nor should space air sensors be affected by the outdoor airstream. Sensors should be shielded from radiation

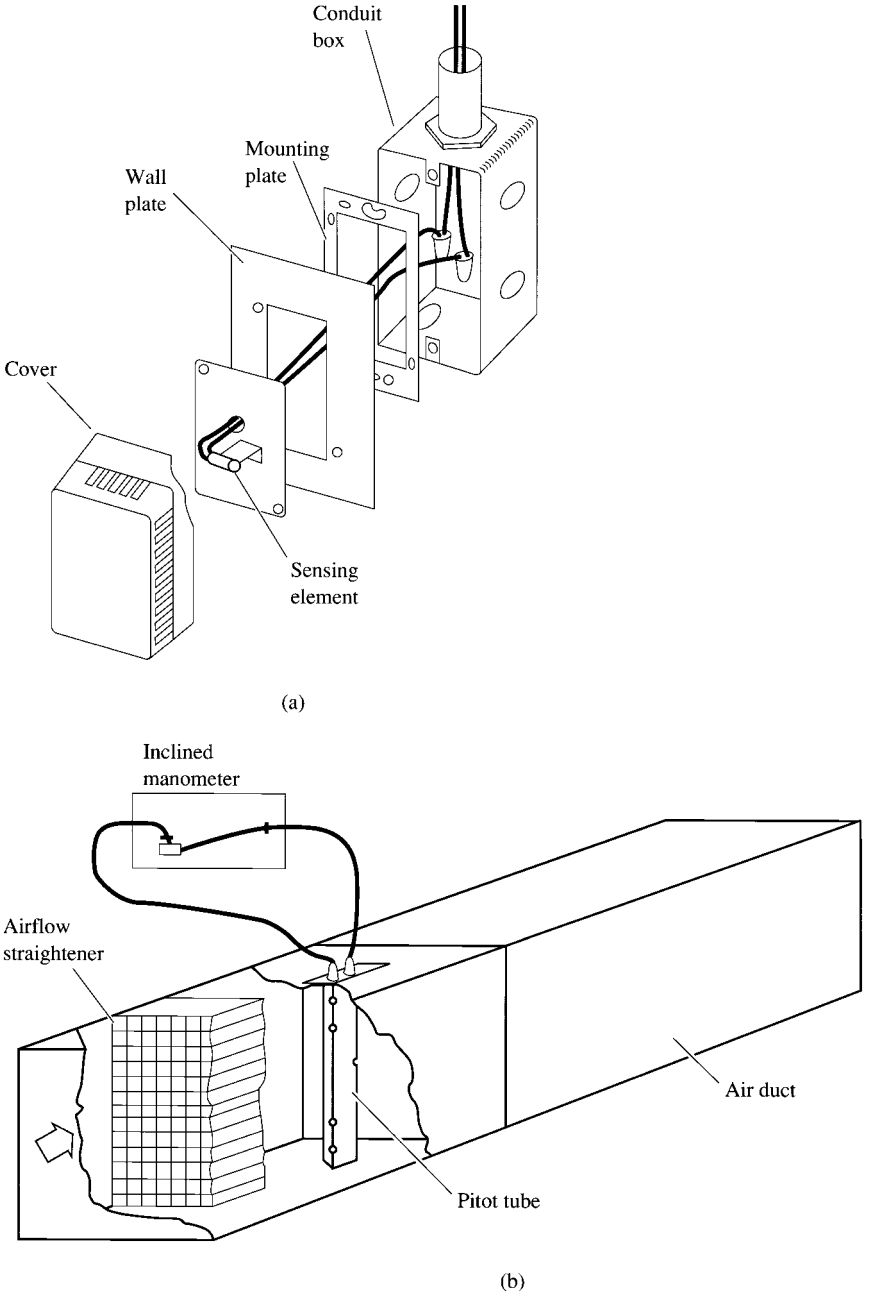


FIGURE 5.11 (*a*) A temperature sensor and (*b*) a typical pitot tube flow-measuring station.

and mounted firmly on a structural member or duct wall, free from vibration. If a space air sensor must be located on a concrete column, thermal insulation should be provided between the sensor and the column.

Temperature Sensors

Temperature sensors fall into two categories: those that produce mechanical signals and those that emit electric signals. Bimetal and rod-and-tube sensors that use sensing elements to produce a mechanical displacement during a sensed temperature change either open or close an electric circuit in an electric control system, as shown in Fig. 5.1, or adjust the throttling pressure by means of a bleeding nozzle in a pneumatic control system. For a sealed bellow sensor, a change in temperature causes a change in the pressure of a liquid in a remote bulb. The expansion and contraction of vapor then move the mechanism of the controller.

Temperature sensors that produce electric signals, as shown in Fig. 5.11*a*, are the same as the sensors for temperature measurement and indication mentioned in Sec. 2.3.

In addition to resistance temperature detectors (RTDs) and thermistors, sensors sometimes use thermocouples. A thermocouple uses wires of two dissimilar metals, such as copper and constantan, or iron and nickel, connected at two junctions, to generate an electromotive force between the junctions that is directly proportional to the temperature difference between them. One of the junctions is kept at constant temperature and is called the cold junction. Various systems have been developed to maintain the cold junction at a constant temperature and to provide compensation if the cold junction is not at 32°F (0°C). This task makes the use of thermocouples more complicated and expensive. The electromotive force produced between the two junctions can be used as the signal input to a controller.

Bimetal and rod-and-tube temperature sensors are simple and low in cost. However, they cannot provide temperature indication and electric signals for DDC and are often used in electric control systems. Platinum and nickel RTDs are stable, reliable, and accurate. They are very expensive compared to thermistors and need calibration to compensate for the effects of having external leading wires. RTDs are widely adopted in DDC for commercial applications. For a project that needs precise temperature control, RTD is often the choice. High-quality thermistors can provide stable, reliable, and interchangeable temperature sensors, and are also widely used in many commercial applications. Petze (1986) reported that some high-quality thermistors exhibited better than ± 0.002°F (± 0.001°C) stability for a 2-year period.

A typical space air temperature sensor has the following characteristics:

Sensing element	Platinum film element, 3000 Ω
	Positive temperature coefficient, 4.8 Ω/°F (8.64 Ω/°C)
Operating range	60 to 90°F (15.6 to 32.2°C)

Humidity Sensors

As mentioned in Sec. 2.8, humidity sensors fall into two categories: mechanical and electronic. When the same humidity sensor is used for both monitoring and DDC, the ion-exchange resistance-type, also called the *bulk polymer resistance-type*, and capacitance-type humidity sensors are often adopted. Humidity sensors usually have different accuracy at very low, 20 to 80 percent mid-range, and high relative humidities. The bulk polymer resistance humidity sensor is not accurate at low relative humidities and generally provides stable and accurate readings within a range of 30 to 90 percent relative humidity. Its performance is affected by the air contamination. Capacitance humidity sensors are accurate at 10 to 80 percent relative humidity. However, they become unstable at high relative humidities.

In an EMCS, an electronic humidity sensing element measures the relative humidity and sends an electric signal to the DDC unit. A typical space air humidity sensor has the following specifications:

Power source	24 V ac
Humidity range (active)	10 to 80 percent
Nominal output range	0 to 5 V dc
Accuracy at 70°F (21.1°C)	± 3 percent at 10 to 60 percent range
	± 4 percent at 60 to 80 percent range
Speed of response	8 min (90 percent of response time)
Mounting	Wall mount

Pressure Sensors

A pressure sensor usually senses the difference in pressure between the controlled medium (air and water) and a reference pressure; or the pressure differential across two points, such as the pressure differential across a filter. The reference pressure may be an absolute vacuum, atmospheric pressure, or the pressure at any adjacent point. Output signals from the pressure sensors may be electric or pneumatic, and analog or binary.

Pressure sensors used for HVAC&R systems can be divided into high-pressure and low-pressure sensors. High-pressure sensors measure in pound per square inch or feet WC (kilo pascals), and low-pressure sensors measure in inches WC (pascals). The sensing elements for high-pressure sensors are usually Bourdon tube, bellows, and sometimes diaphragms. For low-pressure air sensors, large diaphragms or flexible metal bellows are usually used.

To measure the duct static pressure, if a long section of straight duct of a length greater than 10 ft (3 m) is available, a single-point, pitot-tube type of duct static pressure sensor can be used. Otherwise, a multipoint pitot-tube array or flow-measuring station with airflow straighteners should be used. The small hole used to measure static pressure should never be directly opposite to an airstream with a velocity pressure that can affect the reading. A reference pressure should be picked up at a point with low air velocity outside the duct, at a point served by the same air system, or in the ceiling plenum.

In a DDC system, low air pressure is often sensed by measuring the capacitance of two diaphragms; one is allowed to move toward or away from the fixed one, depending on the pressure on two sides of the diaphragm. An electric circuit is used to convert the capacitance to a voltage or milliampere signal. The other method used to sense the low air pressure in DDC systems works because the air pressure differential compresses or stretches a diaphragm as well as a strain gauge. The change of the resistance of the gauge is detected and amplified, and this electric signal in the form of voltage or milliamperes is sent to the controller.

A space air pressure sensor having an operating range of + 0.1 to − 0.1 in WC (+ 25 Pa to − 25 Pa) detects the resistance of a silicon diaphragm, and thus the space air pressure is then measured. Stainless steel, rubber, etc., can also be used as the material of the diaphragms. A space air pressure sensor should be located in an open area of the conditioned space where the air velocity is less than 40 fpm (0.2 m/s) and where its reading is not affected by the opening of the doors. The reference pressure pickup is best located at the rooftop, at a level 10 ft (3 m) above the building to avoid the influence of wind.

Flow Sensors

Flow sensors usually sense the rate of air flow and water flow in cfm (L/s) for air and gpm (L/s) for water. For airflow sensors, the average velocity pressure p_v, in. WC (Pa), is often sensed and

measured. The average air velocity v_a, fpm, is then calculated as

$$v_a = 4005K\sqrt{p_v} \tag{5.7}$$

where K is the flow coefficient, which depends on the type of pitot-tube array used and the dimension of the round or rectangular duct. After that, the volume flow rate can easily be determined as the product v_aA. Here, A represents the cross-sectional area of the air passage, perpendicular to the airflow, in ft^2 (m^2). Various forms of pitot-tube array have been developed and tested to determine the average velocity pressure of a rectangular or circular duct section by measuring the difference between the total and static pressures of the airstream. A typical pitot-tube flow-measuring station with flow straighteners to provide more even airflow is shown in Fig. 5.11.

Electronic air velocity sensors such as hot-wire anemometers and thermistors have also been widely adopted to measure airflow, especially for variable-air-volume (VAV) boxes. Thermistors are cheaper than hot-wire anemometers. Heated thermistors need periodic calibration.

Water flow sensors of the differential-pressure type, such as orifice plates, flowing nozzles, and pitot tubes, have only a limited measurement range. Turbine or magnet-type flowmeters can apply to a wider range, but they are more expensive and also need periodic calibration.

Carbon Dioxide and Air Quality Sensors

A CO_2 sensor detects and indicates the amount of carbon dioxide (CO_2) contained inside the air, which is a reliable indication of the body odor released by the occupants of the conditioned space. Because of a certain relationship that exists between the CO_2 contained in the outdoor air and the CO_2 released by the occupants, the concentration of CO_2 in space air may sometimes be used as a rough indication of amount of outdoor ventilation air supply to the conditioned space under the specific conditions.

The sensing process used in CO_2 sensors includes potentiometric and amperometric electrochemical cells, an infrared detector, etc. A CO_2 sensor usually has an operating range of 0 to 3000 ppm. For measurements within an accuracy of $\pm$ 100 ppm, recalibration may be required on the order of once per year.

An air quality sensor, also called a volatile organic compound (VOC) or a mixed-gas sensor, is used to monitor and detect the relative concentrations of VOC or mixed gas, or total concentration including acetone, ammonia, CO, CO_2, SO_2, chlorine, formaldehyde, CFC-11, CFC-12, etc. The concentration of the contaminant is often expressed in tested and commissioned units, such as 0 to 5 or 0 to 10 units.

The sensing element has a tin dioxide surface which is heated to a temperature above 130°F (54°C). The change of the conductivity and thus its resistance are then amplified and fed to the controller in terms of 0 to 10 V dc in proportion to the contamination. Air quality sensors are less expensive than CO_2 sensors and need less maintenance. The drift of an air quality sensor is unpredictable.

Both CO_2 sensors and air quality sensors are now used for demand-based ventilation control to provide the required amount of outdoor air for acceptable indoor air quality.

Occupancy Sensors

An occupancy sensor detects whether a room is occupied by occupant(s). As a result, the HVAC&R and lighting in this room can be turned off when the room is not occupied, to save energy. There are two types of occupancy sensors: ultrasonic and infrared. An ultrasonic occupancy sensor sends out a rather lower level of ultrasonic wave and detects movement of the occupant when there are changes in receiving patterns. Ultrasonic sensors sweep over the area that surrounds the sensor. False signals may be received due to the motion of the papers or air movement from a diffuser. The sensitivity of the ultrasonic sensor should be adjusted to avoid these problems.

An infrared occupancy sensor senses the movement of the occupant(s) as the sensor receives heat from the occupant when that person is moving. Infrared sensors need to see the occupant. As long as there is no obstruction between the sensor and the occupant in the room, the sensing process of an infrared sensor is effective. Occupancy sensors are often mounted on the ceiling or at a high level on a wall. They are used in the guest rooms of the hotels and motels and other commercial buildings. Ultrasonic occupancy sensors show a wider acceptance than infrared occupancy sensors.

Wireless Zone Sensors and Intelligent Network Sensors

In an HVAC&R system, a wireless zone sensor is an indoor spread spectrum radio-frequency transmitter which sends room temperature and other status information to a local receiver located no more than 1000 ft (300 m) away. A translator converts these data and sends them to a variable-air-volume DDC unit controller through a wired communications link. This type of new technology was developed recently for the sake of providing flexibility for the rearrangements of the office layout.

Kovacs (1996) noted that intelligent network sensors ". . . can linearize the sensor signal, accept an offset adjustment through the network, may have alarm- or decision-making algorithms on board, and include self-test diagnostics to continuously validate performance." An intelligent network sensor may have a sensing element, an analog/digital transducer, a neutron chip—an onboard microprocessor, and a transceiver. Dew-point temperature, enthalpy, wet-bulb temperature, etc. can be provided as a combined variable. An intelligent network sensor will help to drive the system structure towards becoming a more cost-effective distributed control system in the future, a system that has more locally processed signal and data.

Transducers or Transmitters

A transducer is a device that converts energy from one form to another or amplifies an input or output signal. In HVAC&R control systems, a transducer may be used to convert an electric signal to a pneumatic signal (E/P transducer), e.g., a pneumatic proportional relay that varies its branch air pressure from 3 to 15 psig (20 to 103 kPa·g) in direct proportion to changes in the electrical input from 2 to 10 V. Also E/P transducers are used between microprocessor-based or electronic controllers and pneumatic actuators. However, a pneumatic signal can be converted to an electric signal in a P/E transducer. For example, a P/E relay closes a contact when the air pressure falls and opens the contact when the air pressure rises above a predetermined value. A transmitter is used to transmit a signal, pneumatic or electric; through air, water, or other fluids. A sensor is also a transmitter. However, the difference between them is that a sensor only senses the signal of the controlled variable and transmits it.

5.5 CONTROLLERS

A controller receives input from the sensor, compares with the set point or implemention based on its stored computer software, sends an output to or modulates the control device for maintaining a desirable indoor environment. A thermostat is a combination of a temperature sensor and a temperature controller, whereas a humidistat is a combination of a humidity sensor and a humidity controller.

Direct-Acting or Reverse-Acting

A direct-acting controller increases its output signal upon an increase in the sensed controlled variable, and it decreases its output signal upon a decrease in the sensed controlled variable. Conversely,

a reverse-acting controller decreases its output signal upon an increase in the sensed controlled variable and increases its output signal upon a decrease in the controlled variable.

Normally Closed or Normally Open

A controlled device, a valve or a damper, that is said to be in the normally closed (NC) position indicates that when the input signal to the controller falls to zero or below a critical value, the control device will be in the closed position. The closing of the valve or damper is most likely due to the action of a spring or a supplementary power supply. A valve or a damper that is said to be normally open (NO) indicates that when the input signal to the controller falls to zero or below a critical value, then the valve, damper, or associated process plant will be in the open position.

Pneumatic Controllers

In a pneumatic controller, the basic mechanism used to control the air pressure in the branch line supplied to the actuators is a nozzle-flapper assembly plus a restrictor. Figure 5.12 shows a pneumatic controller with such a mechanism. Compressed air supplied from the main line flows through the restrictor and discharges at the opening between the nozzle and the flapper, which has a spring pulling it downward. The nozzle and restrictor are sized in such a manner that when the flapper moves away from the nozzle, all the air escapes from the nozzle and the branch line pressure is zero. When the flapper covers the nozzle or if there is no airflow in the branch line, the pressure of the branch line is equal to that in the main line.

If a sensor, such as a bimetal sensor, moves the flapper upward according to the magnitude of the controlled variable sensed by the sensor, then the input signal from the sensor determines the opening between the nozzle and flapper and hence the compressed air pressure in the branch line.

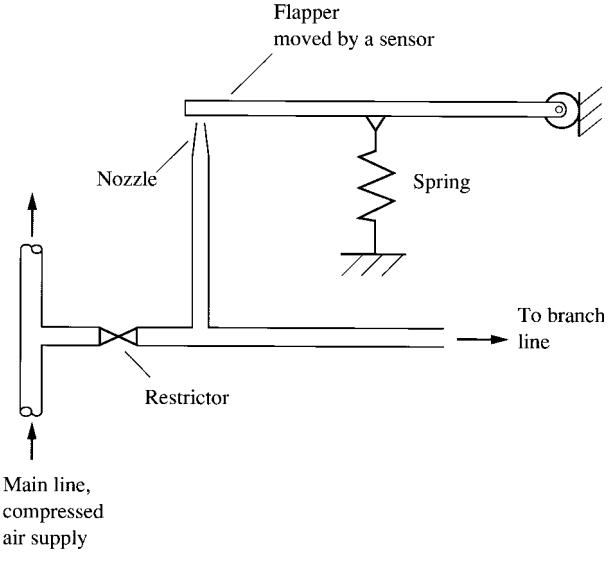


FIGURE 5.12 A pneumatic controller with a nozzle-flapper assembly and a restrictor.

For a direct-acting pneumatic temperature controller used to control the space temperature during summer, the branch line pressure may change from 9 to 13 psig (62 to 89 kPag) when the space temperature increases from 73 to 77°F (22.8 to 25°C). This nozzle-flapper assembly pneumatic controller operates in proportional control mode. Many other more complicated pneumatic controllers have been developed to perform other control modes and additional functions.

Electric and Electronic Controllers

An electric controller uses switches, relays, and a bridge circuit formed by potentiometers to position the actuators in on-off, floating, and proportional control modes according to the input signal from the sensor and the predetermined set point. An electronic controller can provide far more functions than electric controllers can. It may receive input signals from both the main sensor and the compensation sensor with amplification and combination.

In the control circuit, an electronic controller basically provides proportional or proportional-integral control modes. The output signal from the controller can be used to position an actuator or to provide the sequencing of actuators, or to change to two-position, floating, or even PID control modes in conjunction with additional circuits.

Direct Digital Controllers

A direct digital controller has a microprocessor to implement computer programs to provide various control functions. In DDC units, there are analog-to-digital (A/D) and digital-to-analog (D/A) converters to convert analog input to digital signals for processing, or to convert digital signals to analog for actuators, if necessary. DDC units are stand-alone and microprocessor-based controllers. *Stand-alone* means that the controller has sufficient capacity to execute the assigned control functions alone. Today, there are mainly two types of DDC units: system controllers and unit controllers.

System Controllers. A system controller, also called a stand-alone panel (SAP), as shown in Fig. 5.13, has the ability to coordinate communications between system controllers, between the system controller and the personal computer (PC) in the workstation, and between the system controller and the supported unit controllers. It also has the ability to provide and execute the control programs for functional control, and to store user databases and trend log values. A system controller can support 50 to 200 unit controllers on separate unit controller trunk(s).

Unit Controllers. A unit controller, also called a terminal controller, is shown in Fig. 5.14. It usually has limited capacity to execute factory-loaded computer programs and to provide functional control for a terminal or a piece of HVAC&R equipment. Unit controllers are often connected in a separate network and supported by a system controller. The new-generation unit controllers have greater memory to handle complicated control programs, and they provide time and calendar scheduling, data storage, and other functions, such as limited programming.

Hardware. Many system controllers have only a single printed-circuit board. Single-board configuration often offers lower first cost. Its disadvantage is that any component failure requires the replacement of the complete board. Another approach is that the controller is made from various modules. The modular approach isolates the component failures and plugs on the single-module replacement quickly and inexpensively.

Memory. The types of memory included in DDC units are as follows:

- Read-only memory (ROM), which stores the software provided by the manufacturer and should not be modified by the user.

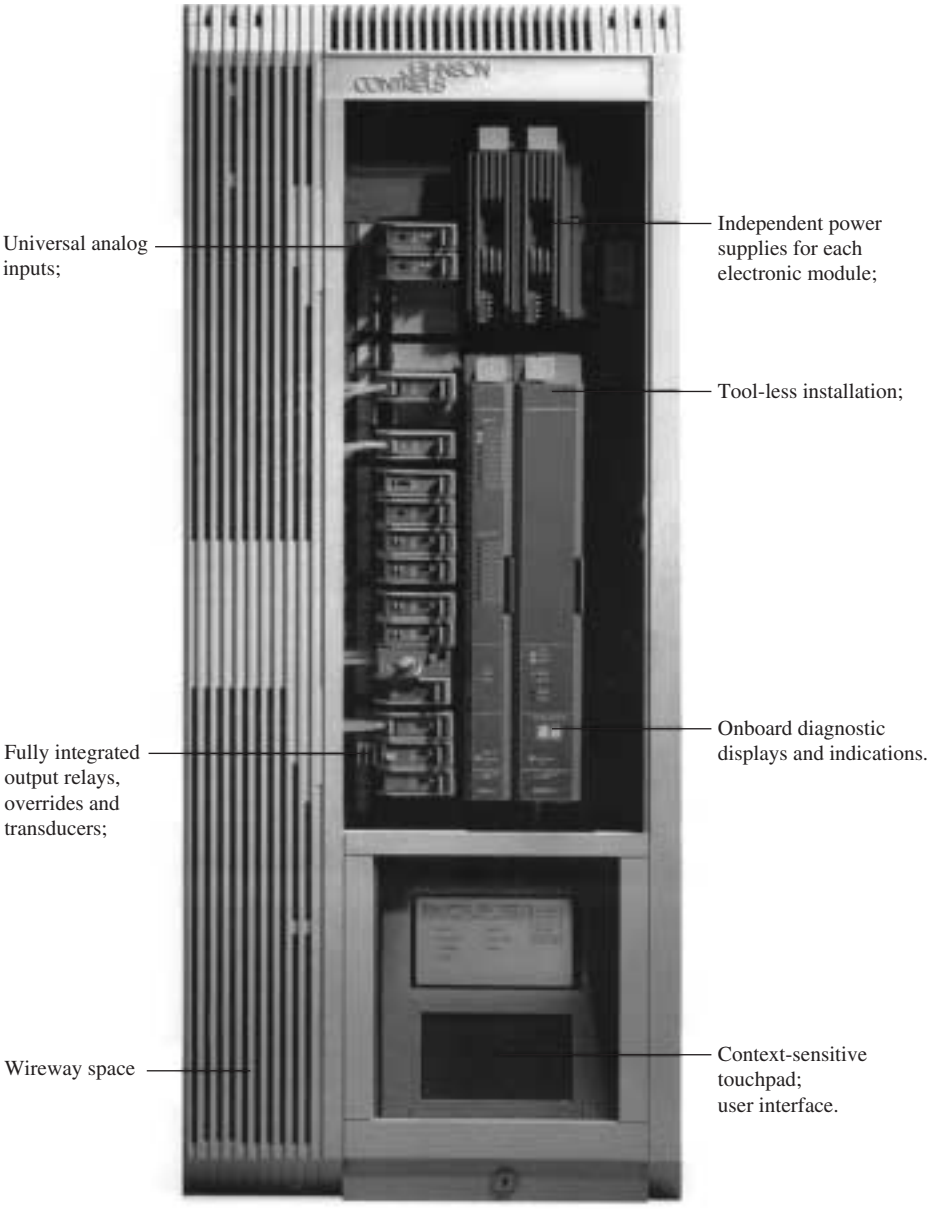

Universal analog inputs;
Independent power supplies for each electronic module;
Tool-less installation;
Fully integrated output relays, overrides and transducers;
Onboard diagnostic displays and indications.
Wireway space
Context-sensitive touchpad; user interface.

FIGURE 5.13 A typical system controller. (*Source: Johnson Controls. Reprinted by permission.*)

- Random-access memory (RAM), which stores custom control software developed during installation or prepared by the user. This type of memory is volatile (i.e., it can be read from and written to) and requires battery backup.
- Electric erasable programmable read-only memory (EEPROM), which stores custom control software and is volatile. The advantage of EEPROM over RAM is that EEPROM does not need

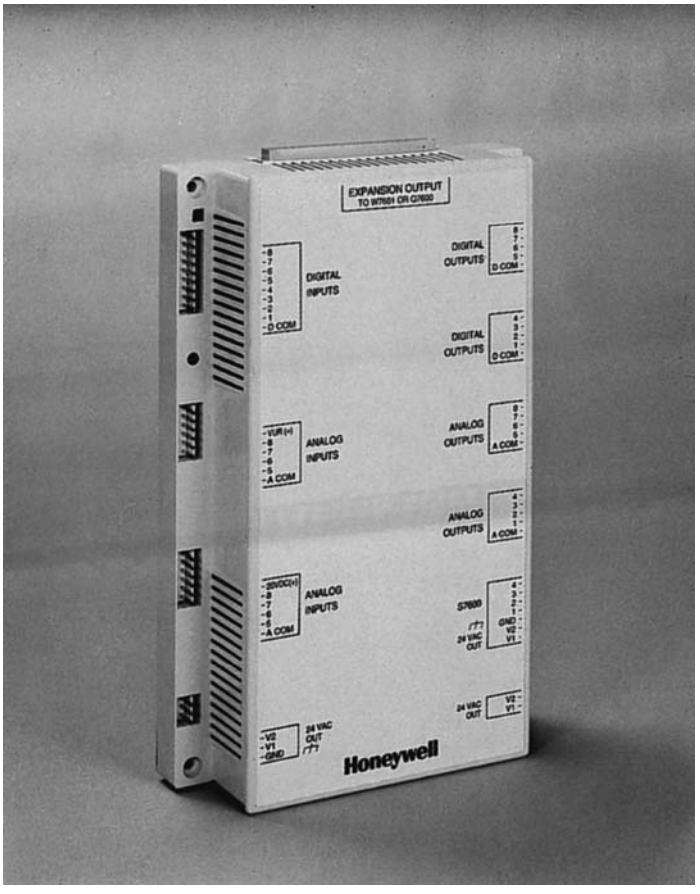


FIGURE 5.14 A typical unit controller. (*Source: Honeywell Inc. Reprinted by permission.*)

battery backup. However, EEPROM cannot be used for as many writing, erasing, and rewriting cycles as RAM.

- Flash erasable programmable read-only memory (flash EPROM), which is a kind of new memory technology and allows the stored control software to remain untouchable indefinitely without power.

System controllers often use a 16-bit microprocessor. A typical system controller has the following memories:

RAM:	256 Kbytes
ROM:	128 Kbytes EEPROM
	512 Kbytes flash EPROM

Input/Output (I/O). There are four kinds of I/O: analog input (AI); digital or binary input (BI); analog output (AO); and pulsed or binary output (BO). Conventionally, a sensor input, a controller output, or a control value, is referred to as a *point* or *object*. Current system controllers allow their

input and output connections to be configured with great flexibility. Each input or output point can be either analog or digital. Typical analog inputs (electric signal) are 0 to 10 V dc or 4 to 20 mA. A system controller often has a total of 20 to 50 I/O points. Some system controllers can be extended to 100 points if necessary. A typical system controller has the following I/O points capacity:

Analog/digital inputs	18
Universal analog/digital input/outputs	6
Digital outputs	12
Totalizer inputs, i.e., pulsed inputs	4

A unit controller or a terminal controller usually has 4 to 20 I/O points. The I/O points in a typical unit controller may take the following forms:

Analog inputs	0 to 10 V dc
Digital inputs	Switch, relay, and transistor
Microbridge sensor	0 to 3 in. WC pressure differential (0 to 750 Pa)
Analog outputs	0 to 10 V dc, 4 to 20 mA
Digital outputs	30 V ac
Pneumatic	3 to 15 psig (20.6 to 103 kPag)
Triac	On/off output for electric heater, fan motor, etc.

5.6 WATER CONTROL VALVES AND VALVE ACTUATORS

Water valves are used to regulate or stop water flow in a pipe either manually or by means of automatic control systems. Water control valves adopted in water systems can modulate water flow rates by means of automatic control systems.

Valve Actuators

An actuator, sometimes called an operator, is a device which receives an electric or pneumatic analog control signal from the controller, either directly or through a digital-to-analog converter. It then closes or opens a valve or damper, modulating the associated process plant, and causes the controlled variable to change toward its set point. Valve actuators are used to position control valves. They are mainly of the following types:

Solenoid Actuators. These use a magnetic coil to move a movable plunger connected with the valve stem. Most solenoid valve actuators operate at two positions (on and off). They are used mainly for small valves.

Electric Actuators. These move the valve stem by means of a gear train and linkage. Different electric motor valve actuators can be classified according to the control mode they use:

1. *On/off mode*. For this type of actuator, the motor moves the valve in one direction, and when the electric circuit breaks, the spring returns the valve stem to the top position (either open or closed position depending on whether it is a normally open or closed valve).
2. *Modulating mode*. The motor can rotate in both directions, with spring return when the electric circuit breaks.
3. *Modulating mode with supplementary power supply*. The motor rotates in two directions and without a spring-return arrangement. When the power is cut off, a bypass signal is usually sent

to the electric motor to drive the valve to its open or closed position, depending on whether it is a normally open or closed valve.

It may take minutes to fully open a large valve using an electric motor valve actuator. Modern electronic actuators use solid-state control boards to determine the speed, the action, and other functions to meet more demanding requirements.

Pneumatic Actuators. A pneumatic valve actuator consists of an actuator chamber whose bottom is made of a flexible diaphragm or bellows connected with the valve stem. When the air pressure in the actuator chamber increases, the downward force overcomes the spring compression and pushes the diaphragm downward, closing the valve. As the air pressure in the actuator chamber decreases, the spring compresses the diaphragm, moving the valve stem and valve upward. A pneumatic valve actuator is powerful, simple, and fast to respond.

Because of the increasing popularity of the DDC systems, there is an increasing demand for electric actuators that can be interfaced with a DDC system.

Types of Control Valves

Water control valves consist mainly of a valve body, one or two valve disks or plugs, one or two valve seats, a valve stem, and a seal packing. Based on their structure, water control valves can be classified into the following types:

1. *Single-seated*. A single-seated valve has only a single valve disk and seat, as shown in Fig. 5.15*b* and *c*. It is usually used for water systems that need a tight shutoff.
2. *Double-seated*. A double-seated valve has two valve disks connected to the same valve stem and is designed so that the fluid pressure exerted on the valve disks is always balanced. Consequently, less force is required for the operation of a double-seated valve, as shown in Fig. 5.15*a*.
3. *Butterfly.* A butterfly valve consists of a cylindrical body, a shaft, and a disk that rotates on an axis, as shown in Fig. 5.15*d*. When the valve closes, it seats against a ring inside the body. A butterfly valve exhibits low flow resistance when it is fully opened. It is compact and is usually used in large water pipes.

According to the pattern of the water flow, water control valves can again be classified as two-way valves or three-way valves. A two-way valve has one inlet port and one outlet port. Water flows straight through the two-way valve along a single passage, as shown in Fig. 5.15*a*.

In a three-way valve, there are three ports: two inlet ports and one common outlet port for a three-way mixing valve, as shown in Fig. 5.15*b*, and one common inlet port and two outlet ports for a three-way diverting valve, as shown in Fig. 5.15*c*. In a three-way mixing valve, the main water stream flows through the coil or boiler, and the bypass stream mixes with the main stream in the common mixing outlet port. In a three-way diverting valve, the supply water stream divides into two streams in the common inlet port. The main water stream flows through the coil, and the bypass stream mixes with the main water stream after the coil. A three-way mixing valve is always located downstream of the coil. But a diverting valve is always located upstream of the coil.

A diverting valve should never be used as a mixing valve. The unbalance pressure difference between the two inlet ports and the outlet port at a closed position may cause disk bouncing and valve wear when the valve disk travels between the two extremes.

Valve Characteristics and Ratings

The different types of control valves and the characteristics that are important during the selection of these valves are as follows:

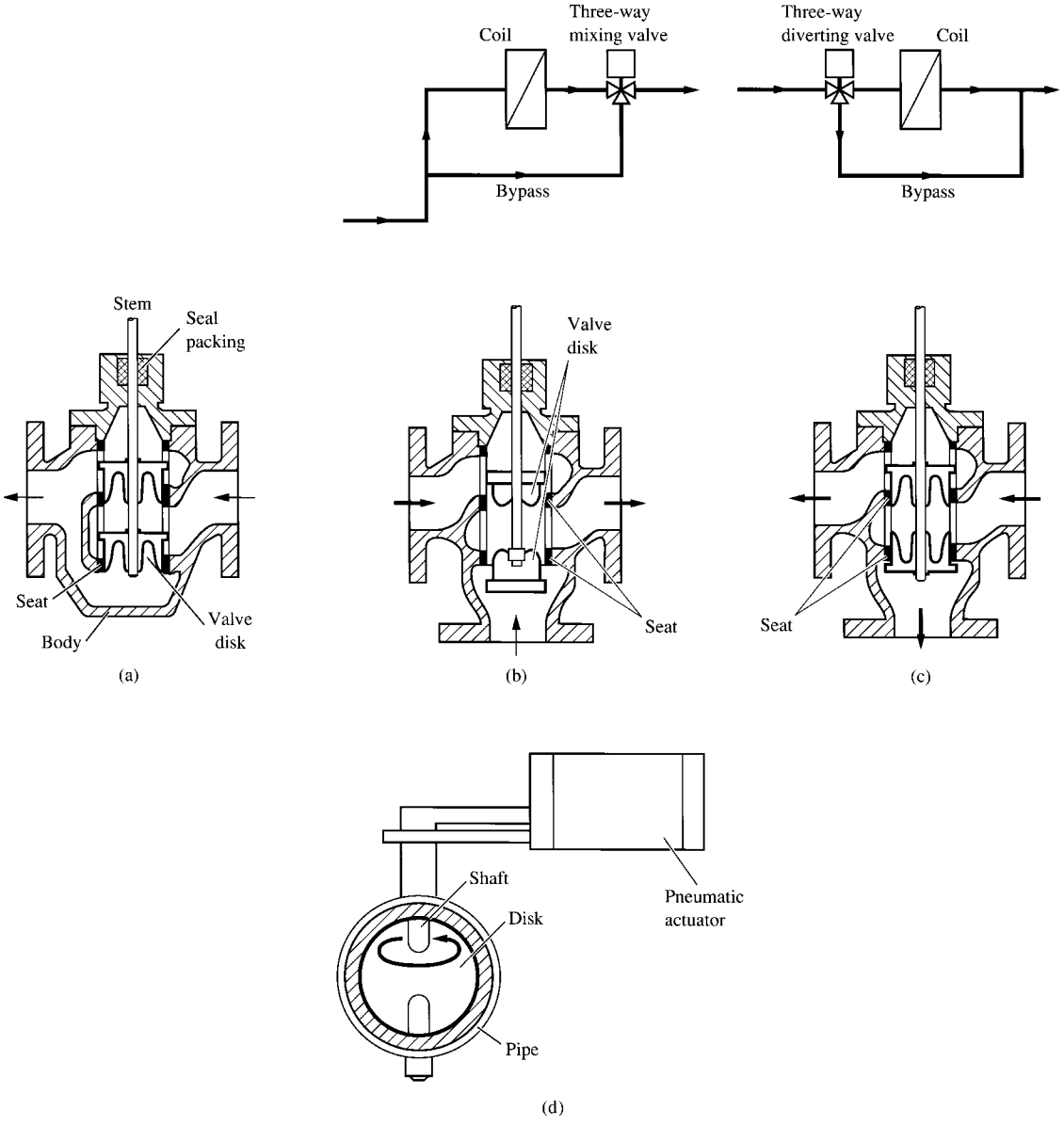


FIGURE 5.15 Various types of control valves: (*a*) Double-seated two-way valve; (*b*) single-seated three-way mixing valve; (*c*) single-seated three-way diverting valve; (*d*) butterfly valve.

1. *Equal-percentage valve*. This control valve changes the water flow rate by a certain percentage for that same percentage of lift in the valve stem when the upstream versus downstream water pressure difference across the valve (its pressure drop) is constant.
2. *Linear valve*. This control valve shows a directly proportional relationship between the flow rate and the lifting of the valve stem for a constant pressure drop.

3. *Quick-opening valve*. This control valve gives the maximum possible flow rate when the valve disk or plug is just lifted from its seat.

Rangeability is defined as the ratio of the maximum flow rate to the minimum flow rate under control. An equal-percentage valve may have a very good rangeability of 50 : 1. A linear valve may have a rangeability of 30 : 1.

The flow characteristics of equal-percentage, linear, and quick-open valves are shown in Fig. 5.16.

The following control valve ratings should be considered during the selection and sizing of a valve:

1. *Body rating*. The nominal body rating of the valve is the theoretical rating of the valve body only, in psig. The actual body rating is the permissible safe water pressure for the valve body, in psig (kPa · g), at a specific water temperature.
2. *Close-off rating*. That is the maximum pressure difference between the inlet and outlet ports that a valve can withstand without leakage when the valve is fully closed, in psi (kPa).
3. *Maximum pressure and temperature*. These are the maximum pressure and temperature of water that the whole valve, including body, disk, seat, packing, etc., can withstand.

Valve Selection

Proper selection of water control valves depends on water system performance, load variations, pipe size, control modes, etc. Today, the use of scaling factors on analog outputs in a DDC system permits a nonlinear device to provide an output of linear response. However, select a control valve having a linear relationship between a change in the controlled variable and the amount of travel of the valve stem, or a linear system control characteristic over the operating range is still desirable when it is cost-effective. Hence, a linear valve is often used for the water system for which the controlled variable has

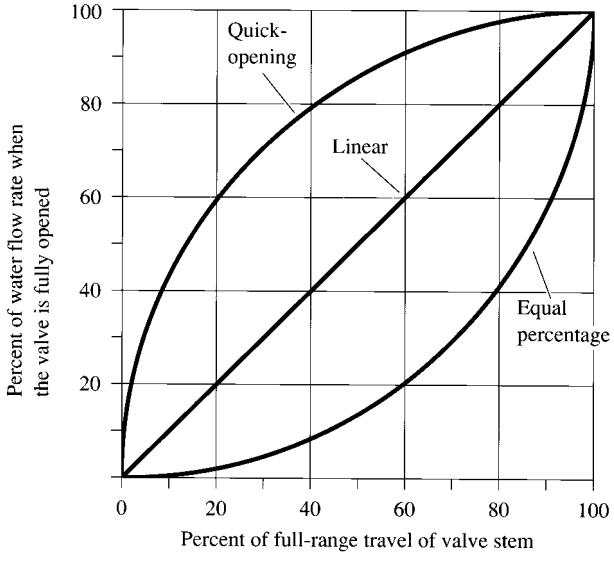


FIGURE 5.16 Typical flow characteristics of various types of control valve.

a linear relationship with the water flow or valve opening; or for applications that do not have wide load variations.

When a control valve is used to modulate the water flow rate of a hot or chilled water coil, a large reduction in the flow rate causes only a small reduction in the heating or cooling output of the coil. Given such circumstances, the nonlinear behavior of an equal-percentage valve combined with the nonlinear output performance of a hot or chilled water coil will provide more linear system behavior.

When three-way valves are used, the water flow rate before or after the common port is approximately constant, no matter how wide the openings of the various ports in the three-way valves. As such, three-way valves are used in constant or approximately constant water flow rate systems, even the coil load changes.

As a two-way valve closes, the flow rate of the water system decreases and its pressure drop across the two-way valve increases. A two-way valve is thus used for water systems that have variable volume flow during a variation in system load, as shown in Fig. 5.17.

Valve Sizing

The size of a control valve affects the controllability of a water system. If a control valve is oversized, then the smallest increment possible may overshoot the controlled variable. An undersized control valve needs great pumping power. The size of a control valve is also closely related to its

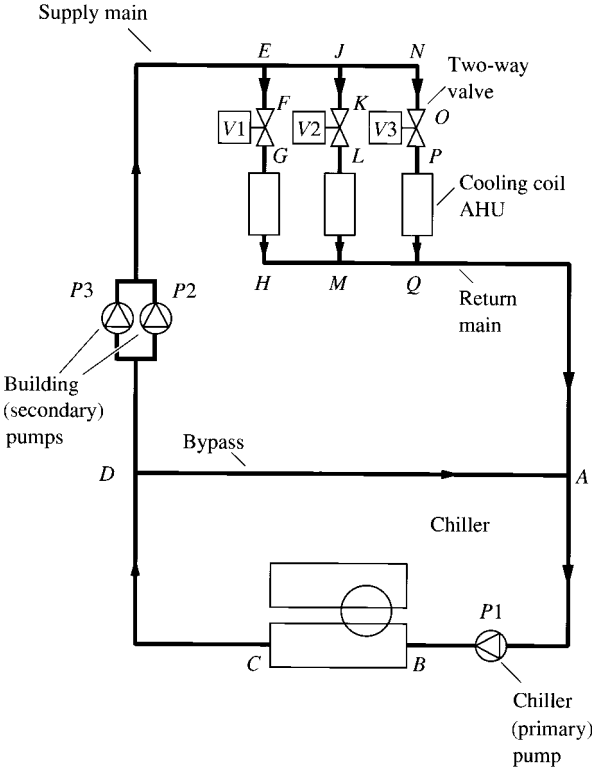


FIGURE 5.17 A typical chilled water system.

design water flow rate $\dot{V}$, gpm (L/s), and the pressure drop across the valve Δp_{vv}, psi (kPa), when the control valve is fully opened. Their relationship can be expressed as

$$\dot{V} = C_v\sqrt{\Delta p_{vv}}$$

$$C_v = \frac{\dot{V}}{\sqrt{\Delta p_{vv}}} \tag{5.8}$$

where C_v is the flow coefficient for a flow rate of 1 gpm at a pressure drop of 1 psi. The flow coefficient of control valves can be found in manufacturers' catalogs.

For a water system that has several cooling coils between supply and return mains, as shown in Fig. 5.17, modulating the water flow is effective only when the opening and closing of the control valve *V*1 in piping section *EH* has a significant effect on the change in the pressure drop Δp_{EH} across piping section *EH* between the supply and return mains. As control valve *V*1 opens wider and Δp_{EH} decreases, if there is no concurrent change in flow resistance of piping sections *JM* and *NQ*, then a greater water flow will pass through piping section *EH*. If Δp_{EH} increases, less water will flow through *EH*. If Δp_{EH}, Δp_{JM}, and Δp_{NQ} all increase, based on the characteristic curve of the centrifugal pump, the water flow rate through pumps *P*2 and *P*3 will decrease for an increase in the system head. If Δp_{EH}, Δp_{JM}, and Δp_{NQ} all drop, the water flow rate through pump *P*2 or *P*3 will increase.

Because of the use of DDC systems with PI or PID control modes and the variable-speed pumping systems, as well as the results of the previous analysis the following hold:

1. The size of a control valve should be determined according to the flow coefficient calculated from Eq. (5.8) and listed in the manufacturer's catalog.
2. The assumed pressure drop Δp_{vv} across the control valve should be appropriate. It is affected by the type of control valve used. It is also a compromise between a higher Δp_{vv} value to provide desirable controllability and a lower Δp_{vv} value to save energy.
3. For a water system using variable-speed pumping and DDC systems with PI or PID control modes, a pressure drop across an equal-percentage control valve Δp_{vv} = 5 to 10 ft WC (15 to 30 kPa) and a rangeability of 30 : 1 or greater is recommended for energy saving.

Example 5.1. A water system using variable-speed pumping with DDC supplies chilled water to the cooling coils of three air-handling units (AHUs), as shown in Fig. 5.17. If the chilled water flowing through two-way valve *V*1 is at 100 gpm (6.31 L/s), select and size control valve *V*1.

Solution.

1. To achieve a nearly linear relationship between the coil load and the travel of the valve stem, an equal-percentage valve is selected for valve *V*1.
2. For a water system using variable-speed pumping with DDC, it is assumed that the pressure drop across the control valve Δp_{vv} = 5 ft WC, or $5 \times 0.433 = 2.165$ psi (15 kPa). From Eq. (5.8), the flow coefficient is

$$C_v = \frac{V}{\sqrt{\Delta p_{vv}}} = \frac{100}{\sqrt{2.165}} = 68$$

From one of the manufacturers' catalog, for C_v = 68, the size of equal-percentage control valve *V*1 is 3 in. (76 mm).

3. From the friction chart of water in steel pipes, for a chilled water flow rate of 100 gpm (6.31 L/s) and a piping head loss of 2.5 ft/100 ft (2.5 m/100 m) of length, the diameter of the branch piping section *EH* is also 3 in. (76 mm). The size of the control valve and the diameter of the branch pipe are the same.

5.7 DAMPERS AND DAMPER ACTUATORS

A damper is a device that controls the airflow in an air system or ventilating system by changing the angle of the blades and therefore the area of its flow passage. In HVAC&R systems, dampers can be divided into volume control dampers and fire dampers. Fire dampers are covered in a later section. In this section, only volume control dampers are discussed.

Types of Volume Control Dampers

Volume control dampers can be classified as single-blade dampers or multiblade dampers according to their construction. Various types of volume control dampers are shown in Fig. 5.18.

Butterfly Dampers. A butterfly damper is a single-blade damper. A butterfly damper is made from either a rectangular sheet mounted inside a rectangular duct or a round disk placed in a round duct, as shown in Fig. 5.18*a*. It rotates about an axle and is able to modulate the air volume flow rate of the duct system by varying the size of the opening of the passage for air flow.

Gate Dampers. A gate damper is a single-blade damper. It also may be rectangular or round. It slides in and out of a slot in order to shut off or open up a flow passage, as shown in Fig. 5.18*b*. Gate dampers are mainly used in industrial exhaust systems with high static pressure.

Split Dampers. A split damper is also a single-blade damper. It is a piece of movable sheet metal that is usually installed at the Y connection of a rectangular duct system, as shown in Fig. 5.18*c*.

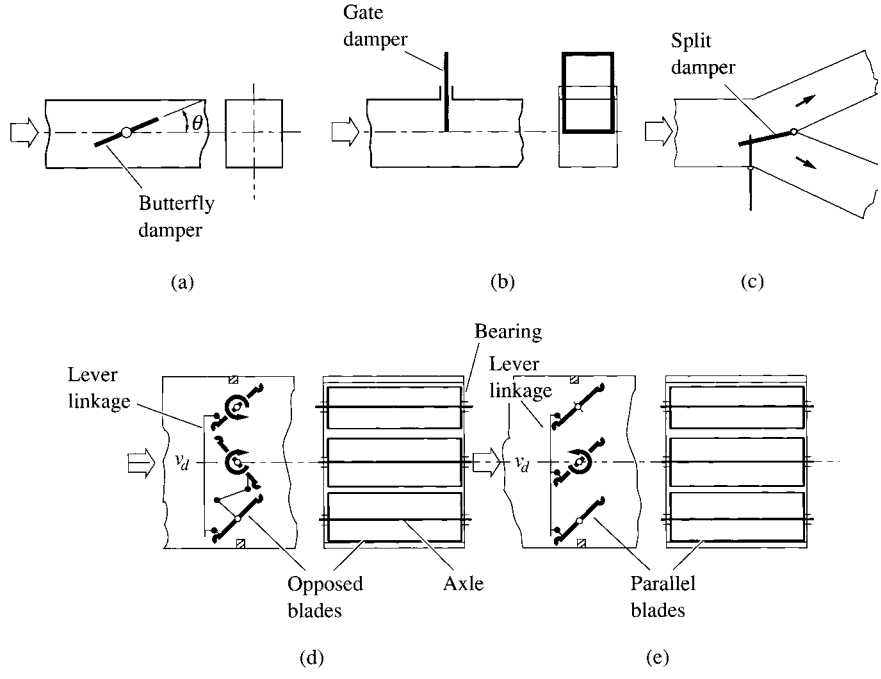


FIGURE 5.18 Various types of volume control dampers: (*a*) Butterfly damper; (*b*) gate damper; (*c*) split damper; (*d*) opposed-blade damper; (*e*) parallel-blade damper.

The movement of the split damper from one end to the other modulates the volume of air flowing into the two legs or branches. A split damper is usually modulated only during air balancing after installation or during periodic air balancing.

Opposed-Blade Dampers. An opposed-blade damper is a type of multiblade damper that is often rectangular, as shown in Fig. 5.18*d*. It is usually used for a flow passage of large cross-sectional area. The damper blades may be made of galvanized steel, aluminum alloy, or stainless-steel sheets, usually not exceeding 10 in. (25.4 cm) in width. Rubber or spring seals can be provided at the fully closed position to control the air leakage rating, which often does not exceed 6 cfm/ft^2 (30 L/s · m^2) at a pressure drop across the damper of 4 in. WC (1000 Pa). The bearing used for supporting the blade axle should be made of a corrosion-resistant material such as copper alloy or nylon. Teflon-coated bearings may also be used to ensure smooth operation of the damper. Lever linkages are used to open and close the damper blades. The characteristics of the opposed-blade dampers are covered later in this section.

The maximum static pressure drop across closed opposed-blade dampers is 6 in. WC (1500 Pa) for a 36-in.- (914-mm-) long damper (the length of the damper blade) and 4 in. WC for a 48-in.- (1219-mm-) long damper.

Parallel-Blade Dampers. A parallel-blade damper is also a type of multiblade damper used mainly for large cross-sectional areas, as shown in Fig. 5.18*e*. The blade material and the requirement for the seals and bearings are the same as those for opposed-blade dampers.

Damper Actuators (Motors)

Damper actuators, also called damper motors, are used to position dampers according to a signal from the controller. As with valve actuators, damper motors can be classified as either electric or pneumatic.

Electric Damper Motors. These either are driven by electric motors in reversible directions or are unidirectional and spring-returned. A reversible electric motor is used often for more precise control. It has two sets of motor windings. When one set is energized, the motor's shaft turns in a clockwise direction; and when the other set is energized, the motor's shaft turns in a counterclockwise direction. If neither motor winding is energized, the shaft remains in its current position. Such an electric motor can provide the simplest floating control mode, as well as other modes if required.

Pneumatic Damper Motors. Their construction is similar to that of pneumatic valve actuators, but the stroke of a pneumatic damper motor is longer. They also have lever linkages and crank arms to open and close the dampers.

Volume Flow Control between Various Airflow Paths

For air conditioning control systems, most of the dampers are often installed in parallel connected airflow paths to control their flow volume, as shown in Fig. 5.19. The types of airflow volume control are as follows:

Mixed-Air Control. In Fig. 5.19*a*, there are two parallel airflow paths: the recirculating path *um* in which a recirculating air damper is installed and the exhaust and intake path *uom*, in which exhaust and outdoor air dampers are installed. The outdoor air and the recirculating air are mixed together before entering the coil. Both the outdoor damper and the recirculating damper located just before the mixing box (mixed plenum) are often called mixing dampers. The openings of the outdoor and recirculating dampers can be arranged in a certain relationship to each other. When the outdoor damper is at minimum opening for minimum outdoor air ventilation, the recirculating

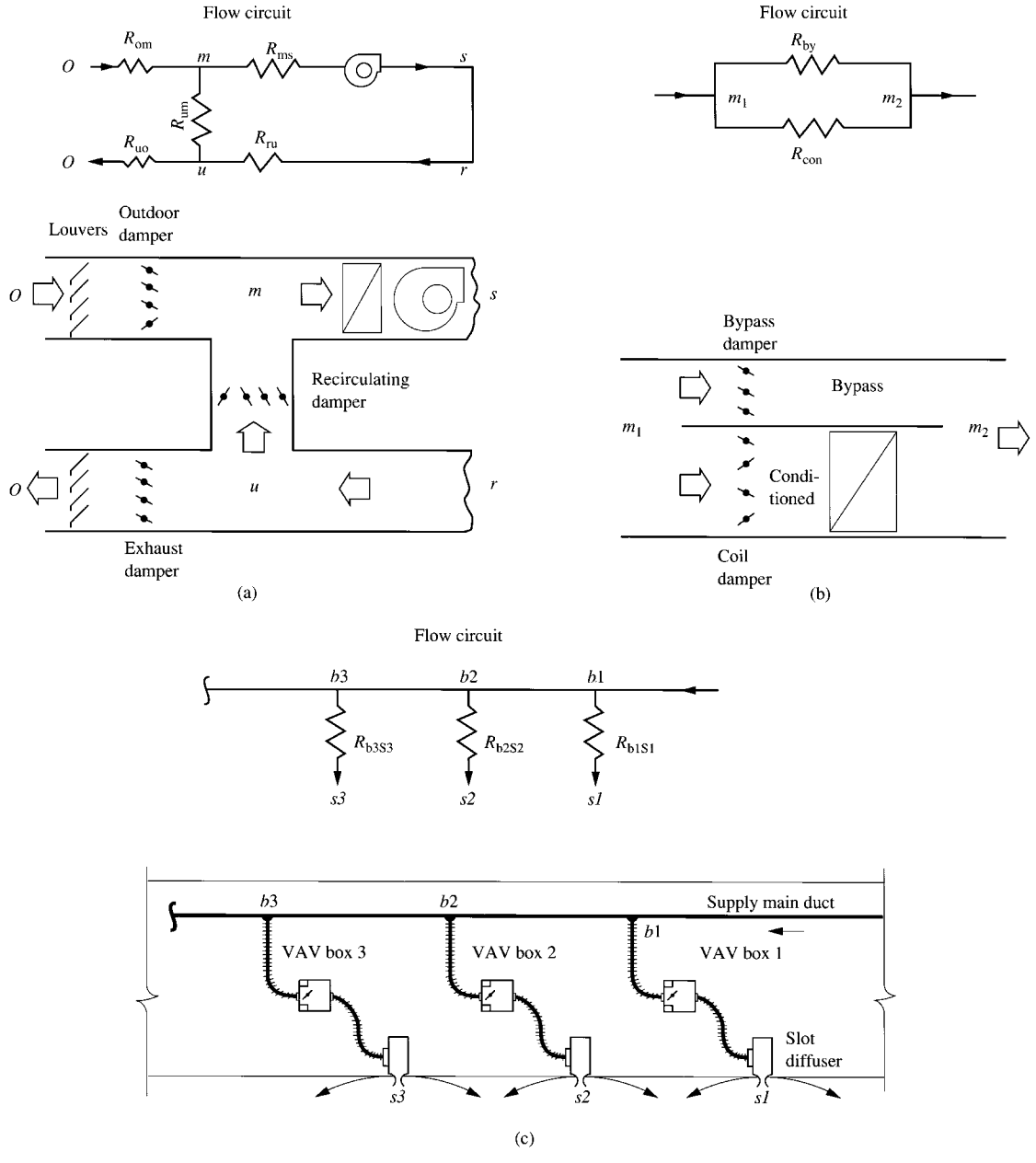


FIGURE 5.19 Airflow paths: (*a*) mixed- air control, (*b*) bypass control, and (*c*) branch flow control.

damper is then fully opened. If the outdoor damper is fully opened for free cooling, the recirculating damper is closed.

Bypass Control. In the flow circuit for bypass control, as shown in Fig. 5.19*b*, the entering air at the common junction m_1 is divided into two parallel airflow paths: the bypass path, in which a bypass damper is installed, and the conditioned path, in which the coil face damper is installed in series with the coil, or the washer damper with the air washer. The bypass and the conditioned airstreams are then mixed together at the common junction m_2. The face and bypass dampers can also be arranged in a certain relationship to each other.

Branch Flow Control. In a supply main duct that has many branch take offs, as shown in Fig. 5.19*c*, there are many parallel airflow path combinations: paths *b*1*s*1 and *b*1*b*2*s*2, *b*2*s*2 and *b*2*b*3*s*3, etc. In each branch flow path, there is a damper in the VAV box, and points *s*1, *s*2, *s*3, etc., are the status points of the supply air.

Parallel airflow paths such as those shown in Fig. 5.19 have the following characteristics:

1. The total pressure losses of the two airflow paths that connect the same endpoints are always equal; for example, $\Delta p_{um} = \Delta p_{uom}$, $\Delta p_{m1\ bym2} = \Delta p_{m1\ conm2}$, etc.
2. The relationship between total pressure loss Δp, in. WC (Pa); flow resistance R, in. WC/(cfm)^2 (Pa·s^2/m^6); and volume flow rate $\dot{V}$, cfm (m^3/s), can be expressed as

$$\Delta p = R\dot{V}^2 \tag{5.9}$$

   Flow resistance is covered in greater detail in Chap. 10.
3. If the total pressure loss Δp remains constant and the flow resistance R_n of one parallel path increases, from Eq. (5.9), the airflow through this path V must be reduced. The airflow in other parallel paths remains the same.
4. The total pressure loss of an airflow path between two common junctions Δp determines the volume flow rate of air passing through that path and can be calculated from Eq. (5.9) as

$$\dot{V} = \sqrt{\frac{\Delta p}{R}}$$

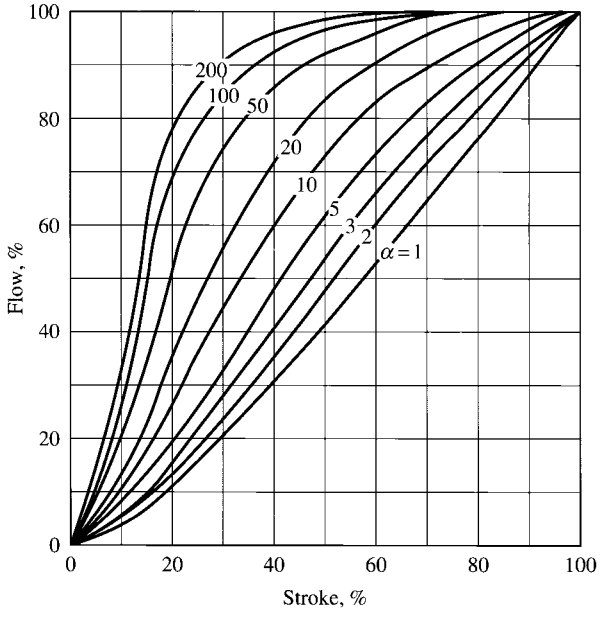
5. When the flow resistances in most of the branches increase because of the closing of the dampers to a small opening in their VAV boxes, the flow resistance of the supply duct system R_{sys} and the system total pressure loss Δp_{sys} both tend to increase, and thus the total air volume flow of the supply duct system $\dot{V}_{\text{sys}}$ will reduce accordingly.

Flow Characteristics of Opposed- and Parallel-Blade Dampers

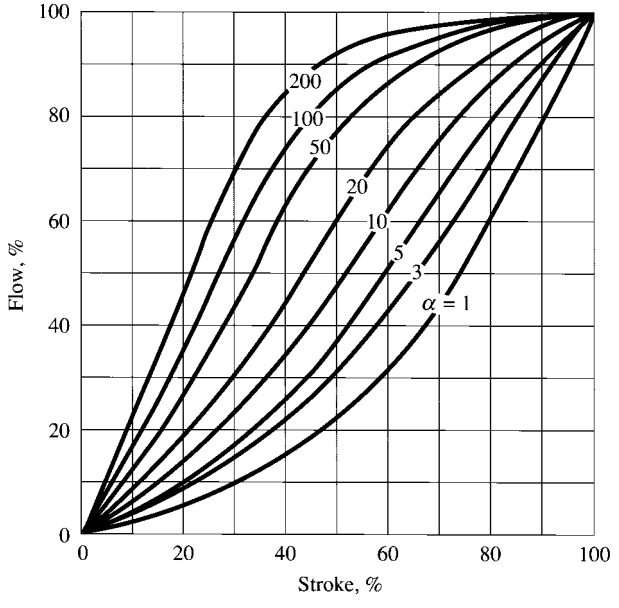
A parallel-blade or an opposed-blade damper that is installed in a single airflow path to modulate airflow is often called a volume control damper (or throttling damper). For volume control dampers, a linear relationship between the percentage of the damper opening and the percentage of full flow is desirable for better controllability and cost effectiveness. (Full flow is the air volume flow rate when the damper is fully opened at design conditions.) The actual relationship is given by the installed characteristic curves of parallel-blade and opposed-blade dampers shown in Fig. 5.20*a* and *b*. For the sake of energy savings, it is also preferable to have a lower pressure drop when air flows through the damper at the fully open condition.

In Fig. 5.20, α is called the *damper characteristic ratio* and may be calculated as

$$\alpha = \frac{\Delta p_{\text{path}} - \Delta p_{od}}{\Delta p_{od}} = \frac{\Delta p_{\text{path}}}{\Delta p_{od}} - 1 = \frac{\Delta p_{p\text{-}od}}{\Delta p_{od}} \tag{5.10}$$



(a)



(b)

FIGURE 5.20 Flow characteristic curves for dampers: (*a*) parallel-blade and (*b*) opposed-blade.

where Δp_{path} = total pressure loss of airflow path, in. WC (Pa)
Δp_{od} = total pressure loss of the damper when it is fully opened, in. WC (Pa)
$\Delta p_{p\text{-}od}$ = total pressure loss of air flow path excluding damper, in. WC (Pa)

Damper Selection

Butterfly dampers are usually used in ducts of small cross-sectional area or in places like VAV boxes.

For volume control dampers in a single airflow path, in order to have better controllability, an opposed-blade damper is recommended if many dynamic losses other than the damper itself (such as coil or air washer, heat exchanger, and louvers) exist in the airflow path. If the damper is the primary source of pressure drop in the airflow path, a parallel-blade damper is often used.

For mixing dampers, a parallel-blade damper is recommended for the recirculating damper as the pressure drop across the damper is often the primary source in its airflow path. An opposed-blade damper is recommended for the outdoor damper and exhaust (relief) damper for better controllability. The parallel blades of the recirculating damper should be arranged so that the recirculating airstream will blow toward the outdoor airstream, resulting in a more thorough mixing. Many packaged units also use parallel-blade outdoor dampers for smaller pressure drop and less energy consumption.

For face and bypass dampers, an opposed-blade coil face damper in an airflow path of greater pressure drop and a parallel-blade bypass damper will give better linear system control characteristics.

For two-position control dampers, a parallel-blade damper is always used because of its lower price.

Damper Sizing

Damper sizing should be chosen to provide better controllability (such as a linear relationship between damper opening and airflow), to avoid airflow noise if the damper is located in the ceiling plenum, and to achieve an optimum pressure drop at design flow to save energy.

The face area of the damper A_{dam}, ft^2 (m^2), in most cases is smaller than the duct area A_d, in ft^2 (m^2). Based on Alley (1988) paper, the local loss coefficient C_{dam} of the damper for different setups can be determined from Fig. 5.21. Then the pressure drop across the damper when the damper is fully opened Δp_{od}, in. WC (Pa), can be calculated as

$$\Delta p_{od} = C_{\text{dam}} \left(\frac{v_{\text{dam}}}{4005} \right)^2 \tag{5.11}$$

$$v_{\text{dam}} = \frac{\dot{V}_{\text{dam}}}{A_{\text{dam}}} \tag{5.12}$$

where v_{dam} = face velocity of the damper, fpm.

1. The damper is generally sized when the air flowing through the damper is at a maximum. For an outdoor damper, the maximum airflow usually exists when the free cooling air economizer cycle is used. For a recirculating damper, its maximum airflow occurs when the outdoor air damper is at minimum opening position, to provide outdoor air ventilation.

2. The face velocity of dampers v_{dam} is usually 1000 to 3000 fpm (5 to 15 m/s), except that the face velocity of a butterfly damper in a VAV box may drop to only 500 fpm (2.5 m/s) for energy savings and to avoid airflow noise. The ratio A_{dam}/A_d is often between 0.5 and 0.9.

3. The outdoor damper may be either made in a one-piece damper or split into two dampers, a larger and a smaller, to match the needs at free cooling and minimum outdoor ventilation.

4. For a bypass damper, its face area should be far smaller than that of an air washer or than a water heating or cooling coil's face damper. When the air washer or coil's face damper is closed, the area of the bypass damper should provide an airflow that does not exceed the system design airflow.

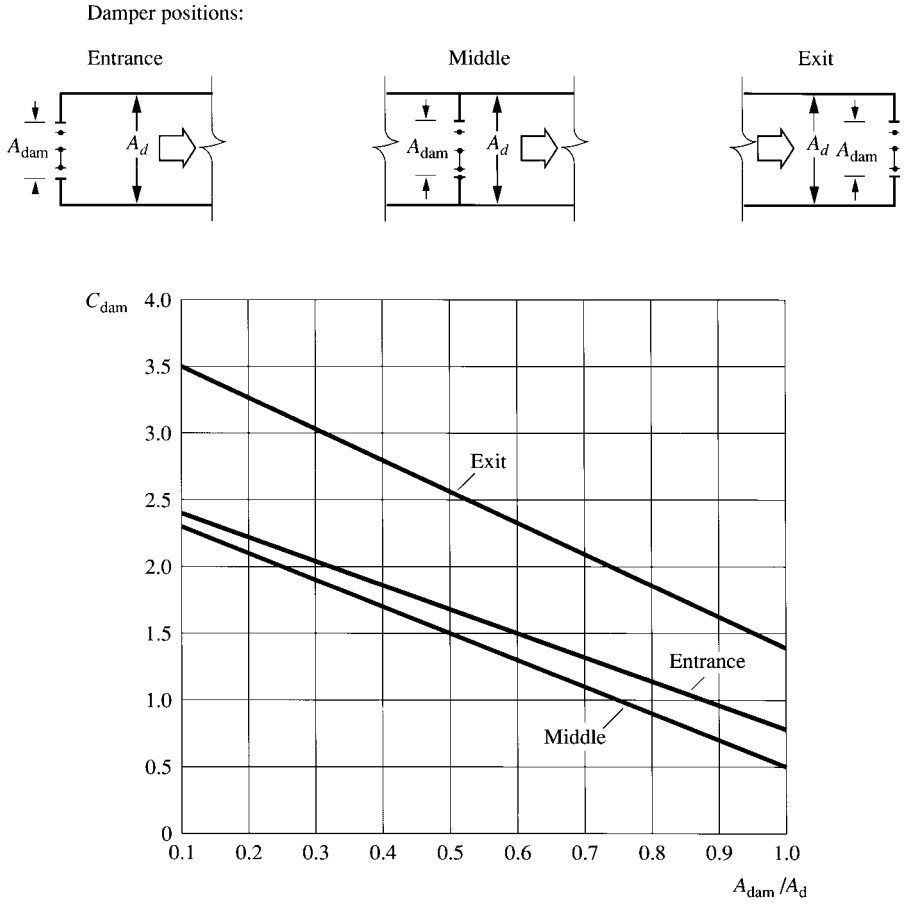


FIGURE 5.21 Local loss coefficient C_{dam} of air damper. (*Source: ASHRAE Transactions 1988, Part I. Reprinted by permission.*)

5.8 SYSTEM ARCHITECTURE

Architecture of a Typical EMCS with DDC

Figure 5.22 shows the system architecture of a typical energy management and control system with direct digital control (EMCS with DDC) for a medium or large building.

Operating Levels. Such an EMCS has mainly two operating levels:

1. *Unit level*. This level is controlled by unit controllers. A unit controller is a small and specialized direct digital controller which is used to control a specific piece of HVAC&R equipment or device such as a VAV box, a fan-coil unit, a water-source heat pump, an air-handling unit, a packaged unit, a chiller, or a boiler. For HVAC&R, most of the control operations are performed at the unit level. Since the software is often factory-loaded, only the time schedules, set points, and tuning constants can be changed by the user. Some of the most recently developed unit controllers are also

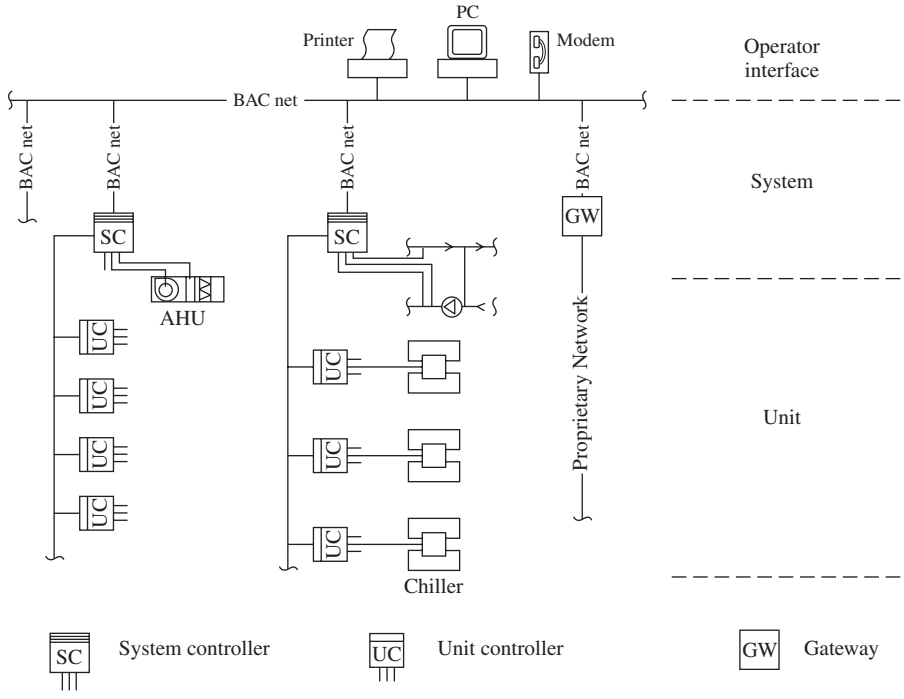


FIGURE 5.22 System architecture of a typical large EMCS.

programmable to various degrees. Sometimes the manufacturer provides a variety of preprogrammed control sequences, such as monitoring, and diagnostics, and designers can specify the required control sequence that best fits their designs.

2. *System/building level*. This level is controlled by system controllers. Since a system controller has an onboard capacity, programmed by an operator or factory-preprogrammed software, to execute complicated HVAC&R and other programs, they are the brain of an EMCS. Generic control software such as scheduling, trending, alarming, diagnostics, and security is also provided in system controllers. Generic control is covered in detail in a later section.

A system controller is used to coordinate the control operations of an HVAC&R system, such as the coordination between the duct static pressure and the total air volume of VAV boxes in an air system, or the sequencing of three centrifugal chillers and cooling towers in a refrigeration plant. A system controller may interface with sensors/transmitters by means of input/output (I/O) connections directly. Unit controllers are also configured on a separate subnetwork and connected to a system controller.

Operator Personal Computer (PC) Workstation. The operator may interface with the EMCS primary through an operator's PC workstation or purpose-built device, either handheld or fixed to the system controller. Each workstation shall consist of a PC, autodial telephone modems, and printers. The central processing unit (CPU) shall be a minimum of an Intel 80486 and operated at a minimum of 33 MHz. Its memory includes a minimum of 8 Mbytes RAM and 212 Mbytes hard disk. The communication ports connected to system controllers and other control systems should be provided. A 14-in. (356-mm) color monitor also shall be provided.

The software in the workstation shall do the following:

- Accommodate processes as well as prioritize applications based on their input/output priority level.
- Provide system graphics including the HVAC&R equipment (such as a display of up to 10 graphic screens at once for comparison) to monitor the operating status of the system. The operator with the proper password is able to add, delete, or change the set points, time scheduling, etc.
- Support the editing of all system applications including the generic control software provided in the system controllers. The edited or custom programmed software shall be downloaded and executed at one or more of the system controllers.
- Automatically save the database and restore the database that has been lost in one of the system controllers.
- Provide scheduling, trends, totalization, alarm processing, security to view and edit data, and system diagnostics.

Communication Network. A peer-to-peer data communication of a local-area network (LAN) adopting either Ethernet or ARCNET will be used between system controllers and between system controllers and PC workstations (or other system). A peer-to-peer communication means that all system controllers or work stations have equitable access to communication resources. For the communication subnetwork between system controllers and unit controllers, and between unit controllers themselves, a master-slave token-passing (MS/TP) technology is often used. A system controller also acts as a medium to provide data communication between the work station and the unit controllers. The network technology is covered in detail in a later section.

Power Source. The temperature sensors and humidity sensors may need up to 12 V dc and 24 V ac as a power source. Many DDC units have a power source of 24 V ac or 120 V ac line voltage. Most of the valve actuators and damper motors need a power source of 24 V ac.

Size of EMCS. The size of an EMCS depends on the number of points (or objects) that belong to its DDC units. An EMCS of 100 points or less can be considered a small project. An EMCS that has 1000 points or more can be considered a large project.

System Characteristics

An architecture of EMCS incorporating DDC such as that in Fig. 5.22 has the following characteristics:

1. All DDC units are independent and stand-alone controllers. If any of the controller fails, there is only a limited effect.

2. The system architecture shows a distributed processing model. Since most of the control operations are performed at the unit controller level, and partly at the system controller level, such an architecture has the advantage that it tremendously reduces the data communication between the unit controllers and the system controller, as well as between the DDC system controller and the PC workstation.

3. Zimmerman (1996) noted that since the introduction of DDC in the 1980s, ". . . microprocessor and memory have declined rapidly in cost while wiring and installation costs have not declined at the same rate." To provide more powerful unit controllers, moving the controllers nearer to the sensors and control devices will reduce a lot of wiring and installation costs as well as the overall system cost.

4. If each HVAC&R piece of equipment has its own unit controller, it surely will be beneficial to the HVAC&R equipment manufacturers to fabricate controllers and other control system components themselves.

Future Development

The development of more powerful, programmable unit controllers using a modular configuration, will reduce the difference between a system controller and a unit controller. As predicted by Hartman (1993), the future architecture of an EMCS may have only a single tier of various kinds of unit controllers. The operator's PC workstation, modems, and unit controllers will all be connected to a peer-to-peer data communication trunk. Such an architecture will simplify the DDC units and the communication network, move the intelligence nearer to the control devices, and finally create more effective control at lower system cost.

5.9 INTEROPERABILITY AND OPEN PROTOCOL BACnet

Interoperability

Turpin (1999) defined *interoperability* as "the ability of systems including equipment and components from different manufacturers to share data and information for the purpose of operation with plug-and-play connectivity." Interoperability is one of the necessary conditions for system integration. System integration is a strategy to integrate various HVAC&R systems of various manufacturers together, and to integrate HVAC&R systems with lighting, fire protection, security, elevator, and electrical systems in a building together.

Robertson and Moult (1999) note that the advantages of system integration include the following:

- It reduces the installation cost of the building automation system.
- It enhances energy management.
- It can apply building automation system features.
- It provides building operator training.
- There is a single user interface.

The greater the interoperability of a single system and the more systems you try to integrate into a single system the more complicated and costly the process will be. Interoperability and system integration is one of the goals of the HVAC&R industry. More and more engineers, facility owners, and manufacturers recognize this need. It takes time to accomplish such a complicated process.

BACnet—Open Data Communication Protocol

BACnet means *b*uilding *a*utomation and *c*ontrol *net*work. It is an open data communication protocol. *Open* means that an independent institution governs its development. All contents are known, fixed, and accessible. A *protocol* refers to the rules by which two or more devices communicate data to each other that must be obeyed. BACnet enables that building automation devices from various manufacturers can talk to one another, share data, and work together following a standard way. BACnet defines all the elements of data communication between devices in a building automation control system. It is specifically tailored for HVAC&R control equipment, but it also provides a basis for integrating lighting, security, and fire-detecting systems. BACnet was developed from 1987 to 1995 by ASHRAE and was adopted as a national standard in 1995 by ANSI, as ANSI/ASHRAE Standard 135-1995. For details, refer to ASHRAE's BACnet.

BACnet will assist building owners, designers, contractors, and operators in three areas:

1. It gives more freedom to select the best equipment and components from different manufacturers in order to have a more efficient system at lower cost.
2. It operates the control system from a single workstation; i.e., it is more effective to operate and easier to maintain, and there is only one system to learn.

3. It collects data from different systems and offers greater flexibility for extended systems in retrofit projects.

Most of the EMCS manufacturers agreed to fabricate BACnet-compliant products from the late 1990s.

There are many other data communication protocols. One was developed by Echelon Corporation, called LonTalk Protocol, and it became a working system in the mid-1990s and was favored by members of the LonMark Interoperability Association. LonTalk protocol is neither an open-to-public protocol nor a standard.

An independent consortium called the OPC foundation, formed as a nonprofit organization in 1996 in Boca Raton, Florida, has dedicated itself to provide interoperability with Microsoft technologies to develop a global specification and multivendor interoperability in industries. In 1990, OPC is leading 140 member companies including Honeywell , Johnson Controls, and Siemens.

Application Layer

Layered Structure. A data communication system often adopts a hierarchical layered structure so that a complex problem is broken into smaller and more easily solved problems. BACnet is based on a four-layer collapsed architecture that corresponds to application, network, data link, and physical layers in an International Organization for Standardization (ISO) model. This is the result of careful consideration of the characteristics and requirements of the building automation control (including HVAC&R) together with a constraint that protocol overhead be as low as possible.

An application layer is the highest layer in BACnet. It serves to define the objects and services (including control operations, information exchange, and control devices) in a building automation control (BAC) system. It also provides communication services and data encoding schemes required by applications to perform monitoring and control functions.

Object Types, Properties, and Devices. The BACnet defines a set of *standard object types* instead of conventional points. Analog input, analog output, binary value, command, file, program, schedule, etc., grouped in 18 types are standard object types, for every device in an EMCS must have a device object. There are 123 *properties* that have been identified by BACnet which fully describe the device, or object type, in the network. Certain properties are required to be specified whereas others are optional. An object identifier specifies its object name, object type, etc., and optional properties such as description and device type. In BACnet, a *device* is defined as any device, real or virtual, that supports digital communication using the BACnet protocol.

Services. In BACnet, *services* are the operations by which one device acquires information from another device, commands another device to do something, or announces that some event happened. BACnet defines 32 services that can be grouped into five categories:

- Alarm and event services refer to changes in conditions detected by a device, such as acknowledgment of an alarm and confirmed change of value notification.
- File access services are used to read and manipulate files that are kept in devices, such as only one read or write operation at a time.
- Object access service provides the means to read, to write, and to modify properties, such as to add one or more items to a property.
- Remote device management offers disparate operations such as to tell a device to stop accepting messages.
- Virtual terminal services are used by an operator to establish a bidirectional connection with an application program implemented in a remote device.

Services are classified as *confirmed* when a reply is usually expected with data and *unconfirmed* when no reply is expected. In BACnet, a given device is not required to implement every service. However, “read property” is required to be executed by all the devices.

Conformance Class, Functional Groups, and Protocol Implementation Conformance Statement (PICS). BACnet defines six levels of conformance for all devices, classes 1 to 6, that are hierarchical to indicate the difference in requirements that must be met to conform to BACnet. The requirement of a class includes all the requirements of all the other classes having a lower number. At the lowest level, conformance class 1 requires only that a device be able to execute (respond to) a "read property service" request, such as a sensor. For conformance class 6, a device is required to implement 21 types of service requests, such as a PC workstation.

A functional group defines a combination of services and object types that are required to perform certain BAC functions. In BACnet, there are altogether 13 functional groups, such as clock, workstation, and event initiation.

The protocol implementation conformance statement (PICS) is a document provided by the manufacturer of a device to identify those options implemented by a particular device. The PICS covers the conformance class of the device, supported functional groups, standard and proprietary services executed and initiated, etc.

Data Encoding. In BACnet, application-layer protocol data units (APDUs) are used to convey the information contained in the service primitives and associated parameters. ISO Standard 8824, *Specification of Abstract Syntax Notation One (ASN.1)*, has been chosen as the method to represent the data content of BACnet services. Each data element consists of three components: (1) identifier octets, (2) length octets, and (3) content octets. The fixed portion of each APDU providing protocol control information is encoded implicitly, and the variable portion of each APDU providing service-specific information is encoded explicitly.

Network Layer

In BACnet, the purpose of the network layer is to provide the means from which messages can be relayed from one BACnet network to another in the internetwork. Two or more BACnet networks are connected by routers to form a BACnet internetwork. A router, a BACnet device, is used to interconnect two disparate BACnet local-area networks (LANs). A network layer directs the messages to a single remote device or broadcasts the messages on a remote network, or to all devices on all networks. A device is located by a network number and a medium access control (MAC) address. Another network layer function is message segmentation and reassembly.

In an EMCS, there are often two networks: one uses Ethernet or ARCNET for high-speed message transmission between system controllers and the PC workstation, and another adopts low-speed message transmission between unit controllers and system controllers. Consequently, a network layer is required for the BACnet protocol.

Data Link/Physical Layer—Network Technology

In BACnet, a data link layer has the capability to address messages to a single device or to all devices. At the data link layer, only incoming BACnet messages received from the physical layer are passed on to the network layer, and a code is added to the outgoing messages to identify them as BACnet messages before they are passed to the physical layer. A physical layer provides the physical medium for message transmission.

For local-area networks, BACnet supports Ethernet, ARCNET, MS/TP, PTP, and LonTalk as alternatives. Various media are used as physical transmission entities. Typical media are twisted-pair wire, fiber-optic cable, and coaxial cable.

Ethernet. It is one of the commonly used LAN technologies. At any time, multiple devices may access the network simultaneously, i.e., multiple access with collision detection. Ethernet is using a peer-to-peer communication with a bus network configuration. Ethernet often runs at a speed of 10 Mbits/s. Two types of coaxial cable are often used in Ethernet: thick-wire and thin-wire. A thick-wire Ethernet segment has a maximum length of 1600 ft (488 m), and up to 100 nodes can be

attached. Thick-wire Ethernet is more expensive. A thin-wire Ethernet segment has a maximum length of 600 ft (183 m) and 50 attached nodes.

ARCNET (Attached Resources Computer Network). This is also a commonly used LAN technology and is lower in cost than Ethernet. ARCNET adopts a token-passing network access method. A token which indicates the permission to use the physical medium is passed from one network node to the next node in a circular manner. As in Ethernet, ARCNET also uses a peer-to-peer communication network, and its network nodes reside on a bus.

Both coaxial cable and twisted-pair wire are used in ARCNET and run at a speed of 2.5 Mbits/s. For each ARCNET segment, up to 8 nodes can be connected. All together, up to 255 nodes can be communicated over an ARCNET network.

Master-Slave/Token-Passing (MS/TP). In BACnet, MS/TP divides all the nodes on the network into two categories: masters and slaves. Only masters can initiate data communication, whereas slaves cannot initiate. Slaves can only respond to requests from masters. The MS/TP also adopts a token-passing network access method. A master node may access the network only when the token (permission to use the medium) is passed to it from the previous master node. The token never passes to the slave nodes. Master nodes in an MS/TP network are at a peer-to-peer communication. The MS/TP network often uses twisted-pair wires. It can operate at a speed of 9600 bits/s, as well as 19.2, 38.4, and 76.8 kbits/s.

Point-to-Point (PTP). This is a data link layer protocol which provides serial communication between two devices. Such a point-to-point communication typically involves a dial-up phone modem or hardwired connection between two nodes. PTP has a simpler medium access mechanism and is often temporary in nature. PTP is much slower than a LAN. Both devices can receive and transmit simultaneously.

LonTalk LAN. This is an option of the physical medium in BACnet and is at the base of the Lon Mark protocol. LonTalk supports a number of choices of physical media.

Connection between BACnet and Proprietary Network

For an extension project, it is possible that the original building is still intended to keep the proprietary network and the extended part is to construct a BACnet network. The proprietary network needs a gateway to connect to the BACnet network, as shown in Fig. 5.22. A proprietary network does not open to others. Typically, information can only be exchanged between EMCS components and the proprietary network of the same manufacturer. For two networks and their computers using different protocols to communicate, some translation must take place. The device that provides this translation is called a *gateway*.

LonTalk Protocol

According to Glinke (1997), the Local Operators Network (LonTalk) protocol is a seven-layer proprietary protocol developed by Echelon Corp. Data communication is implemented on a neuron chip. There are actually three microprocessors within the chip; two are used for the network and one is for specific functions. These devices, which allow different types of existing networks to communicate, use multiple, low-cost media (Ethernet, ARCNET, fiber optics) and provide flexibility. Since this technology is essentially peer-to-peer communication without the need for a central supervisory control node, Lon Mark-certified unit controllers can exist on the same network as equals.

The two promising and widely used protocols, BACnet and LonTalk, were able to communicate with each other in the late 1990s. They will improve themselves through actual operation in the future and produce better choices for engineers and users.

5.10 CONTROL LOGIC AND ARTIFICIAL INTELLIGENCE

In an EMCS, the software in the DDC units determines the control functionality. Since the logic is separate from the hardware, the control functions and sequences are now limited only by the knowledge and innovation of the designer and operators. The development of artificial intelligence in the DDC software—fuzzy logic, expert systems, artificial neural network, etc.—tremendously expands the control flexibility and improves the operating quality to meet the more complex requirements for an HVAC&R energy management and control system.

Fuzzy Logic

Basics. Since many systems have become more and more complex over the past decades, an accurate analytical model based on rigorous and nonlinear mathematics is difficult to develop and to be accepted in the daily management by operators. In 1965 Professor Lotfi Zadeh of the University of California, Berkeley, developed fuzzy set theory which provides a new control logic for a system.

Fuzzy logic has the ability to deal with the imprecision that happens in everyday life. Conventional digital technology is based on bivalence: yes or no, on or off, 1 or 0, and black or white. Fuzzy logic is multivalent. Things can be partly yes to some degree, partly no to some degree, and fuzzy logic deals with shades of gray. Conventional set theory observes that a fact must be either true or false, whereas in fuzzy logic set theory, a fact can be partly true and partly false, can belong to a set and also belong to another set.

According to Lehr (1996) and Scholten (1995), fuzzy logic offers simplicity in the midst of complexity and is a real alternative for system operation and control. The obvious benefits of fuzzy logic are that it makes things more human and more friendly to a person who is less trained, and it is more easily maintainable. Fuzzy logic really is not a vague theory. It is comprised of a set of precise rules based on rigorous mathematics which governs the behavior of a system by means of words and phrases instead of nonlinear models. Fuzzy logic controllers (FLCs) have been widely used in air conditioners, humidifiers, refrigerators, and many other devices such as elevators. There are also many applications for which conventional control logic is better than fuzzy logic controller.

Fuzzy Sets and Membership Function. Fuzzy sets, membership function, and production rules are three primary elements of fuzzy logic. Figure 5.23 shows the fuzzy sets, membership functions, and a diagram of fuzzy logic control. In conventional bivalent crisp set theory, "sets" of thermal comfort of an occupant in an air conditioned space can be categorized according to the indoor temperature T_r exactly as *cool*, a range of 62 to 72°F (16.7 to 22.2°C); and *just right*, a range of 72 to 76°F (22.2 to 24.6°C), etc., as shown in Fig. 5.23. At T_r = 74°F (23.3°C) and T_r = 75°F (23.9°C), both are "just right"—have a membership value of 1 with regard to the just-right set and a membership value of 0 with regard to the cool set.

On the other hand, the fuzzy set "just right" ranges between a membership value of 0 at 70°F (21.1°C), a membership value of 1 at 74°F (23.3°C), and a membership value of 0 again at 78°F (25.6°C). For T_r = 75°F (23.9°C), the membership value of fuzzy sets would be described as 75 percent just right and 25 percent warm. In another way, the assertion of thermal comfort of just right is 75 percent true and 25 percent false.

If T_r = 76°F (24.4°C), then the conventional crisp set will be difficult to determine. In addition, for T_r = 74°F (23.3°C) and T_r = 75°F (23.9°C), the assertion of the conventional crisp set for both is "just right," so there is no difference between them. And the assertion of fuzzy sets for T_r = 74°F (23.3°C) is 100 percent just right, which is different from 75 percent just right and 25 percent warm when T_r = 75°F (23.9°C).

Production Rules. When the fuzzy set of an occupant's thermal comfort is integrated with another fuzzy set of percentage of fan speed variation (positive large, positive small, zero, negative small,

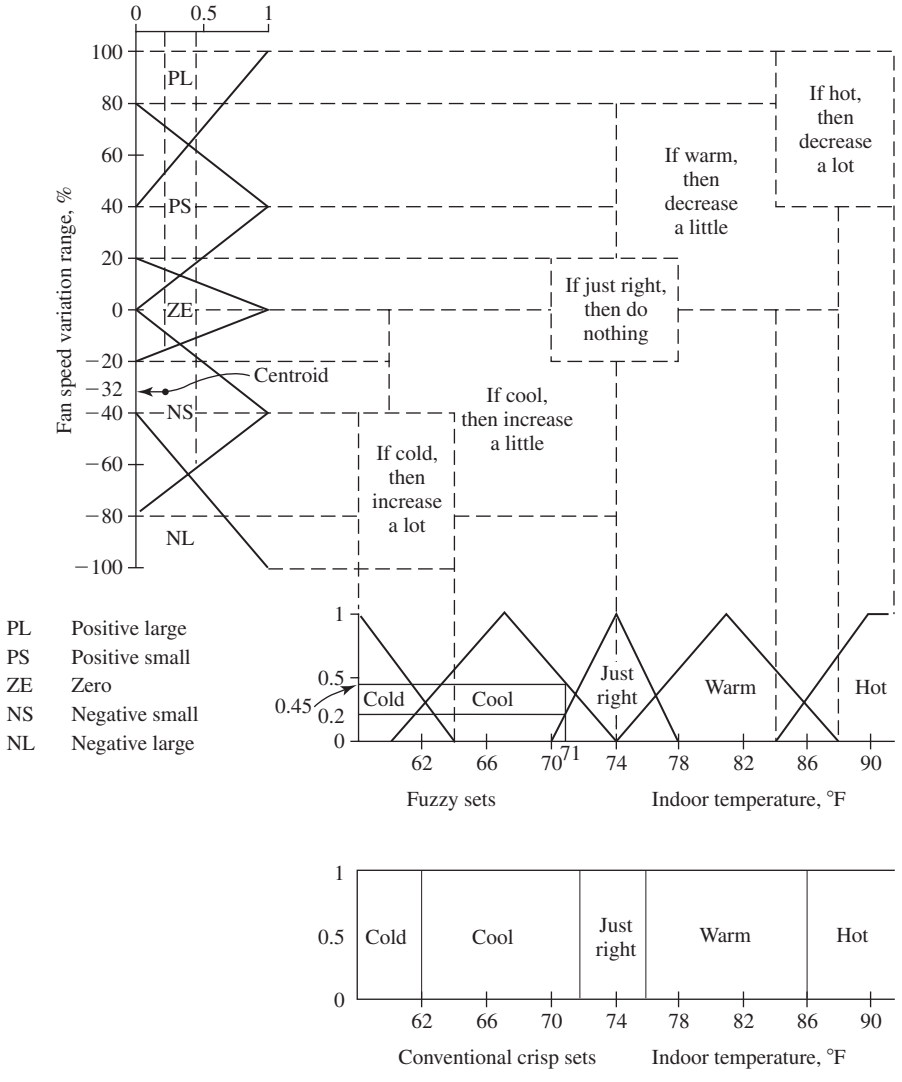


FIGURE 5.23 Fuzzy sets, membership function, and fuzzy logic control.

and negative large), then the production rules, or fuzzy logic rules, can be described as follows:

- If the indoor temperature is hot, then the fan speed rises a lot.
- If the indoor temperature is warm, then the fan speed rises a little.
- If the indoor temperature is just right, then the fan speed stays unchanged.
- If the indoor temperature is cool, then the fan speed reduces a little.
- If the indoor temperature is cold , then fan speed reduces a lot.

Fuzzy Logic Controller. An FLC consists of three parts: a fuzzifier converts ordinary inputs to fuzzy variables, a fuzzy reasoning unit produces fuzzy control signals based on input fuzzy variables, and a defuzzifier converts fuzzy control signals to conventional control outputs.

If the indoor temperature T_r is 71°F (21.7°C), then the membership function of the fuzzy sets of thermal comfort in Fig. 5.23 would be described as 0.2 (20 percent) just right and 0.45 (45 percent) cool. From the production rules, then, "reduce the fan speed a little" has a membership value of 0.75 and "fan speed remains unchanged" has a membership value of 0.2. One way to interpret these two fuzzy outputs in a conventional crisp output is to determine the centroid based on the area of the two truncated triangles. From Fig. 5.23, the output value is 32 (reduce the fan speed by 32 percent of its range).

Huang and Nelson (1994) noted that an FLC is characterized by a set of linguistic fuzzy logic rules. The initial set of these rules is often based on past experience or analysis of the control process. The initial set of rules can be modified by analyzing the performance trajectory on the linguistic plane to obtain an optimal rule set. This is called *rule refinement*. The second significant influence on the behavior of an FLC is the choice of membership functions. The overlap of the fuzzy sets should be moderate to allow for reasoning with uncertainty and the need for completeness of the control rules. Refer to Huang and Nelson's paper for details.

Knowledge-Based Systems and Expert Systems

Basics. A knowledge-based system (KBS) is a knowledge-rich, logic-oriented computer program that mimics human knowledge and reasoning in a specific domain to assist in solving more complex problems. Human knowledge includes expert knowledge, common facts, and knowledge in any form, whereas an expert system, strictly speaking, mimics only an expert's expertise in a given domain to solve specific problems. An expert system can be considered as the core part of a knowledge-based system.

In comparing a knowledge-based system with conventional programmed software, Hall and Deringer (1989) noted that the benefits of a knowledge-based system are due to the following superior abilities:

- *Logical reasoning*. Logical problems are solved by means of sorting, comparing, searching, and reasoning as well as to evaluate alternative choices to a specific problem. More often, several causes (or solutions) of a problem are identified.
- *Resolution of uncertainty*. When precise knowledge is not available and for problems that involve qualitative facts and incomplete data, the certainty of the conclusions of the KBS using heuristic rules depends on the reliability of the heuristic rules themselves.
- *Multiple approaches*. Multiple experts are often used to develop a knowledge base. Experts will rarely agree exactly on the proper approach to solve a problem. Solution of the problem with multiple approaches is one aspect of KBS.
- *Justification of results*. A well-developed KBS has the capability to recall the basis of each decision during the problem-solving process. It can provide a decision trail that will explain the sequence of decisions used to produce the conclusion.

The limitations of a KBS are as follows. First, a KBS can make no decision that is not explicitly contained in the knowledge base. Second, two different KBSs developed using the expertise from different experts may give different advice for the same problem. Third, the KBS is time-consuming and expensive.

Knowledge-based systems will not replace experts and engineering professionals. Instead, the systems will play the role of specialized knowledge-based advisers or consultants.

KBS Structure. An expert system comprises the following four modules, as shown in Fig. 5.24.

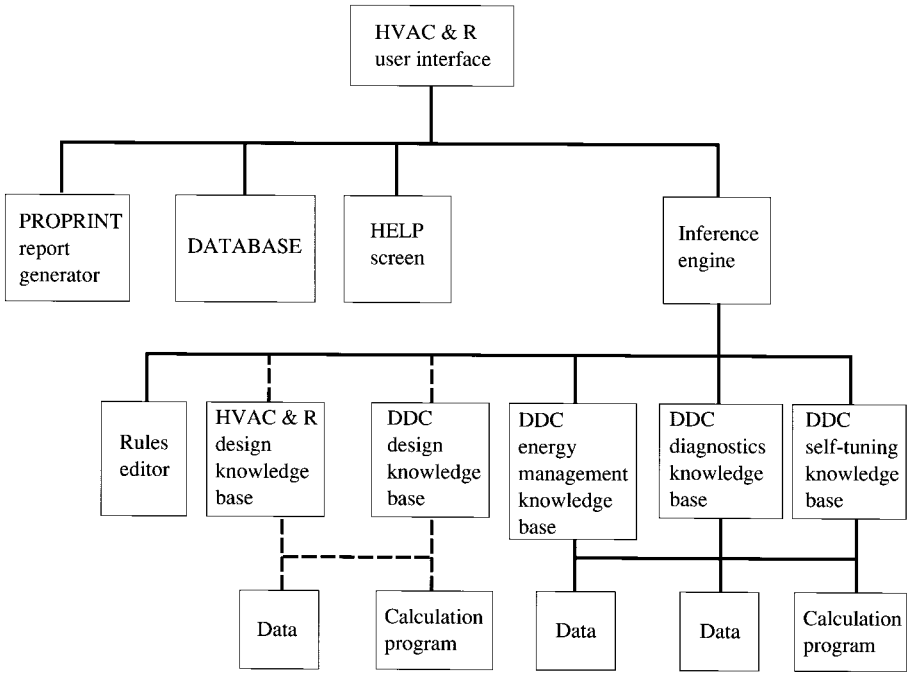


FIGURE 5.24 Structure of a knowledge-based system for an EMCS and HVAC&R design.

1. *Knowledge base.* This is the most important part of an expert system. It contains the common facts and inference rules, often in the form of if-then, and is domain-specific. Every rule in the knowledge base is extracted from the expert's knowledge and experience by the developer of the KBS, the knowledge engineer, or is based on data and information from published handbooks, manufacturers' engineer manuals, and field survey results. Frames are another common knowledge representation method. Hall and Deringer (1989) defined *frames* that ". . . facilitate the description of knowledge about complex objects, which have many subelements, . . ." The content of the knowledge base should be tailored to the user's task and requirements. The user's task can be determined through direct interview with potential end users.

2. *Inference engine*. The inference engine asks for inputs from the user interface, executes reasoning algorithms by applying the knowledge from the knowledge base, and arrives at the conclusion based on the rules. There are two kinds of reasoning: (1) data-driven forward chaining or forward reasoning and (2) goal-driven backward chaining or backward reasoning. They are categorized according to how new information is inferred. Forward reasoning proceeds forward from the user inputs about probable outcomes. In most engineering design problems, forward reasoning is adopted. For instance, knowledge of the building envelope leads to load calculations, which forwards next to determining the equipment capacity, and so on.

Backward reasoning proceeds backward from the user inputs about probable outcomes. In troubleshooting or diagnostics, backward reasoning is used. For an example, collected facts lead to determining the cause of the problem. Forward and backward reasoning were often used in conjunction to control the flow of questioning as well as provide a dialog between the system and the end user.

3. *User interface*. This includes the facilities to support smooth and convenient interaction with the user. Only simple instruction is needed to start the system, end a session, save the contents, and

give a printout. A friendly dialog between the KBS and the user and a simple question-and-answer format with a provided menu of possible answers are expected. An intelligent operating system which accesses various application tools is recommended. Modular new application tools can be plugged into the operating system as required. A user interface with built-in intelligence, graphically based high-resolution display with standardized menus and format may be the preferable answer.

4. *Knowledge acquisition*. A knowledge acquisition module provides strategies to capture the experts' knowledge to develop a KBS. Sometimes, it also checks for consistency and completeness. Knowledge acquisitions are usually accomplished through personal interviews with experts and review of the application literature. Most HVAC&R-related knowledge systems have long-term plans to continue the process of knowledge acquisition. A new trend is machine learning; i.e., the computer learns from the experience how to capture and manipulate new knowledge.

Development of KBSs. Many KBSs are developed by using commercially available development tools called *shells*. A shell consists of mainly a rule editor, a knowledge base, an inference engine, and user interfaces. Usually, the inference engine and user interface are fully developed within a shell. Thus, the user can focus on the collection and the input of the knowledge. He or she can change the existing knowledge base and does not need to change the entire system. KBS is expected to operate on a PC.

The performance of a KBS and the resources needed during development are highly affected by its knowledge engineer. The knowledge base is built through the cooperation of the knowledge engineer and experts in a specific domain. The knowledge engineer has the responsibility to choose an appropriate inference strategy, and a suitable shell and to ensure compliance of the system with the task.

The development cycle of a KBS is an iterative and incremental process. It begins with the initial prototype. The next is an improved, and expanded, one. The process is repeated and may take years.

How the KBS Works. For example, if the space air is too humid and the space cooling load is only one-half of the design load, it is required to find the general cause of these symptoms. The problem is a diagnosis problem. Brothers and Cooney (1989) stated that if-then rules are composed of parameters (symptom, set point) and values (too hot, too humid, correct). In the inference engine, the computer program of the KBS will begin to search for an if-then rule through the knowledge base that will give a value (too humid) for a general cause. If the values do not match those required by the rule, the computer program will search for the next rule, until the following if-then rule has been found:

IF symptom is too humid, and
  cold supply air temperature set point is correct, and
  cold supply air relative humidity is O.K., and
  sensible cooling load is 50 percent of design load
THEN general cause is size of supply airflow rate CF 90

The cause of a too humid space air is that the volume flow of supply air is too great. The term CF is the abbreviation of *certainty factor*. That means the confidence in the answer to the problem is 90 percent.

Testing, Verification, and Validation. In Jafar et al. (1991), testing often detects logical errors related to the knowledge base, syntactic errors, and missing knowledge. Testing only shows errors and does not explain their cause. Verification and validation should be performed at each stage of development, to check the knowledge base for internal inconsistencies, mismatches, etc.

Applications. There are three primary areas of HVAC&R-related applications of KBS, as reported in the paper by Hall and Deringer:

- Monitoring — interpretation of measured data in comparison to expected behavior

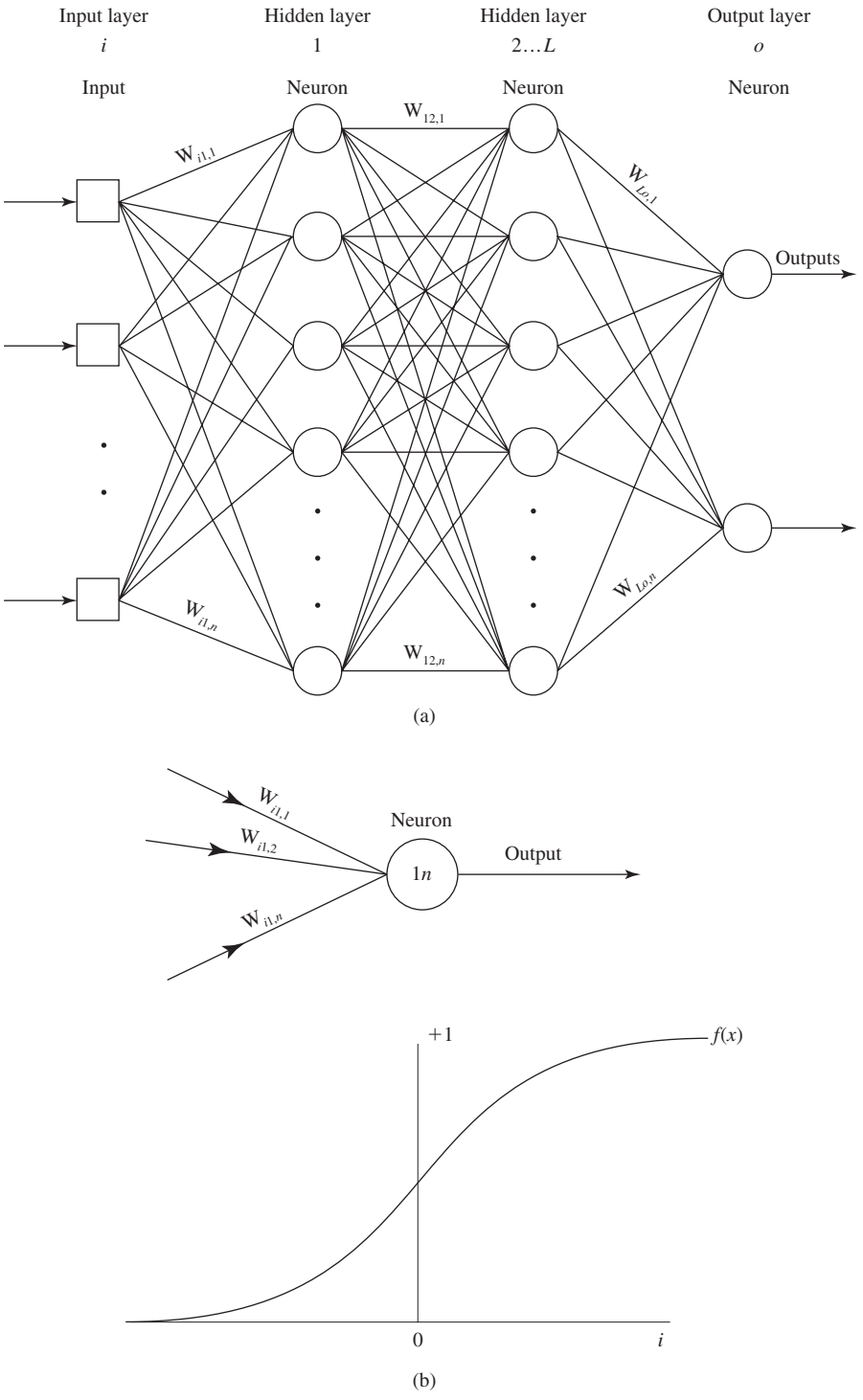


FIGURE 5.25 An artificial neural network (ANN): (*a*) Structure of an ANN; (*b*) sigmoid function.

- Diagnostics—assistance in identifying solutions to complex technical problems
- Design—assistance in the selection of the HVAC&R systems and subsystems

Artificial Neural Networks

Basics. An *artificial neural network* (ANN) is massive interconnected, parallel processing, dynamic system of interacting processing elements that are in some aspect similar to the human brain. The fundamental processing element is called the *neuron*, which is analogous to the neural cell in human brain. The neurons are set in layers, and thus a network is formed as shown in Fig. 5.25. Inputs representing the variables that affect the output of the network are feeding forward to each of the neurons in the following layers with an activation depending on their weighted sum. Finally, an output can be calculated as a function of the weighted sum of the inputs and an additional factor, the biases.

The ability to learn is one of the outstanding characteristics of an ANN. The weights of the inputs are adjusted to produce a predicted output within specified errors. ANNs have been increasingly used in recent years to predict or to improve nonlinear system performance in HVAC&R. An ANN system is characterized by its net topology, neuron activations transfer, and learning method.

Net Topology. The structure of the network of an ANN, or net topology, depends on the data flow mode, the number of layers, and the number of hidden neurons.

- In Miller and Seem (1991), there are two types of data flow modes: state and feed-forward models. In the state models, all the neurons are connected to all the other neurons. In feed-forward models, the neurons are connected between layers, as shown in Fig. 5.25*a*, and the information flows from one layer to the next. Feed-forward models are the most popular and most often analyzed models.
- In an ANN, there is always an input layer with the number of inputs equal to the number of parameters (variables) that affect the output.
- There may be one or more hidden layers of neurons next to the input layer. The selection of number of hidden layers and the number of neurons in each hidden layer remains an art. Curtiss et al. (1996) noted that too many hidden layers and hidden neurons tend to memorize data rather than learning. The hidden layers and hidden neurons must be sufficient to meet the requirement during the learning process for more complex nonlinear systems. More hidden layers and hidden units need more calculations and become a burden.

Among the 10 papers published from 1993 to 1996 in *ASHRAE Transactions* regarding developed ANNs, most have only one hidden layer, some have two layers, and only one has three hidden layers. None exceeds three hidden layers. Kawashima (1994) recommended that in an ANN only one hidden layer is sufficient for load prediction.

- If the relationship between the inputs and output is more complex, i.e., nonlinear, and more inputs are involved, then more neurons are needed in each hidden layer. Kawashima (1994) also suggested that the number of neurons in each hidden layer exceed $2m + 1$. Here m indicates the number of inputs.
- There is always an output layer next to the hidden layer(s). It is preferable to have one neuron (single output) in the output layer for simplicity. There may be two or more neurons for multiple outputs.

Neuron Activation Transfer. In Miller and Seem (1991) and Curtiss et al. (1996), for each neuron in the hidden and output layers:

1. The input activations to a neuron in the first hidden layer h, denoted by i_{1n}, can be

calculated as

$$i_{1n} = \sum i_n W_{i1,n} + B = S + B \tag{5.13}$$

where i_n = normalized input
$W_{i1,n}$ = weights of connection between inputs and neurons in first hidden layer
B = biases

$$i_n = \frac{i_o - i_{\min}}{i_{\max} - i_{\min}} \tag{5.14}$$

where i_o = original input data
$i_{\max}$, $i_{\min}$ = maximum and minimum of original input data

2. The output from the neurons in the first hidden layer o_{1n}, is expressed as a selected sigmoid activation function:

$$o_{1n} = \frac{1}{1 + e^{-(S-B)}} \tag{5.15}$$

There is only one output o for a neuron. This output is transmitted through output connections in which it usually splits into multiple connections with identical activations.

3. The input activation to a neuron in the hidden layer L, denoted i_{Ln}, or a neuron in the output layer o, denoted by i_{on}, is equal to the output of the neuron in the previous layer which split identically to all the neurons in hidden layer L. The output of a neuron in hidden layer L, denoted by o_{Ln}, or output layer o_{on} can be calculated as

$$\begin{aligned} o_{Ln} &= \frac{1}{1 + e^{-(T-B)}} \\ o_{on} &= \frac{1}{1 + e^{-(V-B)}} \end{aligned} \tag{5.16}$$

and

$$\begin{aligned} T &= \sum i_{1n} W_{1L,n} \\ V &= \sum i_{Ln} W_{Lo,n} \end{aligned} \tag{5.17}$$

Learning Method. The learning method, also called the training process, determines the connection weights W_{xyn} through known sets of input/output pairs. The initially assigned connection weights are adjusted repeatedly during the learning process until the error is within the specified values. After training, the ANN can predict the outputs from given inputs. Backpropagation is the most often used systematic method to train multilayer ANNs. Curtiss et al. (1996) and Miller and Seem (1991) recommended the following training procedure:

1. Assign initial random values for connection weights, often between -0.5 and $+0.5$. Select a training input output pair; calculate the normalized input activations i_n and the input activation to the neurons in the first hidden layer according to Eqs. (5.13) and (5.14).
2. Calculate the outputs of the 1, 2, . . ., L hidden layers o_{1n}, o_{2n}, . . ., o_{Ln} and then of the output layer o_{on} from Eqs. (5.15) and (5.16).

3. Evaluate the error δ between the calculated o_{on} and the selected training output (target output) o_t by:

$$\delta = o_{on}(1 - o_{on})(o_t - o_{on}) \tag{5.18}$$

4. Adjust the connection weights from W_j to W_{j+1} to minimize the error by using the following rule:

$$W_{j+1} = W_j + \eta\delta o_{on} \tag{5.19}$$

where η is the learning rate whose value lies between 0 and 1.

5. Repeat the previous steps for all the input/output pairs in the training set until the error for the entire training set is lower than the preset training tolerance. The training set should cover the operating range of the inputs and outputs.

Applications. An ANN can model multiple parameters simultaneously for nonlinear systems. It can also be periodically trained to update the weights. ANNs are now widely used for predictive control, such as energy use prediction, energy optimization, adaptive control, data trending, and optimum start and stop.

5.11 PROGRAMMING OF DDC SYSTEMS

Evolution of DDC Programming

The EMCSs having more complex functional controls are heavily software-driven. From the 1960s to the 1970s, the software for HVAC&R control was mostly performed in a workstation computer and programmed in the control manufacturer's factory. Because of the trend to use distributed microprocessor-based DDC units in the late 1970s, many EMCS manufacturers provide some type of operator control language (OCL) for field-programmable line programming in the controllers using BASIC-type language. This meets the requirements of computer program development, periodic updating, and necessary modifications and improvements.

Since the late 1980s, there is a trend toward the function and object-orientated graphical programming to provide software for DDC units and the PC in the operator's workstation. Today, both traditional line programming using BASIC-like language (sometimes Pascal and C++ languages are also used) and graphical programming are used in DDC systems.

Graphical Programming

Davison (1992) described that graphical programming is a schematic drawing of a desirable control scheme (functional control), such as mechanical cooling or economizer control, using symbols called templates. The templates are displayed on the computer screen interconnected by lines that direct the flow of data. The control scheme shows the inputs, through control operations, to outputs. After the diagram of the graphical programming is completed, it is converted to a program usable by the DDC units and PC through a computer program.

Compared with traditional line programming using a BASIC-like language, graphical programming has the following advantages: First, it is intuitive and easier to use and understand by field HVAC&R engineers and operators. Second, the user is more familiar with it because of meaningful visual symbols represent functions and devices. It is simpler in documentation and easier in troubleshooting. Third, it provides a time-saving tool for the specification and documentation of the HVAC&R energy management and control systems. Finally, graphical programming enables one to improve programming quality at a reduced cost. The limitations of graphical programming are that

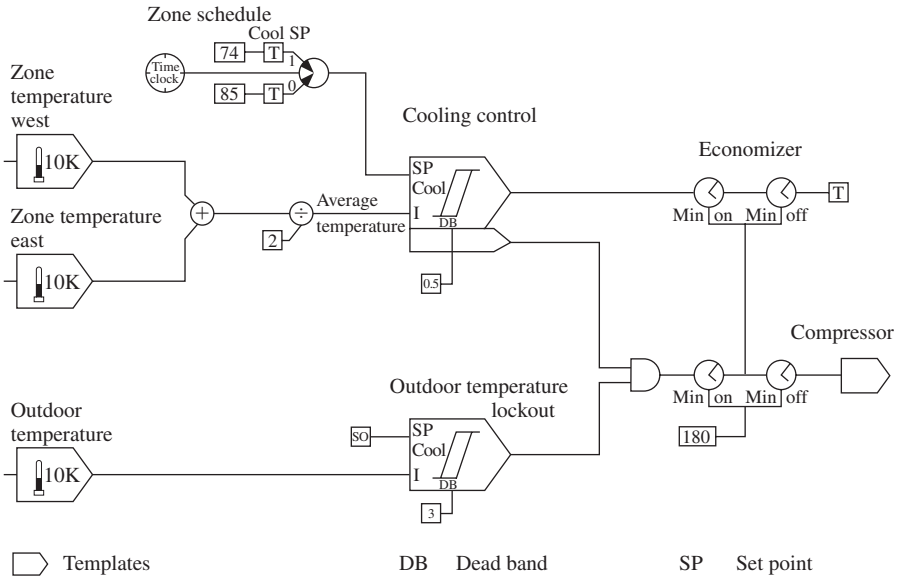


FIGURE 5.26 Graphical programming for mechanical cooling control in a small packaged unit.

it is bulky to display on screen and that it is new to us and needs to develop required supports such as a portable interface that acts directly to the DDC units.

Templates

In Davison's (1992) paper, a template is a graphical symbol (icon) in graphical programming that describes a single or combined control scheme (specific functional control), as shown in Fig. 5.26. A template consists of input and output connections, a small section of computer code, and private data storage registers. A template is function- and object-oriented. Each template performs a small portion of a specific functional control. If several templates are connected by lines, they form a complete computer program for a specific functional control. The template symbol provides a visual reminder or memory of the function that the template accomplished.

A template maintains its own set of private data and variables that cannot be altered by the action of any other template. The state of an instance of a template is contained in its private variables. They can be only modified by the program code contained in that instance. The private variables of an instance of a template are hidden from the action from other templates.

In a template, inputs are used to receive data from other templates or constants. Outputs are used to send data to other templates. Inputs and outputs between two or more templates are connected by lines that direct the data flow between the templates. An input may receive a value from only one output in order to prevent overwriting of data at the input. On the other hand, an output data may be sent and used by any number of inputs.

For a more complex control function that needs several templates, a combination of templates called a macrotemplate is often used to simplify a graphical programming diagram.

Graphical Programming for Mechanical Cooling Control

Figure 5.26 shows the graphical programming for mechanical cooling control for a small packaged unit. This packaged unit has a supply fan, a DX coil, a single-scroll compressor, and an economizer. The graphical programming for the control of this packaged unit includes mechanical cooling, economizer, heating, and fan control.

During the scheduled occupied time, the zone schedule produces a 1.0 signal, and the cooling control template SP selects the input labeled 1.0 and passes the value 74°F (23.3°C) as the cooling set point. During unoccupied periods, SP passes the value 85°F (29.4°C). The average temperature of east and west zones is compared with the current set point of the cooling dead-band control template. If the average temperature exceeds the set point plus the dead-band value, 0.5°F (0.28°C), or 74.5°F (23.6°C), then the output of the cooling control template goes to 1.0, and the compressor is turned on. If the zone average temperature drops below the set point minus the dead-band value, 73.5°F (23.1°C), then the output of the cooling control template goes to 0.0 and the compressor is turned off.

When the outdoor air temperature drops below 50°F (10.0°C) minus the dead-band value 3°F (1.7°C), or 47°F (8.3°C), then the outdoor temperature lockout template produces 1.0, and the compressor is locked out (turned off). Minimum on and minimum off timer templates prevent the compressor from operating at short cycles.

Graphical programming of DDC systems may become supplemental to the sequence of operation and control diagrams, and may become a part of the DDC design documentation.

5.12 TUNING DDC UNITS

Most of the DDC units use PI or PID control modes. Tuning determines the gains and control parameters which have a direct impact on the steady-state error and transient characteristics of the DDC unit and the control system. A well-tuned DDC unit minimizes the steady-state error, or offset from the set point; shows a quick response to disturbance; and provides operating stability at all operating conditions. HVAC&R processes are nonlinear, and system characteristics change when seasons are varied. The DDC unit tuned at one condition may not be appropriate at other operating conditions.

Tuning PI Controllers

For DDC units using PI control modes, proportional gain and integral gain should be properly selected. A high gain decreases the control stability and the offset or error, whereas a low gain will produce a slow response, which increases the control stability and offset. In tuning PI controllers, trial and error is often used by the EMCS subcontractor. The trial-and-error method adjusts the gain until the desirable response to a set-point change is shown. This response should start with a small overshoot and rapidly damp to steady-state conditions. The trial-and-error method is time-consuming. Other tuning methods for PI controllers, such as closed- and open-loop process identification, are also used. Refer to *ASHRAE Handbook 1995, HVAC Applications*, for details. Bekker et al. (1991) recommended a root locus tuning method for first-order processes like most temperature and humidity control for PI controllers to achieve a critically damped response.

Self-Tuning PI and PID Controllers

During the 1980s, different schemes were developed including Astrom and Hagglund's (1984) automatic tuning of PI and PID controllers. The tuning procedure suggested by Astrom and Hagglund is based on the identification of one point on the Nyquist curve of an open-loop system

with relay feedback. In the earlier 1990s, commercial products of automatic tuning DDC units started to appear in the EMCS in buildings.

Based on Astrom and Hagglund's principle, Wallenborg (1991) accomplished automatic tuning of PID controllers in supply air temperature and duct static pressure control experiments. After automatic tuning, its system performance was improved. The autotuner used in these experiments was different from a "true adaptive controller." An autotuner is operated with fixed parameters during normal operation. The tuning experiment is initiated between the normal operation periods by the operator. The required parameters are calculated from the results of the tuning experiments.

Adaptive Control

The controller has the ability to adapt to the control system by determining the optimum PID parameters and adjusting itself accordingly. An adaptive controller, or self-tuning controller, continuously updates its parameter during operation based on some on-line process identifier and computer programs. Self-tuning adaptive controllers are also available as commercial products now.

5.13 FACTORS AFFECTING CONTROL PROCESSES

As defined in Sec. 5.1, the function of an air conditioning (HVAC&R) control system is to modulate the air conditioning system capacity to match the off-design condition, load variation, and climate change, to maintain the indoor environment within desirable limits at optimum energy use.

Load

The term *load* refers to the magnitude of the space load, coil load, refrigeration load, or boiler load that determines the amount of the supply air, chilled water, or hot water needed to control and maintain the controlled variable at the desirable value(s). Load variation and disturbances affect the controlled variable in three circumstances: part-load operation, intermittent operation, and disturbances.

Part-Load Operation. The sizes of an air conditioning system and its components are always selected according to the magnitude of the load at the design condition, which is often called the design load or full load. In actual operation, air conditioning or HVAC&R systems operate at part load most of the time. Many air systems even spend 85 to 90 percent of their annual operating hours at part-load operation. The load drops below the design load because of

1. Changes in the outdoor climate
2. Changes in the internal loads at the time of operation

During part-load operation, the capacity of an air conditioning system must be appropriately reduced so that a desirable indoor environment can be maintained; at the same time, the energy use of the HVAC&R equipment can be saved.

Climate Change

Outdoor climate change affects not only the space load, but also the performance of HVAC&R systems. Outdoor climate parameters for HVAC&R include the dry-bulb temperature, wet-bulb temperature, solar radiation, wind speed, and wind direction. They are usually specified at design conditions for an HVAC&R system or at a rated condition for certain equipment. Outdoor climate

changes during off-design conditions. The general trend in summer when the outdoor dry- and wet-bulb temperatures fall is that the space cooling load drops accordingly. Because each system has its own characteristics, sometimes it is simpler and more convenient to consider the load ratio and climate parameter as two different factors during the calculation and analysis of system performance. For instance, the analysis of the energy use of a water-cooled chiller requires both the load ratio and the entering condenser water temperature (ECWT) to determine the power input to the compressor.

Intermittent Operation. Many air systems do not operate continuously. Some operate only few hours within a diurnal cycle. Others operate only in the daytime and shut down at night when the building is unoccupied. During the warm-up and cool-down periods, space loads vary a great deal from the design load as well as from that of a continuous part-load operation. The space loads represent transient loads of a dynamic model. Optimum starting and stopping of intermittently operated air systems is important for better indoor environmental control and energy savings.

Disturbance. These can be sudden load changes or set-point changes within a short time, say, a fraction of an hour, that affect the controlled variable. The offset of the controlled variable resulting from a disturbance can be eliminated by a DDC system with a proportional-integral control mode. For disturbances resulting from a sudden climate change that affects a 100 percent outdoor air-handling unit, or from a sudden switch-on of the spotlights in a conditioned space, a DDC system that incorporates proportional-integral-derivative control mode may be more suitable.

System Capacity

First, the capacity of an air conditioning (HVAC&R) system and its components should be adequate. An oversized electric heater produces a greater overshoot of space temperature and wastes more energy than an appropriately sized heater. An undersized chiller always causes a higher space temperature during the cooling season and the EMCS can become out of control.

Second, from the point of view of capacity control and energy conservation, a heating and cooling plant installed with multiple units is always better than a single-unit plant. A piece of equipment will operate more efficiently at a higher turndown ratio than at a lower turndown ratio. The turndown ratio R_{TD}, often called part-load capability, is usually expressed as the percentage of design capacity:

$$R_{TR} = \frac{\text{minimum capacity}}{\text{design capacity}} \times 100 \tag{5.20}$$

Third, modulation of capacity continuously is always better for matching the load to outdoor climate change than two-position or step controls. Finally, capacity modulation of large fans, pumps, and compressors using adjustable-frequency variable-speed drives is now often energy-efficient and cost-effective in many circumstances.

Performance of Control Processes

The performance of an EMCS is often represented by the performance of its control processes, or more directly its control loops that perform the control functions. HVAC&R control processes are mostly nonlinear processes whose characteristics cannot be expressed by first-order equations. Nordeen (1995) listed five criteria to use to evaluate the performance of an HVAC&R control process:

- *Stability*. A control system is stable if the controlled variable does not show a continuing trend away from the set point following a disturbance, as shown in Fig. 5.9*a*. As described previously, a wider throttling range and a slower response both are beneficial to system stability. The decay of

the oscillation of the controlled variable is called damping. The more damped a control system is, the less it oscillates.

- *Response time*. The length of time needed for the controlled variable to reach the set point after a step change in set point or load is the response time.
- *Overshoot*. The difference between the maximum controlled variable and the set point following a change in set point or load is the overshoot.
- *Settling time*. The length of time needed for the controlled variable to reach steady-state following a disturbance.
- *Offset*. As defined previously, this is the constant error that exists between the controlled variable and the set point when a steady state is reached.

The problems associated with poor performance include the time lag or dead time, which is large compared to the reaction rate or time constant of a control system; poor measurements of the controlled variables; inappropriate capacity control; and hysteresis within control and control components which create time lag. Time lag or dead time is the delay in time between the change in the controlled variable and when that change is sensed by the sensor, or when the controlled device is modulated, or when the capacity of the process plant varies.

During the sizing of control valves and air dampers, a linear relationship between the coil load and the valve stem travel was the primary design consideration during the past decades. However, many microprocessor-based DDC systems now permit scaling factors to be applied to the analog outputs of the DDC units, and thus an inherently nonlinear system will respond in a linear relationship.

Thermal Capacitance

Thermal capacitance, which is sometimes called *thermal inertia*, is related to the mass of the building envelope, equipment, and system components. Capacitance is usually calculated by the product of specific heat of the material and the mass of the material. The thermal capacitance of the HVAC&R process and the building envelope affects the performance of a control system as follows.

First, the high thermal capacitance of the building envelope, the equipment, chilled and hot water, etc., reduce the effect of a disturbance to vary the controlled variable. For example, consider a sudden increase in the lighting load in a conditioned space. As the space air temperature increases because of heat released from the electric lights, a large portion of heat at the same time will be transferred to the building envelope because of the additional temperature difference between the space air and the building structure. The increase in the space temperature resulting from the disturbance is thus significantly reduced.

Second, during the warm-up and cool-down periods, because of the high thermal capacitance of the building envelope, heat exchanger, ducts, and pipes, a greater system capacity and longer period of time are needed to raise or cool down the space air temperature.

Third, the thermal influence of the weather and of external load changes on the conditions of the indoor space air through the building envelope is rather slow, often needing several hours to reach full effect.

5.14 FUNCTIONAL CONTROLS

The control functions of a microprocessor-based EMCS in a large high-rise building become more and more complex and demanding. To provide the required functional controls with satisfactory system performance is always the primary target of an EMCS. In an EMCS, there are four kinds of functional controls: generic, specific, safety, and diagnostics.

Generic Controls

Generic controls are usually needed by most of the systems, units (equipment), and components. The software of generic controls is usually provided in the PC workstation and in the system controllers except system graphical displays. The editing of the generic software usually takes place in the PC workstation and can be downloaded to any one of the system controllers. In many EMCSs, data and information can be monitored and collected from any point in the system and analyzed in the working station PC and in the system controllers. Graphics displays, trending, totalization, scheduling, alarming, etc., are examples of the generic controls. The following are features provided by various manufacturers (or vendors).

Graphical Displays. The graphical displays provided in the PC workstation become the showpiece of an EMCS. Some EMCS manufacturers demand that the software in the operator PC workstation be graphically oriented. The graphical displays include the following:

- Building floor plan graphics show the selected floor plan and the space temperatures.
- Equipment graphics are provided for each major piece of equipment such as packaged unit, air-handling unit, and chiller with status of all points.
- Schematic graphics show the detailed system drawings which can be created, modified, and saved.
- High-resolution digitized photo-quality displays are also accommodated in some EMCSs.

The graphical display system can allow a display of up to 10 graphical screens for comparison and monitoring of system status. Graphics combined with color coding, such as temperature that is assigned with different colors at various values, will increase viewer's effectiveness. A graphical display system should provide navigation from a facility map down to a specific floor plan. Associated temperature and related parameters are provided on the floor plan. An additional "click" of the mouse on any questionable area will cause the serving HVAC&R system with its operating information to appear on the screen.

Speed and convenience of editing are two key issues in graphical displays. If a 486 processor is used, graphics may take 5 to 7 s to paint. Including the navigating through 5 to 10 screens in search of a problem, too much time will be needed. Because of the use of the Pentium chip, it is possible to reduce the graphical response to 2 to 3 s per screen paint. The operator must create, modify, or edit the graphics easily.

Graphical displays play a small role in increasing the effectiveness of any functional control in an EMCS. However, they enable the user to interact more efficiently with the data and information from the EMCS and to make proper decisions to operate the EMCS and the associated HVAC&R system effectively and efficiently.

Trending. Trending is the ability to provide continuous track of certain parameter(s), or operating status of a piece of equipment. The commonly used time interval for the trend log is once for every 30 min. Trending is mostly used for troubleshooting. For an example, if the output capacity of a heat exchanger is gradually reduced, scale may form on the heat-exchange surface. Totalization is another trend log which records the total accumulated operated time for each of the units in a plant employing multiple units. Totalization is usually helpful for maintenance and troubleshooting purposes.

The trend graph displays the trend data in graphical form. When the user asks for a trend graph, select the point(s). The *Y* axis of the graph is often automatically scaled; the *X* axis to indicate time is appropriately labeled. The user can also choose the capacity of trend points, time interval between two trend data other than the standard 30 min, and selection of both changes of value (COV) that exceed the defined value and the timed trend. The trend graph is a powerful tool. It is very helpful for tuning of PID control loops and analyzing operating problems.

Scheduling. Prior to the DDC systems, the use of a time clock to schedule the start and stop of equipment automatically was most widely adopted. A DDC system usually starts and stops the

equipment according to the predetermined schedule in the system controllers. They can be in the form of day, week, month, and season schedules, and the users can define the executing periods. The graphical schedules are often used for clarification. Modifications of schedules are easily made at the PC workstation and downloaded to the system controllers. Optimum start and stop are actually run optimizing of scheduling.

Some manufacturers provide temporary schedules that will be used only once and supplement the existing schedule. Sometimes an override schedule which supersedes the existing schedule is also used only once.

Alarming. This is the software for tracking and reporting the alarm conditions, i.e., abnormal conditions. The operator needs to determine the limits or the difference from the set points that causes the abnormal condition. The operator also should determine the reactions, logging, printing, displaying messages and graphics, and producing audible announcements that are taken during an alarm. An alarm that is not acknowledged by the operator within a specific time will move to a higher level of priority. The system must automatically lock out alarms when an alarmed system is turned off and appropriate reactions have been taken.

Discriminator Control. Discriminator control is a kind of optimum control which searches for the required lowest cooling supply air temperature and the highest heating supply temperature. This control also minimizes the amount of mixing of cold and warm supply air, terminal reheat, and resets the cold deck supply temperature to reduce the mechanical cooling and the amount of reheat.

Fault Detection and Diagnostics. Fault detection and diagnostics should monitor the operation of HVAC&R equipment, components, and control devices; analyze the data; and identify performance problems to be corrected. Fault detection and diagnostics are discussed in detail in the next section.

Specific Controls

Specific controls including capacity controls are controls for a specific function in an HVAC&R system, a unit (or equipment), or a zone. The following are specific controls that are discussed in the corresponding sections later:

- Zone temperature control
  - Fan-coil unit
  - VAV reheat
  - Fan-powered VAV box
- Air system control
  - Economizer control
  - Discharge air temperature control
  - Minimum ventilation control
  - Duct static pressure control
  - Space humidity control
  - Warm-up or cool-down control
- Water system control
  - Differential pressure control
  - Chilled water temperature reset
  - Condenser water temperature control
  - Low-temperature heating system control

- Central plant control
  - Multiple-chiller optimizing control
  - Multiple-boiler optimizing control
  - Condenser fan cycling
  - Demand limit control
- Safety controls

Commissioning and Maintenance

A properly designed and installed EMCS needs commissioning to test and tune its controllers and system components according to the design specification. A sufficient, clear, and well-followed operations manual and a well-implemented maintenance schedule are key factors for an effective and efficient EMCS. Commissioning and maintenance are covered in detail in Chap. 32.

5.15 FAULT DETECTION AND DIAGNOSTICS

Basics

Modern air-handling units (AHUs), packaged units (PUs), and chillers become more and more complicated because of the IAQ, thermal comfort and energy efficiency requirements, and the use of DDC. An HVAC&R operator is hardly able to monitor and detect the fault operations of the AHUs, PUs, and chillers; find their causes; and correct them. An automatic fault detection and diagnostic system is important for the effective operation and control, for the monitoring and maintenance, and for the optimizing of utilization and continuous improving of the HVAC&R systems. As of the late 1990s, fault detection and diagnostics are already a standard component in many large PUs.

In HVAC&R operations, faults occur when the actual measured operating parameters deviate from the normal operating values. There are two types of detected faults: complete failures and performance degradations. Complete failures are abrupt faults that often cause discontinuation of the operation of a system or component. Symptoms of abrupt faults can be easily observed. Peformance degradation is the result of an evolving fault accumulated during a certain time.

The differences between the actually measured values of an operating parameter, such as temperature T, °F (°C); pressure p, in. WG (Pa); volume flow rate $\dot{V}$, cfm (m^3/s); or mass flow rate $\dot{m}$, lb/s (kg/s) and the expected values (estimated, simulated, or set points) of temperature T_{exp}, pressure p_{exp}, volume flow rate $\dot{V}_{exp}$, or mass flow rate $\dot{m}_{exp}$ under normal operating conditions are called *residual*. A fault can be detected by investigating and analyzing residuals.

A temperature residual T_{res}, °F (°C), a pressure residual p_{res}, in. WC (Pa), or a volume flow rate residual $\dot{V}_{res}$, cfm (m^3/s), can be calculated as

$$\begin{aligned} T_{res} &= T - T_{exp} \\ p_{res} &= p - p_{exp} \\ \dot{V}_{res} &= \dot{V} - \dot{V}_{exp} \end{aligned} \tag{5.21}$$

where the subscript exp indicates expected (predicted) values and the units of T_{exp}, p_{exp}, and $\dot{V}_{exp}$ are the same as those of T, p, and $\dot{V}$. Most of the measured operating parameters in a fault detection and diagnostic system are the same monitored parameters (sensed or measured) in an EMCS.

Residuals are often normalized so that the dominant symptom may have approximately the same

magnitude for different types of faults. The residual R can be normalized as

$$R_{\text{nor}} = \frac{R - R_{\text{min}}}{R_{\text{max}} - R_{\text{min}}} \tag{5.22}$$

where R_{nor} = normalized residual
$R_{\text{max}}, R_{\text{min}}$ = maximum and minimum residuals

In the late 1990s, a large rooftop packaged unit made by one U. S. manufacturer offered a standard fault detection and diagnostic device to review active and historical lists of diagnostic conditions. A total of 49 different diagnostics can be read at the human interface panel, and the last 20 diagnostics can be held in an active history buffer log at the panel. A human interface panel provides 2-line × 40-character liquid-crystal display and a 16-button keypad for monitoring, setting, controlling, and diagnostics.

In the late 1980s, the earlier development of diagnostics of HVAC&R system operation was mainly rule-based expert systems. During the late 1990s, the development of automating fault detection and diagnosis was emphasized. Inputs and outputs of an HVAC&R operating process can be mathematically related by using autoregressive models with exogenous inputs (ARX), artificial neural network (ANN) models, and many other developing models. Both ARX and ANN are called black-box because they require less physical knowledge of the operating process. These technologies are expected to be commercially available after laboratory and field tests in the early 2000s.

Expert System Rule-Based Diagnostics

Expert systems are discussed in Sec. 5.10, and diagnostics is one of the applications used in HVAC&R system operations. Brambley et al. (1998) reported a tool—the outdoor air/economizer (OAE) diagnostician—which monitors the performance of an air-handling unit to provide outdoor air as a constant fraction of its supply flow rate and detects problems of outdoor ventilation air control and economizer operation. An OAE diagnostician uses sensors that are commonly installed for control purposes and diagnoses the operating problems based on rules derived from engineering models of proper and improper performance of the AHUs. These rules are implemented in a decision tree structure in computer software. The diagnostician collects data periodically from the Building Automation System (BAS) to navigate the decision tree and produce conclusions. At each point of the tree, a rule is evaluted according to the collected data, and the results determine in which branch the diagnosis should be. At the end of the branch, a conclusion is reached corresponding to the current operating condition of the AHU. The operator or installer of the diagnostician enters data only once during setup.

The OAE diagnostician was installed and operated on three AHUs in a newly constructed and occupied 200,000-ft^2 (18,600-m^2) DOE William R. Wiley Environmental Molecular Sciences Laboratory and on four AHUs in a 72,700-ft^2 (6760-m^2) Technical Management Center, both in Rishland, Washington.

For each AHU, data were recorded hourly from sensors in the BAS for outside temperature, return air temperature, mixed air temperature, discharge air temperature, on/off status of the supply fan, and open/closed status of the chilled water and hot water valves. No sensors were installed specially for the diagnostician; all are used by the BAS for the control purpose. The data were automatically transferred hourly from the BAS to the diagnostician's database. The diagnostician then processed these data and produced the diagnostic results that can be viewed on the display.

Shortly after initial processing of data, four of the seven AHUs monitored were found to have problems. The problems included a sensor malfunction, a return damper that was not closed fully, a mixed-air sensor problem, and a chilled water control problem. All problems were confirmed after inspection.

The diagnostician used color coding on a display to alert the HVAC&R system operator when the problem happened and provided assistance in modifying the causes as well as making the correction. The OAE diagnostician has proved itself effective in identifying installation and operating problems during the field testing of the outdoor ventilation control and economizer cycle.

Dexter and Benouarets (1996) introduced a fault diagnosis of HVAC&R plant items based on semiqualitative generic reference models. The scheme uses reference models describing fault-free and fault operations that are collected from data producd by simulating a number of HVAC&R systems (or subsystems) of the same type. Results have showed that a fuzzy-logic-based fault diagnosis requiring no training on the actual plant can successfully identify faults in a simulated HVAC&R subsystem if the parameter denoting faults is sufficiently large. The proposed fuzzy method of fault diagnosis is computationally simple enough to be used for more complex subsystems, such as AHUs.

Stylianou and Nikanpour (1996) present a methodology that uses thermodynamic modeling, pattern recognition, and expert knowledge to determine whether a reciprocating chiller is fault-free and to diagnose selected faults. The status of this chiller consists of the off cycle, start-up, and steady-state operation. The off cycle deals with those faults that are more easily detected when the chiller is off. Start-up deals with faults which are related to refrigerant flow characteristics and are generally more apparent during the transient period. During steady-state operation, faults of performance deterioration are detected.

This methodology requires training data. Data can be collected during commissioning and online measurement of variables to build up normal steady-state linear regression models and data collected from manufacturers. This methodology also needs additional treatment to improve the definition of the thresholds to classify emerging patterns and to establish the range of applicability of these patterns. Further study is required to produce a list of faults and their associated transient characteristics and steady-state patterns.

ARX and ANN Model-Based Diagnosis

Peitsman and Bakker (1996), Peitsman and Soethout (1997), Lee et al. (1996), and Yoshida et al. (1996) discussed the application of ARX and ANN model-based fault detection and diagnosis and the related laboratory testing results. Model-based diagnosis is a technique capable of finding diagnoses based on the behavior of the system and components. The behavior is best understood as the interaction of observation and prediction expressed in the difference between the measured (observed) and predicted parameter values as previously covered, called residual. The behavior of the system and components is defined using ARX or ANN models. One advantage of a model-based diagnosis over rule-based diagnosis is that the model-based diagnosis does not rely on symptom-fault patterns. Such patterns are incomplete, and it is difficult for an expert to anticipate all possible faults and to predict their symptoms. Another advantage is that a model-based diagnosis can be used for a new system even when there is no repair experience or only when a system model is available.

ARX Model. Consider a process in an autoregressive model with exogenous inputs (ARX) having an input signal $u(t)$ and an output signal $y(t)$ and described by a simple input/output relationship as

$$y(t) + a_1 y(t-1) + \cdots + a_n y(t-n) = b_1 u(t-1) + \cdots + b_m u(t-m) + e(t) \qquad (5.23)$$

where t = time
$e(t)$ = enter as a direct error
a = autoregressive parameter (order n)
b = exogenous parameter (order m)

The model relating output and input and extra (exogenous) variables is linear. The actual performance of a system is nonlinear. However, such an approximation is allowable in fault detection and

diagnosis. Extra inputs are added, and the use of historical measured inputs ($t - 1$, $t - 2$, . . .) will make it more nearly a nonlinear relationship.

Procedure to Identify Parameters. After the ARX model has been selected, according to Peitsman and Bakker (1996), the procedure to identify parameters a and b in Eq. (5.23) is as follows:

- Measure input/output data. Input and output values of the process are collected in a data set that consists of measured and design (predicted) data under "healthy" (fault-free) conditions. The measured data sets should contain sufficient dynamic conditions of the process.
- Model order (m and n) should be selected as the optimum values by trial and error.
- From the collected input/output data, estimate parameters a and b.
- Validate the selected model. The accuracy of the model is tested by comparing the predicted value with the measured values.

ANN Models. For an ANN model of three layers with an input layer, a hidden layer, and an output layer, as discussed in Sec. 5.10, the outputs can be calculated as a function of the weighted sum of the inputs, and an additional factor, the biases B as expressed in Eqs. (5.16) and (5.17):

$$o_{on} = \frac{1}{1 + e^{-(V-B)}}$$

$$V = \sum i_{Ln} W_{Lon}$$

where i_{Ln} = input activation to a neuron in hidden layer
W_{Lon} = weights of connection between hidden layer and outputs

ANN models are nonlinear. A learning or training process in an ANN model is necessary. The initially assigned connection weights are adjusted repeatedly during the learning and training process until the error is within specified values, as discussed in Sec. 5.10. After training, an ANN model can predict the outputs from given inputs.

Reliability of ARX and ANN Models. Peitsman and Bakker (1996) recognized that the reliability of the ARX and ANN models (the use of mathematic equations to forecast the outputs) is good. One of the conditions needed to produce a reliable model is a healthy and dynamic data set that is measured under optimal and good operating conditions over a wide working range. Another necessary condition is the availability of a fixed time interval (time step) because the regression models use values of the previous time interval of the inputs and outputs to predict the output.

System and Component Models. In Peitsman and Bakker (1996), two types of models are produced with the measured and predicted learned data:

- *System models*. A chiller or an AHU is considered a black box in which multiple independent inputs estimate outputs. Independent variables are necessary to form a system model. This model checks the whole system performance. In a system model if the possibility of a malfunction is detected, the next step is to use the component model to localize this malfunction.
- *Component models*. Major components are considered as black boxes. The configuration of component models is accomplished after studying the structure of the modeled HVAC&R systems. The model becomes a little bit gray (with some physical knowledge) instead of black (without physical knowledge). After the system model has detected a malfunction, the component model can pinpoint the cause of the malfunction with a greater accuracy than the system model can.

Fault Detection and Diagnosis. After the parameters (variables) of a system model or component model are simulated using an ARX or ANN model, the predicted outputs of the model must be

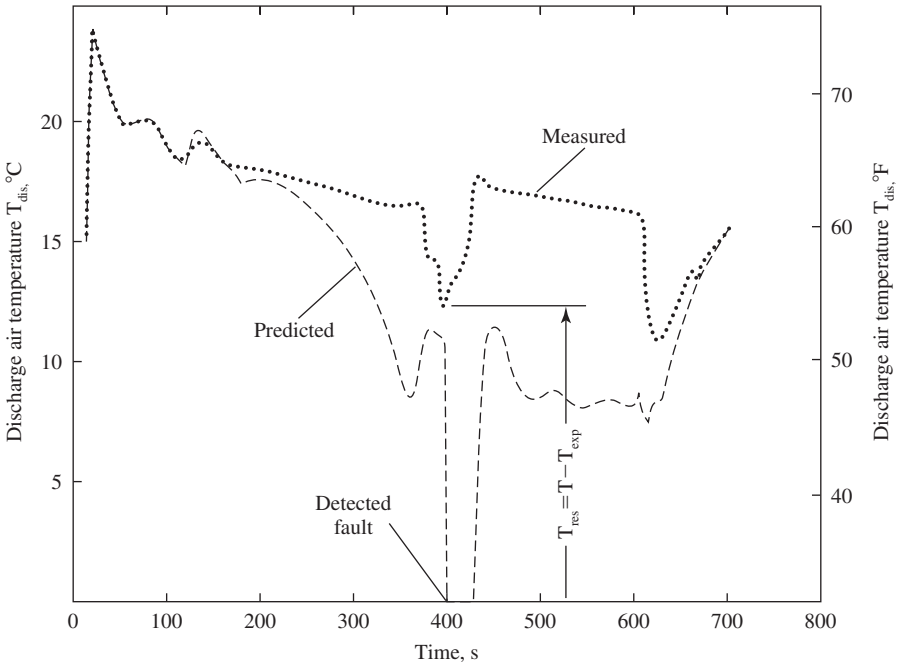


FIGURE 5.27 Comparison of the measured and predicted discharge air temperatures for a VAV system in an ARX system model. (*Source: Peitsman and Bakker (1996) ASHRAE Transactions Part I 6637. Reprinted with permission.*)

compared with the measured outputs of the system or the components (temperature, pressure, volume flow) as shown in Fig. 5.27. Only when the measured output is not within the threshold of the predicted output may there be a detected fault or an incorrect model used.

The threshold is usually determined from the statistical properties of the process. A residual (margin) of $|x - x_{exp}| = 3\sigma$ is used. Here x indicates the measured value, x_{exp} is the assumed predicted or expected mean value, and σ is the standard deviation of the predicted values. The probability that x falls within $[x_{exp} - 3\sigma, x_{exp} + 3\sigma]$ is 0.9973. If $|x - x_{exp}| > 3\sigma$, a detected fault may occur, as shown in Fig. 5.27, in which the difference between the measured and predicted discharge air temperatures is greater than 3σ. The possible causes of the fault include block of chilled water flow, high chilled water temperature, and a fouled cooling coil.

Comparison of ARX and ANN Models. According to Peitsman and Bakker (1996), the comparison of ARX and ANN models for fault detection and diagnosis is as follows:

	ARX	ANN
Prediction of process variables	Linear	Nonlinear
Results	ANN gives slightly better results than ARX models	
Training period	Shorter	Longer

5.16 CONTROLS IN ASHRAE/IESNA STANDARD 90.1-1999

ASHRAE/IESNA Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* specifies the following mandatory provisions for HVAC&R controls. For exceptions and details refer to Standard 90.1-1999.

General

- *Thermostatic controls*. The supply of heating and cooling energy to each zone shall be controlled individually by thermostatic controls using the zone temperature as the response. A dwelling unit is permitted to be considered as a single zone.
- *Dead band*. When both heating and cooling are controlled, zone thermostatic controls shall be capable of providing a temperature range or dead band of at least 5°F (2.8°C). Within that range the heating and cooling energy supplied to the zone is shut off.
- *Set-point overlap*. When heating and cooling supply to a zone are controlled by separate zone thermostatic controls, means such as limit switches, mechanical stops, and software for DDC systems, shall be provided to prevent the heating set point exceeding the cooling set point minus any proportional band.

Off-Hour Controls

HVAC systems with a design heating or cooling capacity greater than 65,000 Btu/h (19,050 W) and fan system power greater than $^3/_4$ hp (0.56 kW) shall have all the following off-hour controls: automatic shutdown, setback controls, optimum start controls, shutoff damper controls, and zone isolation.

- *Automatic shutdown*. HVAC systems shall be equipped with at least one of the following:
  1. Controls that can start and stop the system under different time schedules, are capable of retaining programming and time setting during loss of power for a period at least 10 hours, and include an accessible manual override or equivalent that allows temporary operation of the system for up to 2 hours.
  2. An occupancy sensor that can shut the system off when no occupant is sensed for a period of up to 30 minutes.
  3. A manual operated timer that can adjust the system for two hours.
  4. An interlock to a security system which shuts the system off when the security system is activated.
- *Setback controls*. A heating system located where the heating outdoor design temperature is 40°F (4.4°C) or less shall be equipped with controls that can automatically restart and temporarily operate the system as required to maintain zone temperature above a heating set point adjustable down to 55°F (12.8°C) or lower. A cooling system located where the cooling outdoor design temperature is greater than 100°F (37.8°C) shall be equipped with controls that can automatically restart and temporarily operate the system as required to maintain zone temperature below a cooling set point adjustable up to 90°F (32.2°C) or higher or to prevent high space humidity.
- *Optimum start controls*. An air system with a supply volume flow rate exceeding 10,000 cfm (4720 L/s), served by one or more supply fans, shall have optimum controls. The optimum start shall be at least a function of the difference between space temperature and occupied setpoint, and the amount of time prior to scheduled occupancy.
- *Shutoff damper controls*. Both outdoor air supply and exhaust system shall be installed with motorized dampers that will automatically shut off when the systems or the space are not in use. Out-

door dampers for ventilation can be automatically shut off during morning warm-up and cool-down, or nighttime setback except when ventilation reduces energy costs, or ventilation must be supplied. Both outdoor air or exhaust dampers shall have a maximum air leakage of 3 cfm/ft^2 (15 L/s·m^2) at a pressure difference of 1.0 in WC (250 Pa). As an exception, gravity dampers (nonmotorized) are acceptable in buildings less than three stories in height or for buildings of any height in climates with HDD65 less than 2700. Also excepted are systems with a design outside air intake of 300 cfm (142 L/s) or less that are equipped with motor operated dampers.

- *Zone isolation*. HVAC systems serving zones that are intended to operate or be occupied nonsimultaneously shall be divided into isolation areas. Each isolation area shall be no larger than 25,000 ft^2 (2320 m^2) of floor area nor include more than one floor. Each isolation area shall be equipped wih isolation devices that can automatically shut off the supply of conditoned air, outside air, and exhaust air from the area. Each isolation area can be controlled independently.

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CHAPTER 6
LOAD CALCULATIONS

6.1 SPACE LOAD CHARACTERISTICS 6.2
- Space, Room, and Zone 6.2
- Convective and Radiative Heat 6.2
- Space and Equipment Loads 6.3
- Night Shutdown Operating Mode 6.3
- Influence of Stored Heat 6.6

6.2 COOLING LOAD AND COIL LOAD CALCULATIONS 6.6
- Components of Cooling Load 6.6
- Components of Cooling Coil Load 6.7
- Difference between Cooling Load and Cooling Coil Load 6.8
- Load Profile 6.9
- Peak Load and Block Load 6.9
- Characteristics of Night Shutdown Operating Mode 6.10
- Moisture Transfer from the Building Structures 6.10

6.3 HISTORICAL DEVELOPMENT OF COOLING LOAD CALCULATIONS 6.11

6.4 METHODOLOGY—HEAT BALANCE 6.12
- The Physical Model 6.12
- Heat Balance Equations 6.12
- Heat Balance of Space Air 6.13
- Characteristics of Heat Balance Method 6.14

6.5 METHODOLOGY—TRANSFER FUNCTION 6.14
- Basics 6.14
- Transfer Function and Time Function 6.14
- Calculation Procedure 6.15
- CLTD/SCL/CLF Method 6.15
- TETD/TA Method 6.15

6.6 DETAILED CALCULATION PROCEDURES FOR TFM 6.16
- Conduction Heat Gain through Exterior Walls and Roofs 6.16
- Heat Gain through Ceilings, Floors, and Interior Partition Walls 6.16
- Solar Heat Gain and Conductive Heat Gain through Window Glass 6.17
- Internal Heat Gain 6.17
- Infiltration 6.24
- Cooling Load Conversion Using Room Transfer Function 6.24
- Space Cooling Load Calculation 6.25
- Heat Extraction Rate and Space Air Transfer Function 6.25
- Heat Loss to Surroundings 6.25

6.7 DETAILED CALCULATION PROCEDURE USING CLTD/SCL/CLF METHOD 6.26
- Space Cooling Load due to Heat Gain through Exterior Walls and Roofs and Conductive Gain through Glass 6.26
- Space Cooling Load due to Solar Heat Gain through Fenestration 6.28
- Space Cooling Load due to Heat Gain through Wall Exposed to Unconditioned Space 6.28
- Calculation of Internal Cooling Loads and Infiltration 6.29
- Space Cooling Load of Night Shutdown Operating Mode 6.32

6.8 COOLING COIL LOAD 6.32
- Basics 6.32
- Fan Power 6.33
- Duct Heat Gain 6.33
- Temperature of Plenum Air and Ventilation Load 6.34

6.9 COOLING LOAD CALCULATION BY FINITE DIFFERENCE METHOD 6.34
- Finite Difference Method 6.34
- Simplifying Assumptions 6.36
- Heat and Moisture Transfer at Interior Nodes 6.36
- Heat and Moisture Transfer at Surface Nodes 6.37
- Space Air Temperature and Cooling Loads 6.38

6.10 HEATING LOAD 6.39
- Basic Principles 6.39
- Transmission Loss 6.39
- Adjacent Unheated Spaces 6.40
- Latent Heat Loss and Heat Loss from Products 6.41
- Infiltration 6.41
- Setback of Night Shutdown Operation 6.41

6.11 LOAD CALCULATION SOFTWARE 6.42
- Introduction 6.42
- TRACE 600—Structure and Basics 6.42
- TRACE 600 Input—Load Methodology 6.43
- TRACE 600 Input—Job 6.44
- TRACE 600 Input—External Loads 6.45
- TRACE 600 Input—Schedules 6.45
- TRACE 600 Input—Internal Loads 6.46
- TRACE 600 Input—Partition and Shading Devices 6.47
- TRACE 600—Minimum Input Requirements, Run, and Outputs 6.47

REFERENCES 6.49

6.1 SPACE LOAD CHARACTERISTICS

Space, Room, and Zone

Space indicates either a volume or a site without a partition or a partitioned room or group of rooms. A *room* is an enclosed or partitioned space that is usually treated as a single load. A conditioned room often has an individual control system. A *zone* is a space, or several rooms, or units of space having some sort of coincident loads or similar operating characteristics. A zone may or may not be an enclosed space, or it may consist of many partitioned rooms. It could be a conditioned space or a space that is not air conditioned. A conditioned zone is always equipped with an individual control system. A control zone is the basic unit of control.

Convective and Radiative Heat

Whether heat enters the conditioned space from an external source or is released to the space from an internal source, the instantaneous heat gains of the conditioned space can be classified into two categories: convective heat and radiative heat, as shown in Fig. 6.1. When solar radiation strikes the outer surface of a concrete slab, most of its radiative heat is absorbed by the slab; only a small portion is reflected. After absorption, the outer surface temperature of the slab increases. If the slab and the conditioned space are in thermal equilibrium originally, there is then a convective heat and radiative heat transfer from the surface of the slab to the space air and other surfaces. Meanwhile, heat transfer due to conduction takes place from the surface to the inner part of the slab. Heat is then stored inside the slab. The stored heat is released to the space air when the surface temperature falls below the temperature of the inner part of the slab.

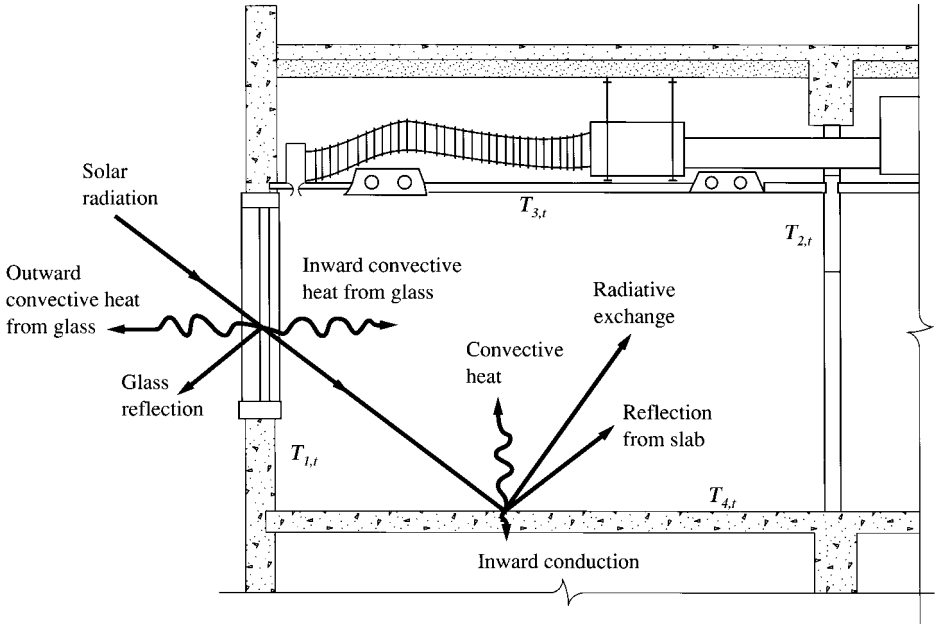


FIGURE 6.1 Convective and radiant heat in a conditioned space and the temperatures of the interior surfaces.

To maintain the preset space air temperature, the heat that has been convected or released to the conditioned space should be removed from the space instantaneously.

Space and Equipment Loads

The sensible and latent heat transfer between the space air and the surroundings can be classified as follows:

1. Space heat gain q_e, in Btu/h (W), represents the rate at which heat enters a conditioned space from an external source or is released to the space from an internal source during a given time interval.

2. Space cooling load, often simply called the cooling load Q_{rc}, Btu/h (W), is the rate at which heat must be removed from a conditioned space so as to maintain a constant temperature and acceptable relative humidity. The sensible cooling load is equal to the sum of the convective heat transfer from the surfaces of the building envelope, furnishings, occupants, appliances, and equipment.

3. Space heating load Q_{rh}, Btu/h (W), is the rate at which heat must be added to the conditioned space to maintain a constant temperature and sometimes a specified relative humidity.

4. Space heat extraction rate Q_{ex}, Btu/h (W), is the rate at which heat is actually removed from the conditioned space by the air system. The sensible heat extraction rate is equal to the sensible cooling load only when the space air temperature remains constant.

5. Coil load Q_c, Btu/h (W), is the rate of heat transfer at the coil. The cooling coil load Q_{cc}, Btu/h (W), is the rate at which heat is removed by the chilled water flowing through the coil or is absorbed by the refrigerant inside the coil.

6. The heating coil load Q_{ch}, Btu/h (W), is the rate at which heat is added to the conditioned air from the hot water, steam, or electric heating elements inside the coil.

7. Refrigerating load Q_{rl}, Btu/h (W), is the rate at which heat is absorbed by the refrigerant at the evaporator. For central hydronic systems, the refrigerating load is the sum of the coil load plus the chilled water piping heat gain, pump power heat gain, and storage tank heat gain. For most water systems in commercial buildings, the water piping and pump power heat gain is only about 5 to 10 percent of the coil load. In an air conditioning system using DX coil(s), the refrigerating load is equal to the DX coil load.

The instantaneous sensible heat gain of a conditioned space is not equal to the instantaneous sensible cooling load because of storage of part of the radiative heat inside the building structures. Such phenomenon results in a smaller instantaneous cooling load than that of the heat gain when it is at its maximum value during a diurnal cycle, as shown in Fig. 6.2.

If the space relative humidity is maintained at an approximately constant value, the storage effect of the moisture in the building envelope and furnishings can be ignored. Then the instantaneous space latent heat gain will be the same as the instantaneous space latent cooling load.

Night Shutdown Operating Mode

In many commercial buildings, air systems are often shut down during the nighttime or during unoccupied periods. The operating characteristics of the conditioned space in a 24-h diel cycle can then be divided into three periods, as shown in Fig. 6.3.

Night Shutdown Period. This period commences when the air system is switched off and ends when the air system is switched on again. When the air-handling unit, packaged unit, or terminal unit is turned off during summer in a hot and humid area, the infiltrated air (through the lift, pipe shafts, and window cracks) and any heat transfer from the window glass, external wall, or roofs will cause a sudden increase in indoor temperature T_r of a few degrees Fahrenheit (Celsius). After that,

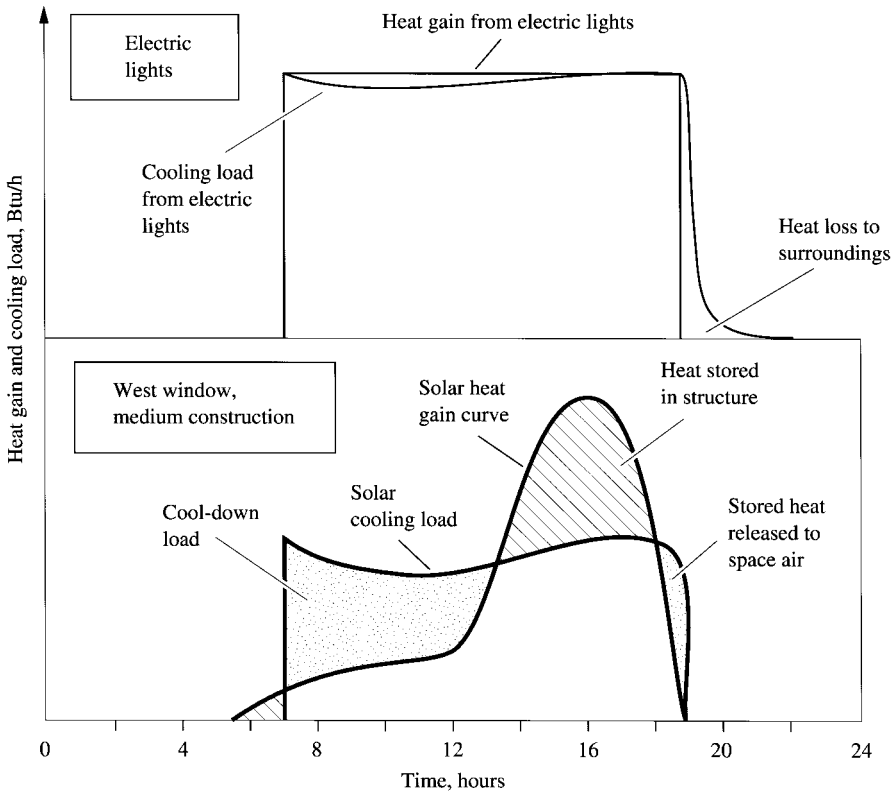


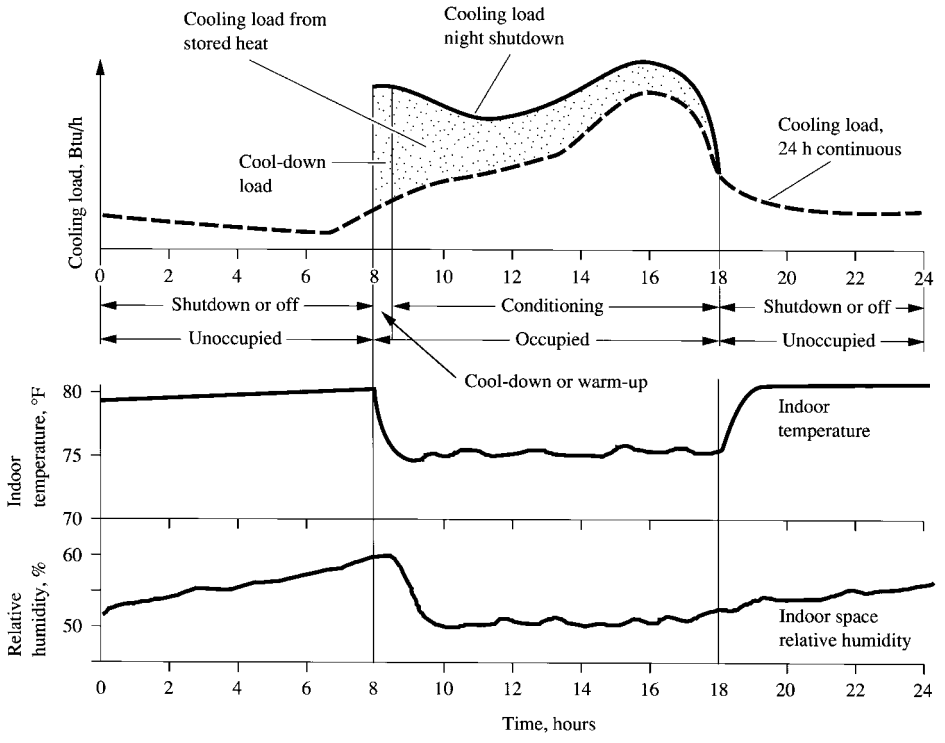
FIGURE 6.2 Solar heat gain and heat gain from electric lights and the corresponding space cooling loads for a night shutdown air system.

T_r will rise farther or drop slightly, depending on the difference between outdoor and indoor temperatures.

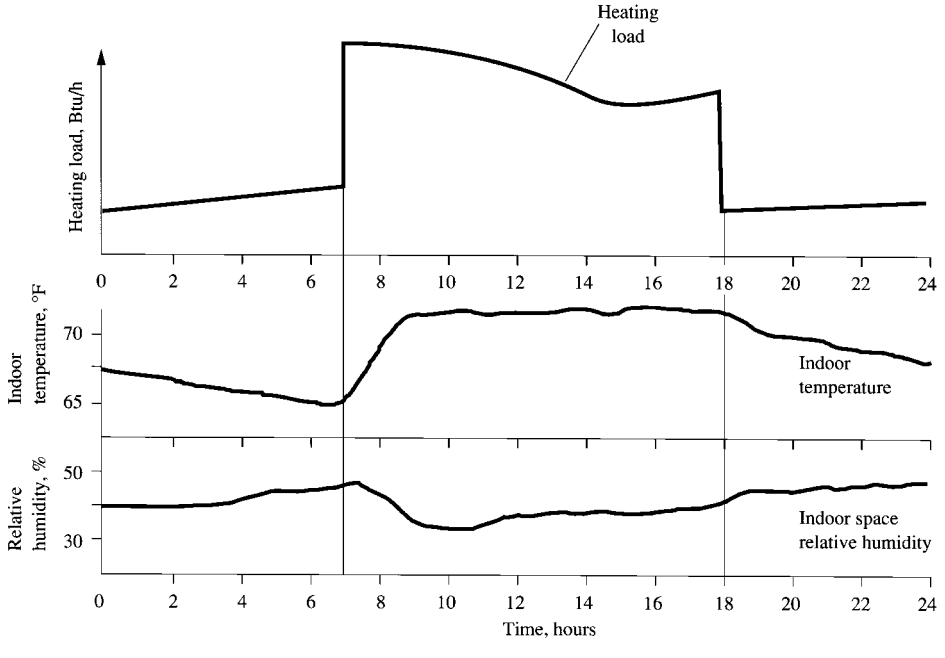
At the same time, the indoor space relative humidity increases gradually because of the infiltration of hot and humid outdoor air and the moisture transfer from the wetted surfaces in the air system. Higher space relative humidity causes a moisture transfer from the space air to the building envelopes and furnishings.

During winter, after the air system is turned off, the indoor temperature drops because of the heat loss through external windows, walls, and roofs, as well as the infiltration of outdoor cold air. Meanwhile, the space relative humidity increases mainly owing to the fall of the indoor space temperature.

Cool-down or Warm-up Period. This period commences when the air system begins to operate and ends when the space temperature or other controlled variables have attained predetermined limits. During summer, the supply of cold and dehumidified air after the air system is turned on causes a sudden drop in space air temperature and relative humidity. Both heat and moisture are transferred from the building envelope to the space air because of the comparatively higher temperature and moisture content of the building envelope. These heat and moisture transfers form the cool-down cooling load. Sometimes, the cool-down load can be the maximum summer design cooling load.



(a)



(b)

FIGURE 6.3 Operating characteristics of an air conditioned space operated at night shutdown mode: (*a*) summer cooling mode (hot and humid area) and (*b*) winter mode.

If refrigeration is used for cooling during the cool-down period, then it usually lasts less than 1 h, depending mainly on the tightness of the building and the differences of the space temperatures and humidity ratios between the shutdown and cool-down periods. If the cool-down of the space air and the building envelope is by means of the free cooling of outdoor air, the cool-down period may last for several hours.

During winter, the supply of warm air during the warm-up period raises the space temperature and lowers the space relative humidity. Because of the higher space temperature, heat transfer from the space air to the colder building envelope and furnishings, and the heat energy required to raise the temperature of the building envelope, forms the warm-up heating load.

Conditioning Period. This period commences when the space air temperature has fallen or risen to a value within the predetermined limits. It ends when the air system is shut down. In summer, cold and dehumidified air is usually supplied to the space to offset the space cooling load and to maintain a required temperature and relative humidity. During winter, warm air is supplied to compensate for heat losses from the conditioned space. Space temperature is often controlled and maintained within predetermined limits by control systems.

In commercial buildings located in areas with a cold winter, the air system may operate in duty-cycling mode during nighttime, unoccupied periods in winter, or intermediate seasons to maintain a night setback temperature. In duty-cycling mode, the fan and heater turn on and off to maintain a desired temperature. Space temperature is set back at night, e.g., to 55 or 60°F (12.8 or 15.6°C), to prevent freezing of water pipes and to produce a comparatively smaller temperature lift during a warm-up period. The operation of the air system in a 24-h diel cycle in winter is then divided into night setback period, warm-up period, and conditioning period. The required load on the heating coil during a morning warm-up period at winter design conditions is usually the winter design heating load.

Influence of Stored Heat

The curve of solar heat gain from a west window is shown in Fig. 6.2, as well as the cooling load curve due to this solar heat gain for a conditioned space operated at night shutdown mode in summer. The difference between the maximum solar heat gain q_{hgm} and the maximum cooling load Q_{clm} during the conditioning period indicates the amount of heat stored inside the building structures. This difference significantly affects the size of air conditioning equipment required. The amount of heat stored depends mainly on the mass of the building envelope (whether it is heavy, medium, or light), the duration of the operating period of the air system within a 24-h cycle, and the characteristics of heat gain, whether radiant heat or convective heat predominates. For solar radiation transmitted through a west window, Q_{clm} may have a magnitude of only 40 to 60 percent of q_{hgm}.

ASHRAE Handbook divides the mass of the building construction into the following three groups:

Heavy construction: approximately 130 lb/ft^2 (634 kg/m^2) floor area

Medium construction: approximately 70 lb/ft^2 (342 kg/m^2) floor area

Light construction: approximately 30 lb/ft^2 (146 kg/m^2) floor area

6.2 COOLING LOAD AND COIL LOAD CALCULATIONS

Components of Cooling Load

Cooling load calculations for air conditioning system design are mainly used to determine the volume flow rate of the air system as well as the coil and refrigeration load of the equipment—to size

the HVAC&R equipment and to provide the inputs to the system for energy use calculations in order to select optimal design alternatives. Cooling load usually can be classified into two categories: external and internal.

External Cooling Loads. These loads are formed because of heat gains in the conditioned space from external sources through the building envelope or building shell and the partition walls. Sources of external loads include the following cooling loads:

1. Heat gain entering from the exterior walls and roofs
2. Solar heat gain transmitted through the fenestrations
3. Conductive heat gain coming through the fenestrations
4. Heat gain entering from the partition walls and interior doors
5. Infiltration of outdoor air into the conditioned space

Internal Cooling Loads. These loads are formed by the release of sensible and latent heat from the heat sources inside the conditioned space. These sources contribute internal cooling loads:

1. People
2. Electric lights
3. Equipment and appliances

If moisture transfers from the building structures and the furnishings are excluded, only infiltrated air, occupants, equipment, and appliances have both sensible and latent cooling loads. The remaining components have only sensible cooling loads.

All sensible heat gains entering the conditioned space represent radiative heat and convective heat except the infiltrated air. As in Sec. 6.1, radiative heat causes heat storage in the building structures, converts part of the heat gain into cooling load, and makes the cooling load calculations more complicated. Latent heat gains are heat gains from moisture transfer from the occupants, equipment, appliances, or infiltrated air. If the storage effect of the moisture is ignored, all release heat to the space air instantaneously and, therefore, they are instantaneous cooling loads.

Components of Cooling Coil Load

If the conductive heat gain from the coil's framework and the support is ignored, the cooling coil load consists of the following components, as shown in Fig. 6.4 by the summer air conditioning cycle *O-m-cc-s-r-rf-m* of a constant volume of supply air, single supply duct, and serving a single zone.

1. Space cooling load Q_{rc}, including sensible and latent load
2. Supply system heat gain q_{ss}, because of the supply fan heat gain q_{sf}, and supply duct heat gain q_{sd}
3. Return system heat gain q_{rs} because of heat gains of recessed electric lights and ceiling plenum q_{rp}, of return duct q_{rd}, and return fan q_{rf}, if any
4. Sensible and latent load because of the outdoor ventilation rates Q_o to meet the requirements of the occupants and others

In Fig. 6.4, the summer air conditioning cycle *O-m-cc-s-r-rf-m* consists of an adiabatic mixing process *O-m-rf*, a cooling and dehumidifying process *m-cc*, a supply system heat gain process *cc-s*, a space conditioning process *r-s*, and a return system heat gain process *r-rf*. Here, *O* indicates the status of outdoor air, *m* the mixture of outdoor air and recirculating air, *cc* the conditioned air

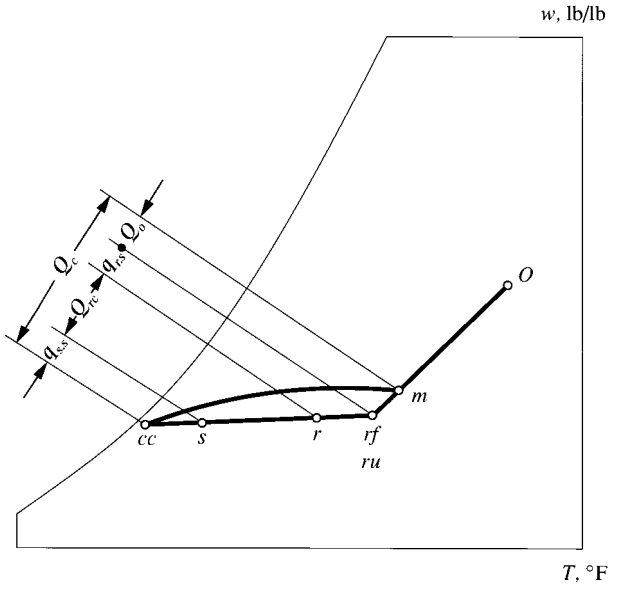


FIGURE 6.4 Difference between cooling load and cooling coil load on psychrometric chart.

leaving the coil, *s* the supply air, *r* the conditioned space, and *rf* the recirculating air. All the cooling loads and heat gains are in Btu/h (W). Usually, both the supply and return system heat gains are sensible loads. These components are absorbed by the supply air and return air and appear as cooling and dehumidifying loads at the cooling coil during summer.

Difference between Cooling Load and Cooling Coil Load

For the same air conditioning cycle shown in Fig. 6.4, note the following:

1. The space cooling load is represented by Q_{rc}, and the cooling coil load is represented by Q_c. Since supply system heat gain q_{ss} and return system heat gain q_{rs} are both instantaneous cooling loads, then

$$Q_c = Q_{rc} + q_{ss} + q_{rs} + Q_o \tag{6.1}$$

   where Q_o = load due to the outdoor ventilation air intake, Btu/h (W).
2. The space cooling load is used to determine the supply volume flow rate $\dot{V}_s$, whereas the coil load is used to determine the size of the cooling coil in an air-handling unit or DX coil in a packaged unit.
3. A cooling load component influences both $\dot{V}_s$ and the size of the cooling coil, whereas a cooling coil load component may not affect $\dot{V}_s$.
4. Infiltration heat gain is an instantaneous cooling load. From Fig. 6.4, it is apparent that the load due to the outdoor ventilaion air Q_o, sometimes called the ventilation load, is a coil load. If Q_o is considered a cooling load, the volume flow rate of the air system will be oversized.

Load Profile

A load profile shows the variation of space load within a certain time period, such as a 24-h operating cycle or an annual operating cycle, as shown in Fig. 6.5. In a load profile, the space load is always plotted against time. For a space cooling load, the magnitude of the curve is positive; for a space heating load, the magnitude is negative. A load profile may be used to illustrate the load variation of an air conditioned space—a room, a zone, a floor, a building, or a project. The shape of a load profile depends on the outdoor climate and, therefore, the latitude, orientation, and structure of the building. It is also affected by the operating characteristics and the variation of the internal loads.

The load duration curve is the plot of number of hours versus the load ratio. The *load ratio* is defined as the ratio of cooling or heating load to the design full load, both in Btu/h, over a certain period. The period may be a day, a week, a month, or a year. The load duration curve is helpful in many operating and energy consumption analyses.

Peak Load and Block Load

The zone peak load is the maximum space cooling load in a load profile of a control zone of the same orientation and similar internal load characteristics calculated according to the daily outdoor dry-bulb and coincident wet-bulb temperature curves containing summer or winter outdoor design conditions. For a zone cooling load with several components, such as solar load through window glass, heat transfer through roofs, or internal load from electric lights, the zone peak load is always

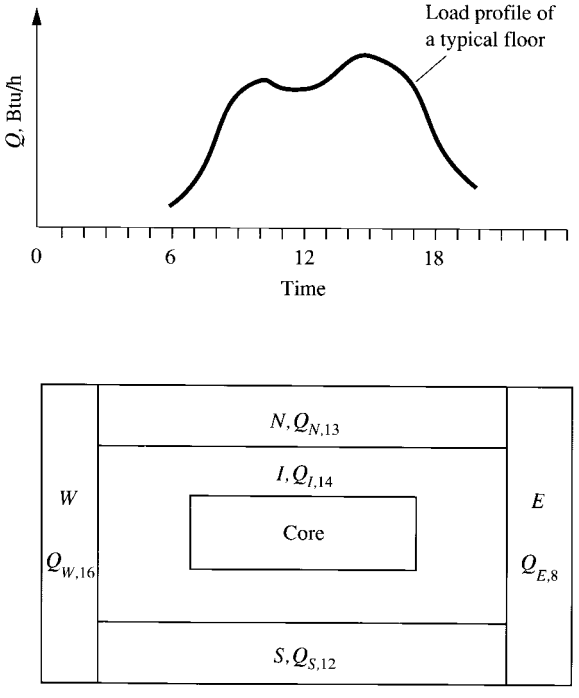


FIGURE 6.5 Load profile, peak load, and block load.

the maximum sum of these zone cooling load components at a given time. The block load is the maximum sum of the zone cooling loads of various load profiles of the control zones of a building floor or a building at the same time. The block load of a space, room, floor, or building is the maximum sum of cooling load components in that space, room, floor, or building at a given time.

For air systems, the supply volume flow rate required for a control zone is calculated based on the zone peak load; and the supply volume flow rate for a specific area, room, floor, or building should be calculated based on the block load (cooling) of this specific area, room, floor, or building. For conditioned space using variable-air-volume systems and space air conditioning systems, the required cooling coil load or refrigeration load can be calculated based on block load of the corresponding specific area that air system serves.

The heating load is usually the peak heating load of a space, room, zone, floor, or building in a load profile. Block load is not needed in heating load calculations because the solar radiation and internal loads are not taken into account. Design heating load depends entirely on indoor-outdoor temperature difference.

Suppose a typical floor of a multistory building has five zones: a north, a south, an east, and a west perimeter zone and an interior zone. For each zone, there is a corresponding daily cooling load profile curve and a zone peak load $Q_{N,13}$, $Q_{s,12}$, $Q_{E,8}$, $Q_{W,16}$, and $Q_{I,14}$. Here, subscripts N, S, E, W, and I indicate north, south, east, west, and interior, respectively; 8, 12, 13, 14, and 16 represent time at 8, 12, 13, 14, and 16 h, respectively.

The block load of this typical floor $Q_{\max,t}$ is not calculated as the sum of the zone peak loads $Q_{N,13}$, $Q_{s,12}$, $Q_{E,8}$, $Q_{W,16}$, and $Q_{I,14}$. Rather the block load is calculated for a specific time. For example, block load $Q_{\max.14}$, Btu/h (W), is the block load at 14 h, i.e.,

$$Q_{\max.14} = Q_{N,14} + Q_{S,14} + Q_{E,14} + Q_{W,14} + Q_{I,14} \tag{6.2}$$

Characteristics of Night Shutdown Operating Mode

Compared with the continuous operating mode, night shutdown operating mode has the following characteristics:

1. A greater peak load, therefore a higher air system and initial cost
2. A higher maximum heat extraction rate
3. A smaller accumulated heat extraction rate over the 24-h operating cycle
4. A lower power consumption for the fans, compressors, and pumps

Moisture Transfer from the Building Structures

In hot and humid climates, the frequency of outdoor wet-bulb temperatures higher than 73°F (22.8°C) exceeds 1750 h during the six warmest consecutive months annually. At the same time, the air infiltrated through the elevator and pipe shafts during shutdown periods often causes a temperature increase of more than 7°F (3.9°C) and a relative humidity increase exceeding 10 percent. These result in a significant increase of the latent load because of the moisture transfer from the building envelope and furnishings to the space air during the cool-down period in summer. Such an increase in the coil's latent load not only necessitates a greater refrigerating capacity, but also lowers the sensible heat ratio of the space conditioning process. A space conditioning process removes heat and moisture from the space by the supply air, or sometimes adds moisture and heat to compensate for space heat losses. In industrial applications where a very low relative humidity is

maintained in the conditioned space, the latent load due to the moisture transfer from the building envelope should be added to the space cooling load calculations during night shutdown operation mode.

6.3 HISTORICAL DEVELOPMENT OF COOLING LOAD CALCULATIONS

In the 1930s, Houghton et al. introduced the analysis of heat transmission through the building envelope and discussed the periodic heat flow characteristics of the building envelope. In 1937, *ASHVE Guide* introduced a systematic method of cooling load calculation involving the division of various load components. In the *ASHVE Guide*, solar radiation factors were introduced and their influence on external walls and roofs was taken into consideration. Both the window crack and number-of-air-changes methods were used to calculate infiltration.

Mackey and Wright first introduced the concept of sol-air temperature in 1944. In the same paper, they recommended a method of approximating the changes in inside surface temperature of walls and roofs due to periodic heat flow caused by solar radiation and outside temperature with a new decrement factor. In 1952, Mackey and Gay analyzed the difference between the instantaneous cooling load and the heat gain owing to radiant heat incident on the surface of the building envelope.

In 1964, Palmatier introduced the term *thermal storage factor* to indicate the ratio between the rate of instantaneous cooling load in the space and rate of heat gain. One year after, Carrier Corporation published a design handbook in which the heat storage factor and equivalent temperature difference (ETD) were used to indicate the ratio of instantaneous cooling load and heat gain because of the heat storage effect of the building structure. This cooling load calculation method was widely used by many designers until the current ASHRAE methods were adopted.

In 1967, ASHRAE suggested a time-averaging (TA) method to allocate the radiant heat over successive periods of 1 to 3 h or 6 to 8 h, depending on the construction of the building structure. Heat gains through walls and roofs are tabulated in total equivalent temperature differentials (TETDs). In the same year, Stephenson and Mitalas recommended the thermal response factor, which includes the heat storage effect for the calculation of cooling load. The thermal response factor evaluates the system response on one side of the structure according to random temperature excitations on the other side of the structure. This concept had been developed and forms the basis of the weighting factor method (WFM) or transfer function method (TFM) in the 1970s. In 1977, ASHRAE introduced a single-step cooling load calculation procedure that uses the cooling load factor (CLF) and cooling load temperature difference (CLTD); these are produced from the simplified TFM.

In 1963, Kusuda and Achenbach used a digital computer to analyze the thermal environment of occupied underground space. The use of computers to design building mechanical systems was first accepted in 1965 when a group of mechanical engineering consultants organized Automated Procedures for Engineers Consultants Inc. (APEC) because of sharing of software and development costs.

Because of the need for computerized load and energy calculations, ASHRAE established a Committee on Energy Consumption in 1965 and renamed the Task Group on Energy Requirements (TGER) for Heating and Cooling in 1967. In the mid-1970s, ASHRAE and the National Bureau of Standards (NBS) published the computerized calculation of heating and cooling loads in energy-estimating programs. In 1980, the U.S. Department of Energy sponsored a computer program for energy estimation and load calculation through hour-by-hour detailed system simulation, called DOE-2, which was published through Los Alamos National Laboratory and Lawrence Berkeley Laboratory. In this program, a custom weighting factor method for various room configurations is used for heating and cooling load calculations. Many computerized thermal load and energy calculating software programs had been developed in the 1980s. Since the 1980s, because of the wide

adoption of personal computers, the use of computer-aided HVAC&R design was rapidly increased and many thermal load and energy analysis programs were developed in this period.

6.4 METHODOLOGY—HEAT BALANCE

The Physical Model

The exact method to calculate the space cooling load is to use heat balance equations to determine the temperature of the interior surfaces of the building structure at time t simultaneously and then to calculate the space sensible cooling load, which is equal to the sum of the convective heat transfer from these surfaces, latent cooling loads, and the cooling load due to infiltrated air at time t.

Consider a typical air conditioned room, as shown in Fig. 6.1. In this room, the building envelope consists of mainly walls, window, ceiling, and floor. There is also heat transfer from occupants, electric lights, and equipment. The heat transfer between various surfaces takes place under the following assumptions:

- Only one-dimensional transient heat flow through the building envelope is considered.
- The room air is perfectly mixed with the supply air so that the resulting room air temperature is uniform.
- The materials of the building envelope are homogeneous. The surface temperature, the surface heat-transfer coefficient, and the absorptivity for each surface are uniform values. Reflectivity is very small and can be ignored.
- The cooling load is calculated based on the mean value of a fixed time interval, such as 1 h.

Heat Balance Equations

Any interior surface of this air conditioned room may receive conduction heat transfer at time t, denoted by $q_{i,t}$, Btu/h (W), from the adjacent building material. Each interior surface receives shortwave solar radiation from the window glass and longwave radiative heat transfer from other interior surfaces and from the surfaces of the lighting fixtures, appliances, equipment, and occupants. Convective heat transfer is also present between these interior surfaces and room air.

The sensible heat balance at the ith surface (for $i = 1, 2, \ldots, m$) at time t can be expressed as follows:

$$q_{i,t} = [h_{ci}(T_{r,t} - T_{i,t}) + \sum_{j=1}^{k} h_{ij}(T_{j,t} - T_{i,t})]A_i + S_{ir,t} + L_{ir,t} + E_{ir,t} + O_{ir,t} \tag{6.3}$$

where h_{ci} = convective heat-transfer coefficient of ith surface, Btu/h · ft^2 · °F (W/m^2 · °C)
h_{ij} = radiative heat-transfer coefficient between interior surface i and j, $i \neq j$, Btu/h · ft^2 · °F (W/m^2 · °C)
$T_{r,t}$ = room air temperature at time t, °F (°C)
$T_{i,t}$ = temperature of ith surface at time t, °F (°C)
$T_{j,t}$ = temperature of jth surface at time t, °F (°C)
A_i = area of ith surface, ft^2 (m^2)
$S_{ir,t}$ = solar radiation transmitted through window glass and absorbed by ith surface at time t, Btu/h (W)
$L_{ir,t}$ = radiative energy from electric lights and absorbed by ith surface at time t, Btu/h (W)
$E_{ir,t}$ = radiative energy from equipment and absorbed by ith surface at time t, Btu/h (W)
$O_{ir,t}$ = radiative energy from occupants and absorbed by ith surface at time t, Btu/h (W)

Because the conductive heat and radiative heat in Eq. (6.3) are all continuous functions of time $f(t)$, the transient conductive heat transfer $q_{i,t}$ either can be solved by a partial differential equation using numerical solutions (finite difference method) or can be solved by means of the conduction transfer function [Eq. (6.10)].

Usually, the room air temperature $T_{r,t}$ is considered as constant, and Eq. (6.3) and the conduction transfer function equation can be solved simultaneously to determine the interior surface temperatures $T_{i,t}$. Then the sensible cooling load at time t, denoted by $Q_{rs,t}$, Btu/h (W), can be calculated as

$$Q_{rs,t} = \sum_{i=1}^{m} h_{ci}(T_{i,t} - T_{r,t})A_i + 60\dot{V}_{if,t}\rho c_{pa}(T_{o,t} - T_{r,t}) + S_{c,t} + L_{c,t} + E_{c,i} + O_{c,t} \tag{6.4}$$

where $\dot{V}_{if,t}$ = volume flow rate of infiltrated air at time t, cfm [m^3/(60 s)]
ρ = air density, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of moist air, Btu/lb · °F (J/kg · °C)
$T_{o,t}$ = temperature of outdoor air at time t, °F(°C)
$S_{c,t}$ = solar heat coming through windows and convected into room air at time t, Btu/h (W)
$L_{c,t}$ = sensible heat from electric lights and convected into room air at time t, Btu/h (W)
$E_{c,t}$ = sensible heat from equipment and convected into room air at time t, Btu/h (W)
$O_{c,t}$ = sensible heat from occupants and convected into room air at time t, Btu/h (W)

Since the latent heat gains convert to latent cooling loads instantaneously, therefore the space latent cooling load at time t, or $Q_{rl,t}$, Btu/h (W), can be calculated as

$$Q_{rl,t} = q_{il,t} + 60\dot{V}_{if}\rho(w_{o,t} - w_{r,t})h_{fg} + E_{l,t} + O_{l,t} \tag{6.5}$$

where $q_{il,t}$ = latent heat from ith interior surface and convected into room air at time t, Btu/h (W)
$w_{o,t}$ = humidity ratio of infiltrated air at time t, lb/lb (kg/kg)
$w_{r,t}$ = humidity ratio of room air at time t, lb/lb (kg/kg)
h_{fg} = latent heat of condensation, Btu/lb (J/kg)
$E_{l,t}$ = latent heat from equipment at time t, Btu/h (W)
$O_{l,t}$ = latent heat from occupants at time t, Btu/h (W)

Then the space cooling load at time t, denoted by $Q_{rc,t}$, Btu/h (W), is

$$Q_{rc,t} = q_{rs,t} + q_{rl,t} \tag{6.6}$$

Heat Balance of Space Air

The conditioned air for this room or space is supplied from the ceiling diffuser at a mass flow rate at time t of $\dot{m}_{s,t} = \dot{V}_{s,t}\rho_s$. Here, $\dot{V}_{s,t}$ indicates the volume flow rate of supply air at time t, cfm (m^3/min), and ρ_s is the density of supply air, lb/ft^3 (kg/m^3). The supply air is then mixed with the space air. The resulting mixture absorbs convective sensible heat and latent heat from various surfaces. At time t, the outdoor air infiltrates into the space at a volume flow rate $\dot{V}_{if,t}$, cfm [m^3/(60 s)]. The sensible heat balance of the space air can be expressed as

$$60\,\dot{V}_{s,t}\rho_s c_{pa}\,(T_{r,t} - T_{s,t})$$

$$= \sum_{i=1}^{m} h_{ci}\,A_i(T_{i,t} - T_{r,t}) + 60\dot{V}_{if,t}\rho c_{pa}\,(T_{o,t} - T_{r,t}) + S_{c,t} + L_{c,t} + E_{c,t} + O_{c,t} \tag{6.7}$$

where $T_{s,t}$ = supply air temperature at time t, °F (°C). If the space air temperature is allowed to float, Eq. (6.3), the transient conductive equation, and Eq. (6.7) must be solved simultaneously.

Characteristics of Heat Balance Method

The heat balance method is more direct and clear in load calculation methodology. Using the heat balance method, the assumption of linear supposition is not required, and the changing of certain parameters, such as the surface convective heat-transfer coefficient, can be modeled as required. If moisture transfer should be included in the cool-down period in nighttime shutdown mode in a location where the outdoor climate is hot and humid in summer, then the heat balance method will give comparatively more accurate results.

However, the heat balance method demands laborious work, more computing time, complicated computer programs, and experienced users. Only expensive mainframe computers could run computer programs adopting the heat balance methodology in the 1970s and early 1980s. The heat balance method is generally used for research and analytical purposes.

6.5 METHODOLOGY—TRANSFER FUNCTION

Basics

The transfer function method or weighting factor method is a simplification of the laborious heat balance method. The wide application of the TFM is due to the user-friendliness of the inputs and outputs of the TFM software and the saving of computing time. In the transfer function method, interior surface temperatures and the space cooling load were first calculated by the exact heat balance method for many representative constructions. The transfer function coefficients (weighting factors) were then calculated which convert the heat gains to cooling loads. Sometimes, transfer function coefficients were also developed through test and experiments.

The room transfer function coefficients (weighting factors) were originally generated based on a typical room configuration of 15 ft × 15 ft with 10-ft (4.5 m × 4.5 m with 3-m) ceiling and one exposure of 30 percent glass in the early 1970s . In the late 1980s, the introduction of 14 influential parameters of zone characteristics by Sowell (1988) enabled the adopted room transfer function coefficients to more closely match the room type to be used. Today, TFM is the most widely adopted computer-aided load calculation method in HVAC&R consulting firms.

Transfer Function and Time Function

The *transfer function K* of an element or a system is the ratio of the Laplace transform of the output Y to the Laplace transform of the input or driving force G, or

$$Y = KG \tag{6.8}$$

When a continuous function of time $f(t)$ is represented at regular intervals Δt and its magnitudes are $f(0), f(\Delta), f(2\Delta), \ldots, f(n\Delta)$, the Laplace transform is given as

$$\varphi(z) = f(0) + f(\Delta)z^{-1} + f(2\Delta)z^{-2} + \cdots + f(n\Delta)z^{-n} \tag{6.9a}$$

where Δ = time interval, h
$z = e^{s\Delta}$

The preceding polynomial in z^{-1} is called the *z transform* of the continuous function $f(t)$.

In Eq. (6.8), Y, K, and G can all be represented in the form of a z transform. Because of the radiative component and the associated heat storage effect, the space sensible cooling load at time t can be related to the sensible heat gains and previous cooling loads in the form of a continuous function of time $f(t)$, which can be expressed as a z transform.

Weighting factors are transfer function coefficients presented in the form of z transform functions. Weighting factors are so called because they are used to weight the importance of current and historical values of heat gain and cooling load on currently calculated cooling loads.

Calculation Procedure

The calculation of space cooling load using the transfer function method consists of two steps. First, heat gains or heat loss from exterior walls, roofs, and floors is calculated using response factors or conduction transfer function coefficients; and the solar and internal heat gains are calculated directly for the scheduled hour. Second, room transfer function coefficients or room weighting factors are used to convert the heat gains to cooling loads, or the heat losses to heating loads.

As described in Sec. 6.2, the sensible infiltration heat gain is the instantaneous sensible cooling load. All latent heat gains are instantaneous latent cooling loads.

The TFM is limited because the cooling loads thus calculated depend on the value of transfer function coefficients as well as the characteristics of the space and how they are varied from those used to generate the transfer function coefficients. In addition, TFM assumed that the total cooling load can be calculated by simply adding the individual components — the superposition principle. However, this assumption can cause some errors.

CLTD/SCL/CLF Method

The cooling load temperature difference (CLTD)/solar cooling load (SCL)/cooling load factor (CLF) method first calculates the sensible cooling load based on the TFM. The result is divided by the U value, shading coefficient, or sensible heat gain to generate the CLTD, SCL, or CLF. Thus, it provides a direct, one-step space cooling load calculation instead of a heat gain–cooling load conversion, a two-step calculation in TFM. Cooling load calculation using the CLTD/SCL/CLF method can be either computer-aided or performed manually for a check or rough estimate. The CLTD/SCL/CLF method is one of the members of the TFM family.

In the CLTD/SCL/CLF method, the CLTD is used to calculate the sensible cooling load for the exterior wall and roofs. Recently, an SCL factor has been added which represents the product of the solar heat gain at that hour and the fraction of heat storage effect due to various types of room construction and floor coverings. CLF is used to calculate internal sensible cooling loads.

The limitations of the TFM are also carried through to the CLTD/SCL/CLF results. Furthermore, the grouping of CLTD/SCL/CLF may cause additional errors.

TETD/TA Method

In the total eqivalent temperature difference (TETD)/time-averaging (TA) method, heat gains of a number of representative exterior wall and roof assemblies q_w, Btu/h (W), are calculated as

$$q_w = AU(\text{TETD}) \tag{6.9b}$$

where A = area of wall or roof, ft^2 (m^2)
U = overall heat-transfer coefficient of wall or roof, Btu/h · ft^2 · °F (W/m^2 · °C)

In Eq. (6.9*b*), TETD, in °F (°C), can be evaluated by:

- Using the conduction transfer function as in TFM to determine q_w. Then it is divided by the U value to generate TETD values.
- Using the following relationship:

$$\text{TETD} = T_{\text{sol},a} - T_r + \lambda(T_{\text{sol},\delta} - T_{\text{sol},a}) \tag{6.9c}$$

where $T_{\text{sol},a}$ = daily average sol-air temperature, °F (°C)
$T_{\text{sol},\delta}$ = sol-air temperature at time lag δ h, °F (°C)
λ = effective decrement factor λ

The internal heat gains and conductive heat gain are calculated in the same manner as in the TFM. The radiant fraction of each of the sensible heat gains is then allocated to a period including the current and successive hours, a total of 1 to 3 h for light construction and 6 to 8 h for heavy construction.

The TETD/TA method is also a member of the TFM family and is developed primarily for manual calculation. TETD/TA is simpler in the conversion of heat gains to cooling loads. However, the time-averaging calculation procedure is subjective—it is more an art than a rigorous scientific method. Also the TETD/TA method inherits the limitations that a TFM possesses if the TFM is used to calculate the TETD.

6.6 DETAILED CALCULATION PROCEDURES FOR TFM

Most of the widely adopted computer software programs for space load calculations are based on the transfer function method, i.e., the weighting factors method. The following are the detailed calculation procedures for TFM.

Conduction Heat Gain through Exterior Walls and Roofs

The sensible heat gain through an exterior wall or a roof using TFM can be calculated by the conduction transfer function. It uses a sol-air temperature at time t, denoted by $T_{\text{sol},t}$, to represent the combined temperature excitation of the outdoor temperature and solar heat on the exterior surface of an exterior wall or roof, and a constant indoor temperature T_r. Conduction-transfer function coefficients are calculated based on an outdoor surface heat-transfer coefficient $h_o = 3.0$ Btu/h·ft²·°F (17 W/m²·°C), and an indoor surface heat-transfer coefficient $h_i = 1.46$ Btu/h·ft²·°F (8.3 W/m²·°C).

The external heat gain through an exterior wall or roof at time t, denoted by $q_{e,t}$, Btu/h (W), can be calculated as

$$q_{e,t} = \left(\sum_{n=0} b_n T_{sol,t-n\Delta} - \sum_{n=1} d_n \frac{q_{e,t-n\Delta}}{A} - T_r \sum_{n=0} c_n \right) A \tag{6.10}$$

where t = time, h
Δ = time interval, h
n = summation index of number of terms
$T_{\text{sol},t-n\Delta}$ = sol-air temperature at time $t - n\Delta$, °F (°C)
$q_{e,t-n\Delta}$ = conduction heat gain at time $t - n\Delta$, Btu/h (W)
b_n, c_n, d_n = conduction-transfer function coefficients; refer to *ASHRAE Handbook* for details
A = interior surface area of wall or roof, ft² (m²)

Harris and McQuiston (1988) provide the conduction-transfer function coefficients for 41 representive wall assemblies and 42 roof assemblies with variations in components, insulation, and mass.

Heat Gain through Ceilings, Floors, and Interior Partition Walls

If the temperature of the adjacent space T_{ad}, °F (°C), is constant, or the variation of T_{ad} is small, the sensible heat gain due to the heat transfer from the adjacent space through the ceiling, floor, or interior partition wall at time t, denoted by $q_{p,t}$, Btu/h (W), can be calculated as

$$q_{p,t} = UA(T_{\text{ad}} - T_r) \tag{6.11}$$

where U = overall heat-transfer coefficient of ceiling, floor, or partition wall, $\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ ($\text{W/m}^2 \cdot {}^\circ\text{C}$)
A = surface area of ceiling, floor, or partition wall, ft^2 (m^2)

If the temperature of the adjacent space $T_{\text{ad},t}$ varies with time, then the sensible heat gain transferring through the ceiling, floor, or partition wall at time $q_{p,t}$, Btu/h (W), can be calculated from Eq. (6.10), except $q_{e,t}$ should be replaced by $q_{p,t}$, $q_{e,t-n\Delta}$ replaced by $q_{p,t-n\Delta}$, and $T_{\text{sol},t}$ replaced by $T_{\text{ad},t}$. Here $q_{p,t-n\Delta}$ indicates the sensible heat gain of the ceiling, floor, or partition wall at time $t - n\Delta$, Btu/h (W); and $T_{\text{ad},t}$ indicates the adjacent space temperature, °F (°C).

Solar Heat Gain and Conductive Heat Gain through Window Glass

Although the inward heat flow from the solar radiation absorbed by the window glass and the heat flow due to the outdoor and indoor temperature difference are actually combined, it is simpler and acceptably accurate to separate this composite heat gain into solar heat gain and conductive heat gain; both are sensible heat gains.

Solar heat gain $q_{\text{so},t}$, Btu/h (W), is given as

$$q_{\text{so},t} = A_{s,t}\,(\text{SC})(\text{SHGF}_t) + A_{\text{sh},t}\,(\text{SC})(\text{SHGF}_{\text{sh},t}) \tag{6.12}$$

where $A_{s,t}$ = sunlit area of window glass at time t, ft^2 (m^2)
$A_{\text{sh},t}$ = shaded area of window glass at time t, ft^2 (m^2)
SC = shading coefficient
SHGF_t = solar heat gain factor at time t considering orientation, latitude, month, and hour, $\text{Btu/h} \cdot \text{ft}^2$ (W/m^2)
$\text{SHGF}_{\text{sh},t}$ = solar heat gain factor for shaded area at time t, considering latitude, month, and hour, $\text{Btu/h} \cdot \text{ft}^2$ (W/m^2)

Usually, the SHGF_t of the vertical surface facing north orientation is taken as $\text{SHGF}_{\text{sh},t}$. Conductive heat gain due to the outdoor-indoor temperature difference $q_{\text{win},t}$, Btu/h (W), is given as

$$q_{\text{win},t} = U_{\text{win}} A_{\text{win}}\,(T_{o,t} - T_r) \tag{6.13}$$

where U_{win} = overall heat-transfer coefficient of window including glass and frame (Table 3.7), $\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ ($\text{W/m}^2 \cdot {}^\circ\text{C}$)
A_{win} = gross area of window including glass and frame, ft^2 (m^2)
$T_{o,t}$ = outdoor temperature at time t considering month, hour, and location, °F (°C)

Internal Heat Gain

The internal sensible heat gain consists of the sensible heat gain from people, from electric lights, and from equipment and appliances.

People. Human beings release both sensible heat and latent heat to the conditioned space. The radiative portion of the sensible heat gain is about 70 percent when the indoor environment of the conditioned space is maintained within the comfort zone. The space sensible heat gain for occupants staying in a conditioned space at time t, denoted by $q_{\text{sp},t}$, Btu/h (W), can be calculated as

$$q_{\text{sp},t} = N_{p,t}\,(\text{SHG}_p) \tag{6.14}$$

where $N_{p,t}$ = number of occupants in conditioned space at time t
SHG_p = sensible heat gain of each person, Btu/h (W)

Space latent heat gain for occupants staying in a conditioned space at time t, denoted by $q_{lp,t}$, Btu/h (W), is given as

$$q_{lp,t} = N_{p,t}(\text{LHG}_p) \tag{6.15}$$

where LHG_p = latent heat gain of each person, Btu/h (W). Table 6.1 lists the heat gain from occupants in conditioned space, as abridged from *ASHRAE Handbook 1989*, *Fundamentals*. In Table 6.1, total heat is the sum of sensible and latent heat. The adjusted heat is based on a normally distributed percentage of men, women, and children among the occupants.

Lighting. The sensible heat gains from the electric lights depend on the types of installation, as follows:

Inside Conditioned Space. For electric lights installed inside the conditioned space, such as light fixtures hung below the ceiling, the sensible heat gain released from the electric lights, the emitting element, and light fixtures $q_{s,l}$ is equal to the sensible heat released to the conditioned space $q_{es,l}$, Btu/h (W); both depend mainly on the criteria of illumination and the type and efficiency of electric lights and can be calculated as

$$\begin{aligned} q_{s,l} &= 3.413\, W_{\text{lamp}}\, F_{usl} F_{\text{al}} \\ &= 3.413\, W_A A_{\text{fl}} \end{aligned} \tag{6.16}$$

where W_{lamp} = rated input of electric lights, W
W_A = wattage per ft^2 of floor area, W/ft^2 (W/m^2)
F_{usl} = ratio of wattage in use to installation wattage

In Eq. (6.16), F_{al} indicates an allowance factor for light fixtures, such as Ballast losses. For rapid-start 40-W fluorescent fixtures, F_{al} varies from 1.18 to 1.3 with a recommended value of 1.2 (*ASHRAE Handbook 1993*, *Fundamentals*).

Recess-Mounted Fixtures Using Return Plenum. For situations in which electric lights are recess-mounted on the ceiling and the ceiling plenum is used as a return plenum, the fraction of the

TABLE 6.1 Rates of Heat Gain from Occupants of Conditioned Spaces*

Degree of activity	Typical application	Total heat of adults, male, Btu/h	Total heat adjusted,† Btu/h	Sensible heat, Btu/h	Latent heat, Btu/h
Seated at theater	Theater — matinee	390	330	225	105
Seated at theater	Theater — evening	390	350	245	105
Seated, very light work	Offices, hotels, apartments	450	400	245	155
Moderately active office work	Offices, hotels, apartments	475	450	250	200
Standing, light work; walking	Department store, retail store	550	450	250	200
Walking; standing	Drugstore, bank	550	500	250	250
Light bench work	Factory	800	750	275	475
Moderate dancing	Dance hall	900	850	305	545
Walking 3 m/h; light machine work	Factory	1000	1000	375	625
Heavy work	Factory	1500	1450	580	870
Heavy machine work; lifting	Factory	1600	1600	635	965
Athletics	Gymnasium	2000	1800	710	1090

*Tabulated values are based on 75°F room dry-bulb temperature. For 80°F room dry-bulb temperature, the total heat remains the same, but the sensible heat values should be decreased by approximately 20 percent and the latent heat values increased accordingly. All values are rounded to nearest 5 Btu/h.

†Adjusted heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male, and that the gain from a child is 75 percent of that for an adult male.

***Sources*:** Adapted with permission from *ASHRAE Handbook 1989, Fundamentals*.

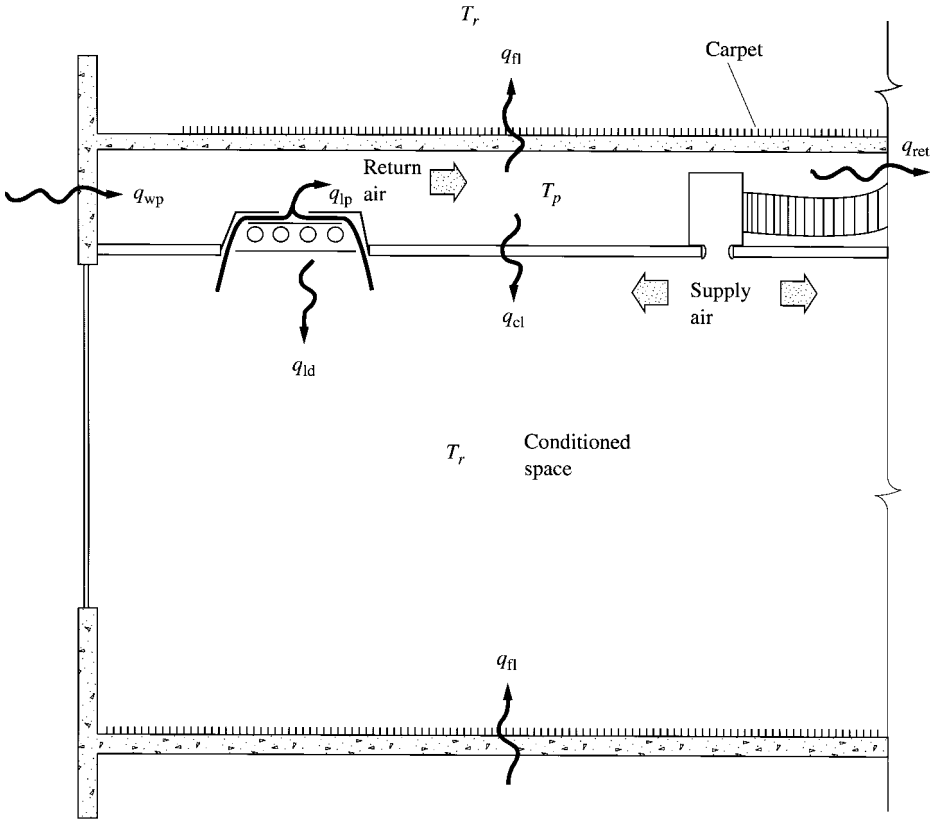


FIGURE 6.6 Heat released from a recess-mounted ventilated lighting fixture.

sensible heat gain of electric lights that enters the conditioned space $q_{es,l}$, Btu/h (W), is closely related to the type of lighting fixture, the ceiling plenum, and the return system.

If the ceiling plenum is used as a return plenum in a multistory building, as shown in Fig. 6.6, the heat released from recessed fluorescent light fixtures and the heat transfer between the outdoor air, plenum air, return air, and space air are as follows:

- Radiative and convective heat transfer from the lighting fixture downward directly into the conditioned space q_{ld}, Btu/h (W), which is calculated as

$$q_{ld} = q_{s,l} - q_{lp} = (1 - F_{lp})\, q_{s,l} \tag{6.17}$$

where q_{lp} = heat released by electric lights to return air, Btu/h (W)
F_{lp} = fraction of heat released from light fixture to plenum air

The fraction that enters conditioned space q_{ld} depends on the volume flow rate of the return air flowing through the lighting fixture and the type of fixture. Its magnitude should be obtained from the lighting fixture manufacturer. In Fig. 6.7 is shown the relationship between F_{lp} and the intensity of volume flow rate of return air $\dot{V}_r/A_{fl}$ if the lighting fixture is ventilated. Usually, F_{lp} varies between 0.4 and 0.6 for a ventilated fixture in a return air plenum. For unventilated lighting fixtures recess-mounted in a return air plenum, F_{lp} varies between 0.15 and 0.5.

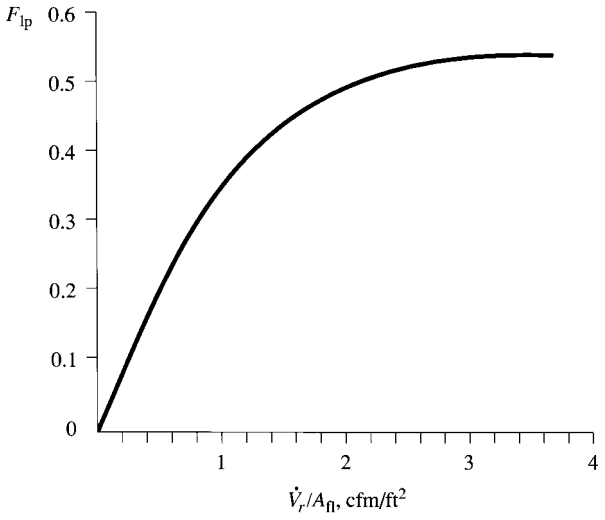


FIGURE 6.7 Relationship between F_{lp} and the intensity of return air $\dot{V}_r / A_{fl}$.

- Heat carried away by return air from the ceiling plenum q_{ret}, Btu/h (W), which is calculated as

$$q_{ret} = 60\, \dot{V}_r \rho_r c_{pa} (T_p - T_r) \tag{6.18}$$

where $\dot{V}_r$ = volume flow rate of return air, cfm [m^3/(60 s)]
ρ_r = density of return air, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of return air, Btu/lb · °F (J/kg · °C)
T_p, T_r = temperatures of plenum air and space air, °F (°C)

- Heat transfer from the plenum air to the conditioned space through the suspended ceiling q_{cl} and heat transfer from the plenum air to the conditioned space through the composite floor q_{fl}, both in Btu/h (W), can be calculated as

$$q_{cl} = U_{cl} A_{cl} (T_p - T_r) \tag{6.19a}$$

$$q_{fl} = U_{fl} A_{fl} (T_p - T_r) \tag{6.19b}$$

where U_{cl}, U_{fl} = overall heat-transfer coefficient of ceiling and composite floor, Btu/h · ft^2 · °F (W/m^2 · °C)
A_{cl}, A_{fl} = area of ceiling and composite floor, ft^2 (m^2)

- Heat transfer between the outdoor air and the plenum air through the exterior wall of the ceiling plenum $q_{wp,t}$, Btu/h (W), can be calculated by conductive transfer function as shown in Eq. (6.10). Because $q_{wp,t}$ is comparatively small, it is often ignored.
- For a multistory building where all the floors are air conditioned, heat gain from the electric lights that enters the conditioned space, heat to space, $q_{es,l}$ is given as

$$q_{es,l} = q_{ld} + q_{cl} + q_{fl} \tag{6.20}$$

Of this, about 50 percent is radiative, and the rest is convective. Heat released from the electric lights to the return air including radiative transfer upward, heat to plenum, q_{lp}, Btu/h (W), is

given as

$$q_{lp} = q_{\text{ret}} + q_{\text{cl}} + q_{\text{fl}} \tag{6.21}$$

In a return air plenum, precise calculation of the plenum air temperature T_p is rather complicated. A simplified method is to assume a steady-state heat transfer between the plenum air and the conditioned space air and to use a cooling load temperature difference CLTD_{wp} (CLTD_{wp} can be found from Table 6.2) to calculate the heat transfer through the exterior wall of the plenum. Then, based on the heat balance at the plenum air,

$$q_{lp} + U_{wp}A_{wp}\,(\text{CLTD}_{wp}) = (60\,\dot{V}_r\rho_r c_{pa} + U_{\text{cl}}A_{\text{cl}} + U_{\text{fl}}\,A_{\text{fl}})\,(T_p - T_r) \tag{6.22}$$

where U_{wp} = overall heat-transfer coefficient of exterior wall, Btu/h · ft^2 · °F (W/m^2 · °C)
A_{wp} = area of exterior wall, ft^2 (m^2)
CLTD_{wp} = cooling load temperature difference of exterior wall of ceiling plenum, °F (°C)

Plenum air temperature T_p can thus be determined.

Surface-Mounted Fixtures under Ceiling. If the lighting fixtures are surface-mounted under the suspended ceiling, then the fraction of heat gain downward that enters the conditioned space $q_{es,l}$, Btu/h (W), can be calculated as

$$q_{es,l} = F_{es,l}\,q_{s,l} \tag{6.23}$$

where $F_{es,l}$ = fraction of sensible heat gain entering the conditioned space for surface-mounted lighting fixtures, which varies from approximately 0.8 to 0.95.

Equipment and Appliances. With equipment and appliances, all energy inputs are converted to heat energy, e.g., in the motor windings, combustion chamber, rubbing surfaces, even at components where mechanical work is performed. A portion of heat released may be exhausted locally by a mechanical ventilation system.

In many industrial applications, the space sensible heat gain due to the machine load when a motor is located inside the conditioned space $q_{s,e}$, Btu/h (W), can be calculated as

$$q_{s,e} = 2546P_{\text{hp}}\,F_{\text{load}}\,F_{\text{use}}\,\frac{1 - F_{\text{exh}}}{\eta_{\text{mo}}} \tag{6.24}$$

where P_{hp} = rated horsepower of machine, hp
F_{load} = load factor indicating ratio of actual power required to rated power
F_{use} = use factor indicating ratio of actually used equipment and appliance to total installed
F_{exh} = heat removal factor due to mechanical exhaust system
η_{mo} = motor efficiency

In Table 6.3 are listed the efficiencies of motors from a fraction of a horsepower to 250 hp (187 kW). The bigger the motor, the higher the motor efficiency. A high-efficiency motor is often cost-effective. The Energy Policy Act of 1992 (EPACT), after a phase-in period of 5 years which ended on October 24 1997, requires that all covered motors meet increased minimum efficiency levels. If the motor is located outside the conditioned space, then in Eq. (6.24), $\eta_{\text{mo}} = 1$.

In many types of equipment and appliances installed with exhaust hoods in the conditioned space, energy input and F_{exh} are better determined according to the actual performance of similar projects. For example, for food preparation appliances equipped with exhaust hoods, only radiation from the surface of the appliances, which as a fraction of the total heat input is between 0.1 and 0.5, should be counted; or F_{exh} is between 0.9 and 0.5. If there is no mechanical exhaust system, $F_{\text{exh}} = 0$.

TABLE 6.2 CLTD for Calculating Sensible Cooling Loads from Sunlit Walls of North Latitude, °F

	Solar time, h																								Hours of maximum	Minimum	Maximum	Difference in
Facing	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	CLTD	CLTD	CLTD	CLTD
Group C walls: typical, outside 1-in. stucco, 2-in. insulation (5.7lb/ft^3), 4-in. concrete, 0.75-in. plaster or gypsum, inside U = 0.119 Btu/h·ft^2·°F; mass, 63 lb/ft^2																												
N	15	14	13	12	11	10	9	8	8	7	7	8	8	9	10	12	13	14	15	16	17	17	17	16	22	7	17	10
NE	19	17	16	14	13	11	10	10	11	13	15	17	19	20	21	22	22	23	23	23	23	22	21	20	20	10	23	13
E	22	21	19	17	15	14	12	12	14	16	19	22	25	27	29	29	30	30	30	29	28	27	26	24	18	12	30	18
SE	22	21	19	17	15	14	12	12	12	13	16	19	22	24	26	28	29	29	29	29	28	27	26	24	19	12	29	17
S	21	19	18	16	15	13	12	10	9	9	9	10	11	14	17	20	22	24	25	26	25	25	24	22	20	9	26	17
SW	29	17	25	22	20	18	16	15	13	12	11	11	11	13	15	18	22	26	29	32	33	33	32	31	22	11	33	22
W	31	29	27	25	22	20	18	16	14	13	12	12	12	13	14	16	20	24	29	32	35	35	35	33	22	12	35	23
NW	25	23	21	20	18	16	14	13	11	10	10	10	10	11	12	13	15	18	22	25	27	27	27	26	22	10	27	17
Group D walls: typical, outside 1-in. stucco, 4-in. concrete, 1- or 2-in. insulation (2 lb/ft^3), 0.75-in. plaster or gypsum, inside U = 0.119–0.20 Btu/h·ft^2·°F; mass, 63 lb/ft^2																												
N	15	13	12	10	9	7	6	6	6	6	6	7	8	10	12	13	15	17	18	19	19	19	18	16	21	6	19	13
NE	17	15	13	11	10	8	7	8	10	14	17	20	22	23	23	24	24	25	25	24	23	22	20	18	19	7	25	18
E	19	17	15	13	11	9	8	9	12	17	22	27	30	32	33	33	32	32	31	30	28	26	24	22	16	8	33	25
SE	20	17	15	13	11	10	8	8	10	13	17	22	26	29	31	32	32	32	31	30	28	26	24	22	17	8	32	24
S	19	17	15	13	11	9	8	7	6	6	7	9	12	16	20	24	27	29	29	29	27	26	24	22	19	6	29	23
SW	28	25	22	19	16	14	12	10	9	8	8	8	10	12	16	21	27	32	36	38	38	37	34	31	21	8	38	30
W	31	27	24	21	18	15	13	11	10	9	9	9	10	11	14	18	24	30	36	40	41	40	38	34	21	9	41	32
NW	25	22	19	17	14	12	10	9	8	7	7	8	9	10	12	14	18	22	27	31	32	32	30	27	22	7	32	25
Group G walls: typical, outside 1-in. stucco, airspace; 1-, 2-, or 3-in. insulation (2 lb/ft^3); 0.75-in. plaster or gypsum, inside U = 0.081–0.78 Btu/h·ft^2·°F; mass, 16 lb/ft^2																												
N	3	2	1	0	−1	2	7	8	9	12	15	18	21	23	24	24	25	26	22	15	11	9	7	5	18	−1	26	27
NE	3	2	1	0	−1	9	27	36	39	35	30	26	26	27	27	26	25	22	18	14	11	9	7	5	9	−1	39	40
E	4	2	1	0	−1	11	31	47	54	55	50	40	33	31	30	29	27	24	19	15	12	10	8	6	10	−1	55	56
SE	4	2	1	0	−1	5	18	32	42	49	51	48	42	36	32	30	27	24	19	15	12	10	8	6	11	−1	51	52
S	4	2	1	0	−1	0	1	5	12	22	31	39	45	46	43	37	31	25	20	15	12	10	8	5	14	−1	46	47
SW	5	4	3	1	0	0	2	5	8	12	16	26	38	50	59	63	61	52	37	24	17	13	10	8	16	0	63	63
W	6	5	3	2	1	1	2	5	8	11	15	19	27	41	56	67	72	67	48	29	20	15	11	8	17	1	72	71
NW	5	3	2	1	0	0	2	5	8	11	15	18	21	27	37	47	55	55	41	25	17	13	10	7	18	0	55	55

Direct applications and adjustments are stated in the text.

***Source*:** Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

TABLE 6.3 Heat Gain from Typical Electric Motors

Motor nameplate or rated horsepower	Nominal rpm	Full-load motor effciency, percent	Location of motor and driven equipment with respect to conditioned space or airstream: Motor in, driven equipment in, Btu/h	Motor out, driven equipment in, Btu/h	Motor in, driven equipment out, Btu/h
Motor type: shaded pole					
0.05	1,500	35	360	130	240
0.125	1,500	35	900	320	590
Motor type: split-phase					
0.25	1,750	54	1,180	640	540
0.33	1,750	56	1,500	840	660
0.50	1,750	60	2,120	1,270	850
Motor type: three-phase					
0.75	1,750	72	2,650	1,900	740
1	1,750	75	3,390	2,550	850
1	1,750	77	4,960	3,820	1,140
2	1,750	79	6,440	5,090	1,350
3	1,750	81	9,430	7,640	1,790
5	1,750	82	15,500	12,700	2,790
7.5	1,750	84	22,700	19,100	3,640
10	1,750	85	29,900	24,500	4,490
15	1,750	86	44,400	38,200	6,210
20	1,750	87	58,500	50,900	7,610
25	1,750	88	72,300	63,600	8,680
30	1,750	89	85,700	76,300	9,440
40	1,750	89	114,000	102,000	12,600
50	1,750	89	143,000	127,000	15,700
60	1,750	89	172,000	153,000	18,900
75	1,750	90	212,000	191,000	21,200
100	1,750	90	283,000	255,000	28,300
150	1,750	91	420,000	382,000	37,800
200	1,750	91	569,000	509,000	50,300
250	1,750	91	699,000	636,000	62,900

For motors operating more than 750 h/year, it is usually cost-effective to use a high-efficiency motor. Typical efficiency ratings are as follows: 5 hp, 89.5%; 10 hp, 91.0%; 50 hp, 94.1%; 100 hp, 95.1%; 200 hp, 96.2%.

***Source*:** Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

Because of the widespread installation of microcomputers, display terminals, printers, copiers, calculators, and facsimile machines, the heat released from machines in office buildings has increased considerably in recent years. Komor (1997) listed the measured office plug loads from 17 U.S. office buildings in the 1990s which varied from 0.44 to 1.11 W/ft^2. Precise office equipment heat gain can be calculated from manufacturers' data using a load factor F_{load} between 0.3 and 0.5.

The latent heat gain from the equipment and appliances $q_{l,e}$, Btu/h (W), can be calculated from the mass flow rate of water vapor evaporated $\dot{m}_w$, lb/h (kg/h), as

$$q_{l,e} = 1075\, \dot{m}_w \tag{6.25}$$

Infiltration

Infiltration is the uncontrolled inward flow of outdoor air through cracks and openings in the building envelope due to the pressure difference across the envelope. The pressure difference may be caused by any of the following:

1. Wind pressure
2. Stack effect due to the outdoor and indoor temperature difference
3. Mechanical ventilation

In summer, for low-rise commercial buildings that have their exterior windows well sealed, and if a positive pressure is maintained in the conditioned space when the air system is operating, normally the infiltration can be considered zero.

For high-rise buildings, infiltration should be considered and calculated in both summer and winter. Infiltration is discussed again in Sec. 20.4.

As soon as the volume flow rate of infiltrated air $\dot{V}_{\text{inf}}$, cfm (m^3/min), is determined, the space sensible heat gain from infiltration q_{inf}, Btu/h (W), can be calculated as

$$q_{s,\text{inf}} = 60\dot{V}_{\text{inf}}\,\rho_o c_{pa}\,(T_o - T_r) \tag{6.26}$$

where ρ_o = density of outdoor air, lb/ft^3 (kg/m^3). The space latent heat gain from infiltration $q_{l,\text{inf}}$, Btu/h (W), can be calculated as

$$q_{l,\text{inf}} = 60\dot{V}_{\text{inf}}\rho_o\,(w_o - w_r)h_{fg,32} \tag{6.27}$$

where w_o, w_r = humidity ratio of outdoor and space air, respectively, lb/lb (kg/kg)
$h_{fg,32}$ = latent heat of vaporization at 32°F, Btu/lb (J/kg)

Cooling Load Conversion Using Room Transfer Function

The conversion of space sensible heat gains $q_{rs,t}$, Btu/h (W), having radiative only or radiative and convective components to space sensible cooling loads $Q_{rs,t}$, Btu/h (W), using the transfer function method and room transfer function coefficients can be expressed as follows:

$$Q_{rs,t} = \sum_{i=1}(v_0 q_{s,t} + v_1 q_{s,t-\Delta} + v_2 q_{s,t-2\Delta} + \cdots) - (w_1\,Q_{rs,t-\Delta} + w_2\,Q_{rs,t-2\Delta} + \cdots) \tag{6.28}$$

where i = number of heat gain components in same group
Δ = time interval
$t-n\Delta$ = time at $t-n\Delta$

Here v_0, v_1, v_2, . . ., w_1, w_2, . . . are the coefficients of the room transfer function; refer to the *ASHRAE Handbook* for details. Their relationship can be expressed from Eqs. (6.8) and (6.9) as

$$K(z) = \frac{v_0 + v_1 z^{-1} + v_2 z^{-2} + \cdots}{1 + w_1 z^{-1} + w_2 z^{-2} + \cdots} \tag{6.29}$$

The magnitude of the coefficients depends on the duration of the time interval, fraction of the radiative component, and heat storage capacity because of the 14 influential parameters of zone characteristics, such as zone geometry, height, exterior wall construction, interior shade, furniture, zone location, glass percentage, and type of partition, midfloor, slab, ceiling, roof, and floor.

Space Cooling Load Calculation

The types of conversion of space heat gains to space cooling loads can be grouped into the following two categories:

1. Space sensible cooling loads $Q_{rs,t}$ that are only a fraction of the space sensible heat gain $q_{rs,t}$. These kinds of sensible heat gains have both radiative and convective components, and it is difficult to separate the convective component from the radiative component, such as sensible heat gains from exterior walls and roofs, and solar heat gains through windows. Equation (6.28) will be used to convert these types of sensible heat gains $q_{rs,t}$ to cooling loads $Q_{rs,t}$.
2. Space heat gains $q_{\text{in},t}$ are instantaneous space cooling loads $Q_{\text{in},t}$, both in Btu/h (W), or

$$Q_{\text{in},t} = q_{\text{in},t} \tag{6.30}$$

This category includes all latent heat gains q_{rl}, convective sensible heat gains, infiltration sensible heat gain $q_{\text{inf},s}$, and sensible heat gains whose convective component can be separated from the radiative component, such as air-to-air heat gain from windows, lights, and people.

The space cooling load $Q_{rc,t}$, Btu/h (W), is their sum, or

$$Q_{rc,t} = Q_{rs,t} + Q_{\text{in},t} + Q_{l,t} = Q_{s,t} + Q_{l,t} \tag{6.31}$$

where $Q_{s,t}$, $Q_{l,t}$ = space sensible cooling load and latent load, respectively, Btu/h (W).

As mandated in ASHRAE/IESNA Standard 90.1-1999, *Optimum Start Controls*, pickup loads either for cooling or heating depend on the difference between space temperature and occupied set point and the amount of time prior to scheduled occupancy, and is often determined by computer software.

Heat Extraction Rate and Space Air Transfer Function

In the calculation of the conduction heat gain through the exterior wall and roofs by using the conduction transfer function [Eq. (6.10)] and then converting to sensible cooling load from the room transfer function, the space temperature T_r is found to be a constant. Most direct-digital control (DDC) zone control systems are now adopting proportional-integral control mode. When the air system and the DDC system are in operation, T_r is a constant once it has achieved a stable condition.

However, for many air systems operated at nighttime shutdown mode, T_r will drift away from the set point during the shutdown period. The space air transfer function relates the heat extraction rate at time t, denoted by $Q_{\text{ex},t}$, Btu/h (W), to the space air temperature at time t, denoted by $T_{r,t}$, °F (°C), as

$$\sum_{i=0}^{1} p_i \left(Q_{\text{ex},t-\Delta} - Q_{rc,t-i\Delta}\right) = \sum_{i=0}^{2} g_i \left(T_{r,\text{con}} - \mathrm{T}_{r,t-i\Delta}\right) \tag{6.32}$$

where p_i, g_i = space air transfer function coefficients, refer to *ASHRAE Handbook* for details
$Q_{\text{ex},t-\Delta}$ = heat extraction rate at time $t-\Delta$, Btu/h (W)
$Q_{rc,t-i\Delta}$ = calculated space cooling load at time $t-i\Delta$, Btu/h (W)
$T_{r,\text{con}}$ = assumed constant space air temperature, °F (°C)

Heat Loss to Surroundings

There is always a radiant heat loss from the outer surface of the building to the sky vault without clouds because atmospheric temperature is lower at high altitudes, as described in Sec. 3.12. In many locations in the United States, there are also radiant heat losses to the surroundings at

nighttime in summer due to the lower ground temperature. *ASHRAE Handbook 1993*, *Fundamentals*, recommends a simplified calculation procedure so that the sum of sensible cooling loads from heat gain by conduction through exterior walls and roofs, from conduction and solar heat gain through windows, and from heat gain through interior partitions, ceilings, and floors at time t, denoted by $Q_{\text{ex},t}$, Btu/h (W), plus the sensible cooling loads from the radiant component of internal heat gain at time t, denoted by $Q_{\text{in},t}$, Btu/h (W), will be multiplied by a factor F_{sur} to take into account these heat losses. The corrected sensible cooling load at time t, denoted by $Q_{rs,\text{cor},t}$, Btu/h (W), is

$$Q_{rs,\text{cor},t} = F_{\text{sur}} (Q_{\text{ex},t} + Q_{\text{in},t}) \tag{6.33}$$

The factor of heat loss to surroundings F_{sur} can be calculated as

$$F_{\text{sur}} = 1 - 0.02 \left(\frac{1}{L_{\text{ex}}}\right)(U_{\text{roof}} A_{\text{roof}} + U_{\text{wall}} A_{\text{wall}} + U_{\text{win}} A_{\text{win}} + U_{\text{part}} A_{\text{part}}) \tag{6.34}$$

where L_{ex} = length of space exterior wall, ft
U = overall heat-transfer coefficient (subscript win for window and part for interior partitions), Btu/h · ft^2 · °F (W/m^2 · °C)
A = area of component of building envelope, ft^2 (m^2)

6.7 DETAILED CALCULATION PROCEDURE USING CLTD/SCL/CLF METHOD

The following sections describe in greater detail the principles of the CLTD/SCL/CLF method. They also provide a simple, manual cooling load calculation procedure in case an estimate or a rough check of computer-aided cooling load calculation is required.

Space Cooling Load due to Heat Gain through Exterior Walls and Roofs and Conductive Gain through Glass

If the ratio of sensible cooling load to sensible heat gain through the exterior wall or roof $Q_{rs,w}/q_{rs,w}$ = CLTD/ΔT, for an exterior sunlit wall or roof under the combined effect of solar radiation and the outdoor-indoor temperature difference, the one-step calculation of space sensible cooling load $Q_{rs,w}$, Btu/h (W), can be performed as

$$Q_{rs,w} = UA \text{ (CLTD)} \tag{6.35}$$

where U = overall heat-transfer coefficient of exterior wall or roof, Btu/h · ft^2 · °F (W/m^2 · °C)
A = area of exterior wall, roof, or window including frame or sash, ft^2 (m^2)
CLTD = cooling load temperature difference, °F (°C)

The CLTD values recommended by ASHRAE for calculating the space sensible cooling load through flat roofs and sunlit walls of various constructions are listed in Tables 6.2 and 6.4, respectively. The values in both Table 6.2 and Table 6.4 were calculated under the following conditions. In other words, these are the conditions under which the listed data can be applied directly without adjustments:

- Indoor temperature of 78°F (25.6°C)
- Outdoor maximum temperature of 95°F (35°C) with an outdoor daily mean of 85°F (29.4°C) and an outdoor daily range of 21°F (11.7°C)
- Solar radiation of 40° north latitude on July 21
- Roof with dark, flat surface

TABLE 6.4 CLTD for Calculating Sensible Cooling Loads from Flat Roofs, °F

Description of construction	Weight, lb/ft^2	*U* value, Btu/h·ft^2·°F	Solar time, h 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	Hours of maximum CLTD	Minimum CLTD	Maximum CLTD	Difference in CLTD
Without suspended ceiling																														
2.5-in. wood with 2-in. insulation	13	0.093	30	26	23	19	16	13	10	9	8	9	13	17	23	29	36	41	46	49	51	50	47	43	39	35	19	8	51	43
4-in. wood with 2-in. insulation	18	0.078	38	36	33	30	28	25	22	20	18	17	16	17	18	21	24	28	32	36	39	41	43	43	42	40	22	16	43	27
With suspended ceiling																														
1-in. wood with 2-in. insulation	10	0.083	25	20	16	13	10	7	5	5	7	12	18	25	33	41	48	53	57	57	56	52	46	40	34	29	18	5	57	52
2.5-in. wood with 1-in. insulation	15	0.096	34	31	29	26	23	21	18	16	15	15	16	18	21	25	30	34	38	41	43	44	44	42	40	37	21	15	44	29
8-in. lightweight concrete	33	0.093	39	36	33	29	26	23	20	18	15	14	14	15	17	20	25	29	34	38	42	45	46	45	44	42	21	14	46	32
4-in. heavyweight concrete with 2-in. insulation	54	0.090	30	29	27	26	24	22	21	20	20	21	22	24	27	29	32	34	36	38	38	38	37	36	34	33	19	20	38	18
2.5-in. wood with 2-in. insulation	15	0.072	35	33	30	28	26	24	22	20	18	18	18	20	22	25	28	32	35	38	40	41	41	40	39	37	21	18	41	23
Roof terrace system	77	0.082	30	29	28	27	26	25	24	23	22	22	22	23	23	25	26	28	29	31	32	33	33	33	33	32	22	33	22	11
6-in. heavyweight concrete with 2-in. insulation	77	0.088	29	28	27	26	25	24	23	22	21	21	22	23	25	26	28	30	32	33	34	34	34	33	32	31	20	21	34	13
4-in. wood with 2-in. insulation	19 20	0.082 0.064	35	34	33	32	31	29	27	26	24	23	22	21	22	22	24	25	27	30	32	34	35	36	37	36	23	21	37	16

Conditions of direct application and adjustments are stated in the text.

Source: Abridged with permission from *ASHRAE Handbook 1989, Fundamentals*.

- Outer surface R value $R_o = 0.333$ h·ft²·°F/Btu (0.06 m²·°C/W) and inner surface $R_i = 0.685$ h·ft²·°F/Btu (0.123 m²·°C/W)
- No attic fans or return air ducts in suspended ceiling space

The following formula can be used for adjustments when the conditions are different from those mentioned:

$$\text{CLTD}_{\text{corr}} = \text{CLTD} + 78 - T_r + T_{om} - 85 \tag{6.36}$$

where $78 - T_r$ = indoor temperature correction; T_r is indoor temperature, °F (°C)
$T_{om} - 85$ = outdoor temperature correction; T_{om} is outdoor mean daily temperature, °F (°C)

In Table 6.4, the roof terrace system includes the following:

- 2-in. (50-mm) lightweight concrete
- Airspace
- 2-in. (50-mm) insulation [5.7 lb/ft³ (91.2 kg/m³)]
- 0.5-in. (13-mm) slag or stone
- 0.375-in. (9.5-mm) felt and membrane

For a pitched roof with a suspended ceiling, the area A in Eq. (6.35) should be the area of the suspended ceiling. If a pitched roof has no suspended ceiling under it, then the actual CLTD is slightly higher than the value listed in Table 6.4 because a greater area is exposed to the outdoor air.

Space Cooling Load due to Solar Heat Gain through Fenestration

The space sensible cooling load from solar heat gain transmitted through the window facing a specific direction $Q_{rs,s}$, Btu/h (W), can be calculated as follows:

$$\begin{aligned} Q_{rs,s} &= Q_{\text{sun}} + Q_{\text{sh}} \\ &= A_s \times \text{SCL}_s \times \text{SC} + A_{\text{sh}} \times \text{SCL}_{\text{sh}} \times \text{SC} \end{aligned} \tag{6.37}$$

where Q_{sun} = space cooling load from solar heat gain through sunlit area of window glass, Btu/h (W)
Q_{sh} = space cooling load from solar heat gain through shaded area of window glass, Btu/h (W)
A_s, A_{sh} = sunlit and shaded area, ft² (m²)
SC = shading coefficient
SCL_s = solar cooling load for sunlit glass facing specific direction, Btu/h·ft² (W/m²)
SCL_{sh} = solar cooling load for shaded area as if glass is facing north, Btu/h·ft² (W/m²)

Zone types for use with SCL_s and SCL_{sh} tables, single-story building, are listed in Table 6.5. July solar cooling loads for sunlit glass 40° north latitude are listed in Table 6.6. Refer to *ASHRAE Handbook 1993*, *Fundamentals*, for zone types for multistory buildings and other details.

In Eq. (6.37), at a given time, $A_s + A_{\text{sh}} = A_{\text{glass}}$. Here A_{glass} indicates the glass area of the window, in ft² (m²). In the northern hemisphere for a conditioned space with southern orientation, the maximum SCL_s may occur in December instead of June.

Space Cooling Load due to Heat Gain through Wall Exposed to Unconditioned Space

When a conditioned space is adjacent to an area that is unconditioned, and if the temperature fluctuation in this area is ignored, then the sensible heat gain q_{rs} transferred through the partitioned walls and interior windows and doors, in Btu/h (W), can be calculated as

TABLE 6.5 Zone Types for Use with SCL and CLF Tables, Single-Story Building

Zone parameters				Zone type			Error band	
No. walls	Floor covering	Partition type	Inside shade	Glass solar	People and equipment	Lights	Plus	Minus
1 or 2	Carpet	Gypsum	*	A	B	B	9	2
1 or 2	Carpet	Concrete block	*	B	C	C	9	0
1 or 2	Vinyl	Gypsum	Full	B	C	C	9	0
1 or 2	Vinyl	Gypsum	Half to none	C	C	C	16	0
1 or 2	Vinyl	Concrete block	Full	C	D	D	8	0
1 or 2	Vinyl	Concrete block	Half to none	D	D	D	10	6
3	Carpet	Gypsum	*	A	B	B	9	2
3	Carpet	Concrete block	Full	A	B	B	9	2
3	Carpet	Concrete block	Half to none	B	B	B	9	0
3	Vinyl	Gypsum	Full	B	C	C	9	0
3	Vinyl	Gypsum	Half to none	C	C	C	16	0
3	Vinyl	Concrete block	Full	B	C	C	9	0
3	Vinyl	Concrete block	Half to none	C	C	C	16	0
4	Carpet	Gypsum	*	A	B	B	6	3
4	Vinyl	Gypsum	Full	B	C	C	11	6
4	Vinyl	Gypsum	Half to none	C	C	C	19	−1

A total of 14 zone parameters are defined. Those not shown in this were selected to achieve the minimum error band shown in the right-hand column for solar cooling load.

*The effect of inside shade is negligible in this case.

***Source*:** Adapted from *ASHRAE Handbook 1997, Fundamentals*. Reprinted with permission.

$$q_{rs} = AU(T_{un} - T_r) \tag{6.38}$$

where T_{un}, T_r = daily mean air temperature of adjacent area that is unconditioned and space temperature, respectively, °F (°C). For an adjacent area that is not air conditioned and has heat sources inside, such as a kitchen or boiler room, T_{un} may be 15°F (8.3°C) higher than the outdoor temperature. For an adjacent area without any heat source other than electric lights, $T_{un} - T_r$ may be between 3 and 8°F (1.7 and 4.4°C). For floors built directly on the ground or located above a basement that is neither ventilated nor conditioned, the space sensible cooling load from the heat gain through the floor can often be ignored.

Calculation of Internal Cooling Loads and Infiltration

The calculation of the internal heat gains of people, lights, equipment, and appliances $q_{int,s}$ and heat gains of infiltration q_{inf}, all in Btu/h (W), using the CLTD/SCL/CLF method is the same as that which uses TFM. The sensible internal heat gain that contains the radiative component is then multiplied by a cooling load factor CLF_{int} to convert to space sensible cooling load $Q_{int,s}$, Btu/h (W), and can be calculated as

$$Q_{int,s} = q_{int,s}\,(CLF_{int}) \tag{6.39}$$

In conditioned spaces in which an air system is operated at nighttime shutdown mode, CLF_{int} is equal to 1 during the occupied period when the air system is operating. Refer to *ASHRAE Handbook 1993*, *Fundamentals*, for CLF_{int} when the air system is operated 24 h continuously.

Since internal latent heat gains $q_{int,l}$ are instantaneous internal latent cooling loads $Q_{int,l}$ both in Btu/h (W),

$$Q_{int,l} = q_{int,l} \tag{6.40}$$

TABLE 6.6 July Solar Cooling Load for Sunlit Glass, 40° North Latitude

Glass face	Solar time, h																								
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Zone type A																								
N	0	0	0	0	1	25	27	28	32	35	38	40	40	39	36	31	31	36	12	6	3	1	1	0
NE	0	0	0	0	2	85	129	134	112	75	55	48	44	40	37	32	26	18	7	3	2	1	0	0
E	0	0	0	0	2	93	157	185	183	154	106	67	53	45	39	33	26	18	7	3	2	1	0	0
SE	0	0	0	0	1	47	95	131	150	150	131	97	63	49	41	34	27	18	7	3	2	1	0	0
S	0	0	0	0	0	9	17	25	41	64	85	97	96	84	63	42	31	20	8	4	2	1	0	0
SW	0	0	0	0	0	9	17	24	30	35	39	64	101	133	151	152	133	93	35	17	8	4	2	1
W	1	0	0	0	0	9	17	24	30	35	38	40	65	114	158	187	192	156	57	27	13	6	3	2
NW	1	0	0	0	0	9	17	24	30	35	38	40	40	50	84	121	143	130	46	22	11	5	3	1
Horiz.	0	0	0	0	0	24	69	120	169	211	241	257	259	245	217	176	125	70	29	14	7	3	2	1
Zone type B																								
N	2	2	1	1	1	22	23	24	28	32	35	37	38	37	35	32	31	35	16	10	7	5	4	3
NE	2	1	1	1	2	73	109	116	101	73	58	52	48	45	41	36	30	23	13	9	6	5	3	3
E	2	2	1	1	2	80	133	159	162	143	105	74	63	55	48	41	34	25	15	10	7	5	4	3
SE	2	2	1	1	1	40	81	112	131	134	122	96	69	58	49	42	35	26	15	10	8	6	4	3
S	2	2	1	1	1	8	15	21	36	56	74	86	87	79	63	46	37	27	16	11	8	6	4	3
SW	6	5	4	3	2	9	16	22	27	31	36	58	89	117	135	138	126	94	46	31	21	15	11	8
W	8	6	5	4	3	9	16	22	27	31	35	37	59	101	139	166	173	147	66	43	30	21	15	11
NW	6	5	4	3	2	9	16	22	27	31	34	37	37	46	76	108	128	119	51	33	22	16	11	8
Horiz.	8	6	5	4	3	22	60	104	147	185	214	233	239	232	212	180	137	90	53	37	27	19	14	11
Zone type C																								
N	5	5	4	4	4	24	23	24	27	30	33	34	35	34	32	29	29	34	14	10	8	7	6	6
NE	7	6	6	5	6	75	106	107	88	61	49	47	45	43	40	36	31	25	16	13	11	10	9	8
E	9	8	8	7	8	83	130	148	145	124	89	62	56	52	47	43	37	30	20	17	15	13	12	11
SE	9	8	7	6	6	45	82	107	121	121	107	82	59	51	47	42	36	29	19	16	14	13	11	10
S	7	7	6	5	5	12	18	23	36	54	70	79	79	70	54	40	33	26	16	13	12	10	9	8
SW	14	12	11	10	9	15	21	26	29	33	36	57	86	110	124	125	111	80	37	28	23	20	17	15
W	17	15	13	12	11	17	22	27	31	34	36	37	59	98	132	153	156	128	50	35	28	24	21	19
NW	12	11	10	9	8	14	20	25	29	32	34	36	36	44	73	102	118	107	39	26	21	17	15	13
Horiz.	24	21	19	17	16	34	68	107	144	175	199	212	215	207	189	160	123	83	53	44	38	34	30	27
Zone type D																								
N	8	7	6	6	6	21	21	21	24	27	29	31	32	31	30	28	29	32	17	14	12	11	10	9
NE	11	10	9	8	9	63	87	90	77	58	49	48	46	44	42	39	35	29	22	19	17	15	14	12
E	15	13	12	11	11	70	107	123	124	110	85	65	60	57	53	48	43	37	29	25	22	20	18	16
SE	14	13	11	10	10	39	68	90	102	104	95	78	60	55	51	47	42	35	27	24	21	19	17	16
S	11	10	9	8	7	12	17	21	32	46	59	67	69	63	52	41	36	30	22	19	17	15	14	12
SW	21	19	17	15	14	18	22	25	28	31	34	51	74	94	106	109	100	78	45	37	33	29	26	23
W	25	23	20	18	17	21	24	28	30	33	34	35	53	84	112	130	135	116	57	46	39	35	31	28
NW	18	16	15	13	12	17	21	24	27	30	32	33	34	41	64	87	101	94	42	33	29	25	22	20
Horiz.	37	33	30	27	24	38	64	95	124	150	171	185	191	188	176	156	128	96	72	63	56	50	45	41

Notes:
1. Values are in Btu/h·ft^2.
2. Apply data directly to standard double-strength glass with no inside shade.
3. Data apply to 21st day of July.
4. For other types of glass and internal shade, use shading coefficients as multiplier. For externally shaded glass, use north orientation.

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Internal load density (ILD), W/ft^2 (W/m^2), indicates the total internal heat gains of people, lights, and equipment, and it can be calculated as

$$\text{ILD} = \frac{\text{SHG}_p + \text{LHG}_p}{3.413\, A_{\text{fl}}} + W_{A,l} + W_{A,e} \tag{6.41}$$

where SHG_p, LHG_p = sensible and latent heat gains for occupants, respectively, Btu/h (W)
A_{fl} = floor area, ft^2 (m^2)
$W_{A,l}$, $W_{A,e}$ = lighting and equipment power density, respectively, W/ft^2 (W/m^2)

Both infiltration sensible heat gain $q_{\text{inf},s}$ and infiltration latent heat gain $q_{\text{inf},l}$, in Btu/h (W), are instantaneous space cooling loads; they also can be expressed as

$$Q_{\text{inf},s} = q_{\text{inf},s} \qquad Q_{\text{inf},l} = q_{\text{inf},l} \tag{6.42}$$

where $Q_{\text{inf},s}$, $Q_{\text{inf},l}$ = infiltration sensible and latent cooling loads, respectively, Btu/h (W).

Example 6.1. A return air plenum in a typical floor of a multistory building has the following construction and operating characteristics:

Wattage of electric lights	1.5 W/ft^2 (16.1 W/m^2)
Return air volume flow rate	11,800 cfm (334 m^3/min)
Density of return air ρ_r	0.073 lb/ft^3 (1.168 kg/m^3)
Fraction of heat to plenum F_{lp}	0.5
U values: exterior wall of plenum	0.2 Btu/h·ft^2·°F (1.136 W/m^2·°C)
suspended ceiling	0.32 Btu/h·ft^2·°F (1.817 W/m^2·°C)
floor	0.21 Btu/h·ft^2·°F (1.192 W/m^2·°C)
Area of ceiling and floor	11,900 ft^2 (1106 m^2)
Area of exterior wall of plenum	1920 ft^2 (178 m^2)
CLF of electric lights	1.0
CLTD of exterior wall of plenum	24°F (13.3°C)
Space temperature	75°F (23.9°C)

Determine the return air plenum temperature and the space cooling load from the electric lights when they are recess-mounted on the ceiling and the ceiling plenum is used as a return plenum.

Solution. From Eq. (6.17),

$$q_{lp} = F_{lp} q_{s,l} = 0.5 \times 1.5 \times 11{,}900 \times 3.413 = 30{,}461 \text{ Btu/h (W)}$$

Since $60\dot{V}_r \rho_r c_{pa} = 60 \times 11{,}800 \times 0.073 \times 0.243 = 12{,}559$ Btu/h·°F, and given Eq. (6.22), the temperature of return air in the return plenum is

$$T_p = T_r + \frac{q_{lp} + U_{wp} A_{wp} \text{CLTD}_{wp}}{60\dot{V}_r \rho_r c_{pa} + U_{\text{cl}} A_{\text{cl}} + U_{\text{fl}} A_{\text{fl}}}$$

$$= 75 + \frac{30{,}717 + 0.2 \times 1920 \times 24}{12{,}559 + 0.32 \times 11{,}900 + 0.21 \times 11{,}900}$$

$$= 75 + \frac{39{,}933}{18{,}866} = 77.12°\text{F (25.07°C)}$$

From Eqs. (6.19*a*) and (6.19*b*), heat transfer from the plenum air to conditioned space through the ceiling and floor is calculated as

$$q_{cl} = U_{cl} A_{cl} (T_p - T_r)$$
$$= 0.32 \times 11{,}900(77.1 - 75) = 7997 \text{ Btu/h (W)}$$

$$q_{\text{fl}} = U_{\text{fl}} A_{\text{fl}} (T_p - T_r)$$
$$= 0.21 \times 11{,}900(77.1 - 75) = 5248 \text{ Btu/h (W)}$$

Then, from Eq. (6.20), heat to space is calculated as

$$q_{es,l} = q_{ld} + q_{\text{cl}} + q_{\text{fl}}$$
$$= 30{,}461 + 7997 + 5248 = 43{,}706 \text{ Btu/h (W)}$$

From Eq. (6.39), the space sensible cooling load from electric lights is

$$Q_{rs,l} = \text{CLF}_{\text{int}}\, q_{es,l} = 1.0 \times 43{,}692 = 43{,}692 \text{ Btu/h (12,802 W)}$$

Space Cooling Load of Night Shutdown Operating Mode

In commercial buildings, air systems are often operated in night shutdown mode during unoccupied hours in summer. The accumulated stored heat because of the external heat gains increases the space cooling load during cool-down and conditioned periods. On the other hand, heat losses to the surroundings due to the radiant heat exchange between the outer surface of the building and the sky vault and surroundings decrease the accumulated stored heat as well as the space cooling load, although the radiative heat losses to the sky vault and surroundings partly compensate the stored heat released to the space. However, overlooking the remaining stored heat released to the space during cool-down and conditioned periods in summer is the limitation of the CLTD/SCL/CLF method compared to TFM, especially when peak load occurs during the cool-down period. As in TFM, an increase of up to 10 percent is recommended by ASHRAE/IES Standard 90.1-1989 for pickup load during the cool-down period for air systems operated at nighttime shutdown mode.

6.8 COOLING COIL LOAD

Basics

Based on the principle of heat balance, the cooling coil load is given as

Total enthalpy of entering air = total enthalpy of leaving air
+ cooling coil load (or cooling capacity) + heat energy of condensate

Since the heat energy of the condensate is small and can be ignored, the cooling coil load Q_{cc}, Btu/h (W), can be calculated as

$$Q_{cc} = 60\dot{V}_s \rho_s (h_{ae} - h_{cc}) \tag{6.43}$$

where $\dot{V}_s$ = volume flow rate of supply air, cfm [$m^3/(60\ s)$]
ρ_s = density of supply air, lb/ft^3 (kg/m^3)
h_{ae}, h_{cc} = enthalpy of entering air and conditioned air leaving coil, respectively, Btu/lb (J/kg)

Of this, the sensible cooling coil load Q_{cs}, Btu/h (W), is

$$Q_{cs} = 60\dot{V}_s \rho_s c_{pa} (T_{ae} - T_{cc}) \tag{6.44}$$

where T_{ae}, T_{cc} = temperature of entering air and conditioned air leaving coil, respectively, °F (°C). And the latent coil load Q_{cl}, Btu/h (W), is

$$Q_{cl} = 60\dot{V}_s\rho_s(w_{ae} - w_{cc})h_{fg,32} \tag{6.45}$$

where w_{ae}, w_{cc} = humidity ratio of entering air and conditioned air leaving coil, respectively, lb/lb (kg/kg). Also,

$$Q_{cc} = Q_{cs} + Q_{cl} \tag{6.46}$$

From Fig. 6.4, alternatively, the sensible cooling coil load can be calculated as

$$Q_{cs} = Q_{rs} + q_{s,s} + q_{r,s} + Q_{o,s} \tag{6.47}$$

where $q_{s,s}$, $q_{r,s}$ = supply and return system heat gain (as mentioned in preceding section, both are instantaneous cooling loads), Btu/h (W)
$Q_{o,s}$ = sensible load from outdoor air intake, Btu/h (W)

And the latent coil load can be calculated as

$$Q_{cl} = Q_{rl} + Q_{o,l} \tag{6.48}$$

where $Q_{o,l}$ = latent load from outdoor air intake, Btu/h (W). The supply system heat gain consists of mainly the supply fan power heat gain q_{sf} and supply duct heat gain q_{sd}; and the return system heat gain comprises the return fan power heat gain q_{rf}, return duct heat gain q_{rd}, and ceiling plenum heat gain q_{rp}, all in Btu/h (W).

Fan Power

In the air duct system, the temperature increase from the heat released to the airstream because of frictional and dynamic losses is nearly compensated by the expansion of air from the pressure drop of the airstream. Therefore, it is usually assumed that there is no significant temperature increase from frictional and dynamic losses when air flows through an air duct system.

Fan power input is almost entirely converted to heat energy within the fan. If the fan motor is located in the supply or return airstream, the temperature increase across the supply (or return fan) ΔT_f, °F (°C), can be calculated as

$$\Delta T_f = \frac{0.37\Delta p_t}{\eta_f \eta_m} \tag{6.49}$$

where Δp_t = fan total pressure, in. WC
η_f, η_m = total efficiency of fan and efficiency of motor

If the motor is located outside the airstream, then, in Eq. (6.49), $\eta_m = 1$.

The Δp_t of the return fan for a central hydronic air conditioning system in commercial buildings is usually 0.25 to 0.5 of the Δp_t of the supply fan. Therefore, the temperature increase of the return fan is far smaller than that of the supply fan. The temperature increase of the relief fan or exhaust fan affects only the relief or exhaust airstream. It is not a part of the supply and return system heat gain. A relief fan is used to relieve excess space pressure when 100 percent outdoor air is flowing through the supply fan for free cooling.

Duct Heat Gain

Duct heat gain is the heat transfer caused by the temperature difference between the ambient air and the air flowing inside the air duct. Duct heat gain is affected by this temperature difference, the

thickness of the duct insulation layer, air volume flow rate, and the size and shape of the duct. Detailed calculations are presented in Secs. 17.4 and 20.16. A rough estimate of the temperature increase of the supply air for an insulated duct is as follows:

Supply air velocity	Air temperature rise
$<$ 2000 fpm (10 m/s)	1°F/100 ft (0.6°C/30 m) duct length
$\geq$ 2000 fpm (10 m/s)	0.75°F/100 ft (0.45°C/30 m) duct length

Temperature of Plenum Air and Ventilation Load

For a ceiling plenum using a return plenum, the plenum air temperature can be calculated from Eq. (6.22). The temperature increase of the plenum air, caused by the heat released from the electric lights ($T_p - T_r$), is affected by their power input, type of lighting fixture, return air volume flow rate, and construction of the ceiling plenum. The temperature increase of plenum air $T_p - T_r$ is usually between 1 and 3°F (0.6 and 1.7°C).

From Eqs. (6.26) and (6.27), the sensible and latent loads $Q_{o,s}$ and $Q_{o,l}$, Btu/h (W), which are attributable to the outdoor air intake, can be similarly calculated, except $\dot{V}_{\text{inf}}$ in Eqs. (6.26) and (6.27) should be replaced by the volume flow rate of outdoor air $\dot{V}_o$, cfm (m^3/min).

System heat gains are mainly due to convective heat transfer from the surfaces. For simplification, they are considered instantaneous cooling coil loads.

6.9 COOLING LOAD CALCULATION BY FINITE DIFFERENCE METHOD

When both heat and moisture transfer from the surface of the walls, ceiling and carpet or floors should be considered in the space cooling load calculation during the cool-down period in summer in a location where the outdoor climate is hot and humid, the finite difference method might be the best choice.

Finite Difference Method

Because of the rapid increase in the use of microcomputers in the HVAC&R calculations, it is now possible to use a finite difference method, a numerical approach, to solve transient simultaneous heat- and moisture-transfer problems in heating and cooling load calculations and energy estimations. The finite difference method divides the building structures into a number of sections. A fictitious node i is located at the center of each section or on the surface, as shown in Fig. 6.8. An energy balance or a mass balance at each node at selected time intervals results in a set of algebraic equations that can be employed to determine the temperature and moisture for each node in terms of neighboring nodal temperatures or moisture contents, nodal geometry, and the thermal and moisture properties of the building structure. The stored heat energy and moisture are expressed as an increase of internal energy and moisture content at the nodes.

Heat conduction can be approximated by using the finite difference form of the Fourier law, as

$$q_{i+1\to i} = \frac{kA_i(T_{i+1}^t - T_1^t)}{\Delta x} \tag{6.50}$$

where k = thermal conductivity, Btu/h·ft·°F (W/m·°C)
A_i = area of building structure perpendicular to direction of heat flow, ft^2 (m^2)

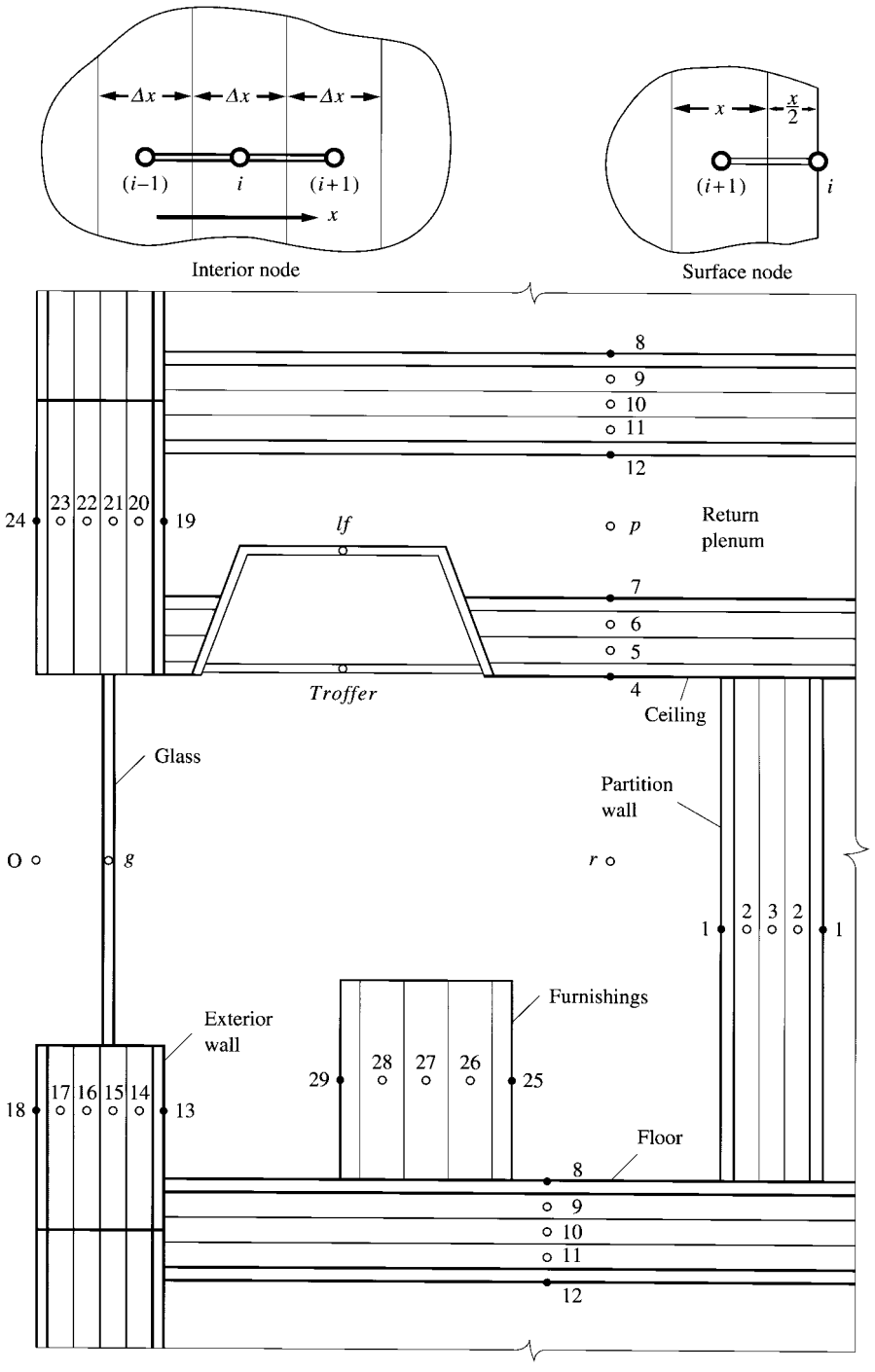


FIGURE 6.8 Building structures and nodes for a typical room.

T = temperature, °F (°C)
Δx = spacing between the nodal points, ft (m)

In Eq. (6.50), superscript t denotes at time t. Each nodal equation is solved explicitly in terms of the future temperature of that node. The explicit method is simpler and clearer than the more complex implicit method. The time derivative is then approximated by a forward finite difference in time, or

$$\frac{\partial T_i}{\partial t} \approx \frac{T_i^{t+\Delta t} - T_i^t}{\Delta t} \tag{6.51}$$

Compared with the transfer function method to calculate the cooling load, the finite difference method has the following benefits:

1. It solves heat- and moisture-transfer load calculations simultaneously.
2. The concept and approach are easily understood.
3. It permits custom-made solutions for special problems.
4. It allows direct calculation of cooling loads and energy estimates.

Its drawbacks are mainly due to the great number of computerized calculations and comparatively fewer computer programs and less information and experience are available.

Simplifying Assumptions

When the finite difference method is used to calculate space cooling loads, simplifications are often required to reduce the number of computer calculations and to solve the problem more easily. The errors due to simplification should be within acceptable limits.

For a typical room in the building, as shown in Fig. 6.8, the following are the simplifying assumptions:

- Heat and moisture flow are one-dimensional.
- Thermal properties of the building materials are homogeneous.
- The properties of the airstream flowing over the surface of the building structures are homogeneous.
- The surface temperature differences between the partition walls, ceiling, and floors are small; therefore, the radiative exchange between these surfaces can be ignored.
- The radiative energy received on the inner surface of the building structures can be estimated as the product of the shape factor and the radiative portion of the heat gains, and the shape factor is approximated by the ratio of the receiving area to the total zone area.
- During the operating period, the heat capacity of the space air is small compared with other heat gains; therefore, it can be ignored.
- When the air system is not operating during the night shutdown period, the heat capacity of the space air has a significant influence on the space air temperature; therefore, it should be taken into account.
- Different heat- and mass-transfer coefficients and analyses are used for the operating period and shutdown periods.

Heat and Moisture Transfer at Interior Nodes

Consider an interior node i as shown in the upper part of Fig. 6.8. For a one-dimensional heat flow, if there is no internal energy generation, then according to the principle of heat balance,

Conduction heat from node $i-1$ + conduction heat from node $i+1$
= rate of change of internal energy of node i

$$q_{i-1\to i} + q_{i+1\to i} = \frac{\partial U_i}{\partial t} \approx \frac{\rho_b c_b A_i \Delta x\,(T_i^{t+\Delta t} - T_i^t)}{\Delta t} \tag{6.52}$$

where U_i = internal energy of node i, Btu/lb (J/kg)
ρ_b, c_b = density and specific heat of building material, lb/ft^3 (kg/m^3) and Btu/lb·°F (J/kg·°C)
Δt = selected time interval, s or min

Substituting Eq. (6.50) into (6.52) and solving for $T_i^{t+\Delta t}$, we have

$$T_i^{t+\Delta t} = \text{Fo}(T_{i-1}^t + T_{i+1}^t) + (1 - 2\text{Fo})T_i^t \tag{6.53}$$

In Eq. (6.53), Fo is the Fourier number and is defined as

$$\text{Fo} = \frac{k_b}{\rho_b c_b}\frac{\Delta t}{(\Delta x)^2} \tag{6.54}$$

Subscript b indicates the building material.

The choice of spacing Δx and the time interval Δt must meet some criterion to ensure convergence in the calculations. The criterion is the stability limit, or

$$\text{Fo} \le \tfrac{1}{2} \tag{6.55}$$

Similarly, for moisture transfer at the interior nodes,

$$X_i^{t+\Delta t} = \text{Fo}_{\text{mass}}(X_{i-1}^t + X_{i+1}^t) + (1 - 2\text{Fo}_{\text{mass}})X_i^t \tag{6.56}$$

and

$$\text{Fo}_{\text{mass}} = \frac{D_{lv}\Delta t}{(\Delta x)^2} \le \tfrac{1}{2} \tag{6.57}$$

where X = moisture content, dimensionless
D_{lv} = mass diffusivity of liquid and vapor, ft^2/s (m^2/s)

Heat and Moisture Transfer at Surface Nodes

For a one-dimensional heat flow, the energy balance at surface node i of the partition wall as shown in Fig. 6.8 is

Conductive heat from node $i+1$ + convective heat transfer from space air
+ latent heat of moisture transfer from space air + radiative heat from internal loads
= rate of change of internal energy of node i

$$\frac{kA_i(T_{i+1}^t - T_i^t)}{\Delta x} + h_{ci}A_i(T_r^t - T_i^t) + \rho_a h_{mi}A_i X_i^t(w_r^t - w_{is}^t)\,h_{fg}$$

$$+ F_{1\to i}L_r + F_{p\to i}O_r + F_{m\to i}M_r = \rho_b c_b A_i \frac{\Delta x}{2}\frac{T_i^{t+\Delta t} - T_i^t}{\Delta t}$$

where h_{ci} = convective heat-transfer coefficient of surface i, Btu/h·ft²·°F (W/m²·°C)
ρ_a = density of space air, lb/ft³ (kg/m³)
h_{mi} = convective mass-transfer coefficient of surface i, ft/s (m/s)
w_{is}^t = humidity ratio corresponding to surface i at time t, lb/lb (kg/kg)
h_{fg} = latent heat of vaporization at surface temperature, Btu/lb (J/kg)
$F_{1\to i}, F_{p\to i}, F_{m\to i}$ = shape factor between surface of lights, occupants, appliances and surface i
L_r, O_r, M_r = radiative portion of heat energy from lights, occupants, and appliances and machines, Btu/h (W)

Solving for $T_i^{t+\Delta t}$ gives

$$T_i^{t+\Delta t} = 2\text{Fo}\left\{T_{i+1}^t + \text{Bi}\left[T_r^t + \frac{\rho_a h_{mi} h_{fg} X_i^t (w_r^t - w_{is}^t)}{h_{ci}} + \frac{F_{1\to i} L_r}{h_{ci} A_i} + \frac{F_{p\to i} O_r}{h_{ci} A_i} + \frac{F_{m\to i} M_r}{h_{ci} A_i}\right]\right\} + [1 - 2\text{Fo}(1 + \text{Bi})]T_i^t \tag{6.58}$$

In Eq. (6.58) Bi is the Biot number and can be expressed as

$$\text{Bi} = \frac{h_c \Delta x}{k} \tag{6.59}$$

The stability limit of the surface nodes requires that

$$\text{Fo}(1 + \text{Bi}) \le 1/2 \tag{6.60}$$

Similarly, according to the principle of conservation of mass, the moisture content at surface node i is given as

$$X_i^{t+\Delta t} = (1 - 2\text{Fo}_{\text{mass}})X_i^t + 2\text{Fo}_{\text{mass}} X_{i+1}^t + \frac{2\rho_a h_{mi} \Delta t X_i^t (w_r^t - w_{is}^t)}{\rho_b \Delta x} \tag{6.61}$$

The temperature and moisture content at other surface nodes such as the floor, ceiling, exterior walls, glass, and Plexiglas of the lighting fixture can be found in the same manner.

According to *ASHRAE Handbook 1993*, *Fundamentals*, the radiative and convective portions of the heat gains are as follows:

	Radiative, percent	Convective, percent
Fluorescent lights	50	50
People	33	67
External walls and roofs	60	40
Appliance and machines	20–80	80–20

Space Air Temperature and Cooling Loads

If the infiltrated air is ignored, the heat balance on the space air or the plenum air can be described by the following relationship:

Internal energy of supply air + convective heat transfer from building structures + convective heat transfer from internal loads = internal energy of space air

$$60\dot{V}_s \rho_s c_{pa} T_s^{t+\Delta t} + \sum_{k=1}^{n} h_{ck} A_k (T_i^{t+\Delta t} - T_r^{t+\Delta t}) + L_c + O_c + M_c = 60\dot{V}_s \rho_s c_{pa} T_r^{t+\Delta t}$$

Solving for $T_r^{t+\Delta t}$, we have

$$T_r^{t+\Delta t} = \frac{60\dot{V}_s\rho_s c_{pa}T_s^{t+\Delta t} + \sum_{k=1}^{n} h_{ck}A_k T_i^{t+\Delta t} + L_c + O_c + M_c}{60\dot{V}_s\rho_s c_{pa} + \sum_{k=1}^{n} h_{ck}A_k} \tag{6.62}$$

where $\dot{V}_s, T_s$ = volume flow and temperature of supply air, cfm ($m^3/(60\ s)$) and °F (°C)
L_c, O_c, M_c = convective heat from lights, occupants, and appliances, Btu/h (W)

The temperature of the return plenum air can be similarly calculated. The space sensible cooling loads, therefore, can be calculated as

$$Q_{rs}^{t+\Delta t} = \sum_{k=1}^{n} h_{ck}A_k(T_i^{t+\Delta t} - T_r^{t+\Delta t}) + L_c + O_c + M_c \tag{6.63}$$

The latent heat gains are instantaneous latent cooling loads.

6.10 HEATING LOAD

Basic Principles

The design heating load, or simply the heating load, is always the maximum heat energy that might possibly be required to supply to the conditioned space at winter design conditions to maintain the winter indoor design temperature. The maximum heating load usually occurs before sunrise on the coldest days. The following are the basic principles of heating load calculation that are different from those for the cooling load calculation:

- All heating losses are instantaneous heating loads. The heat storage effect of the building structure is ignored.
- Solar heat gains and the internal loads are usually not taken into account except for those internal loads Q_{in}, Btu/h (W), that continuously release heat to the conditioned space during the operating period of the whole heating season.
- Only that latent heat Q_l , Btu/h (W), required to evaporate liquid water for maintaining necessary space humidity is considered as heating load.

For a continuously operated heating system, the heating load Q_{rh}, Btu/h (W), can be calculated as

$$Q_{rh} = Q_{tran} + Q_{if,.s} + Q_l + Q_{mat} - Q_{in} \tag{6.64}$$

where Q_{tran} = transmission loss, Btu/h (W)
$Q_{inf,s}$ = sensible heat loss from infiltrated air, Btu/h (W)
Q_{mat} = heat added to entering colder product or material, Btu/h (W)

Transmission Loss

Transmission loss Q_{tran}, Btu/h (W), is the sum of heat losses from the conditioned space through the external walls, roof, ceiling, floor, and glass. If the calculation is simplified to a steady-state heat flow, then

$$Q_{tran} = \Sigma AU(T_r - T_o) \tag{6.65}$$

where A = area of walls, roof, ceiling, floor, or glass, ft^2 (m^2)
U = overall heat-transfer coefficient of walls, roof, ceiling, floor, or glass, Btu/h·ft^2·°F (W/m^2·°C)

When the winter outdoor design temperature is used for T_o, the heat loss calculated by transient heat transfer will be less than that from Eq. (6.65) because of the cyclic fluctuations of the outdoor temperature and the heat storage in the building structures.

For concrete slab floors on a grade, heat loss Q_{fl}, Btu/h (W), is mostly through the perimeter instead of through the floor and the ground. It can be estimated as

$$Q_{\text{fl}} = PC_{\text{fl}}(T_r - T_o) \tag{6.66}$$

where P = length of perimeter, ft (m)
C_{fl} = heat loss coefficient per foot (meter) of perimeter length, Btu/h·ft·°F (W/m·°C)

For areas having an annual total of heating degree-days HDD65 = 5350 and for a concrete wall with interior insulation in the perimeter having an R value of 5.4 h·ft^2·°F/Btu (0.97 m^2·°C/W), C_{fl} = 0.72 Btu/h·ft·°F (1.25 W/m·°C). Refer to *ASHRAE Handbook 1989*, *Fundamentals*, chapter 25, for more details.

For basement walls, the paths of the heat flow below the grade line are approximately concentric circular patterns centered at the intersection of the grade line and the basement wall. The thermal resistance of the soil and the wall depends on the path length through the soil and the construction of the basement wall. A simplified calculation of the heat loss through the basement walls and floor $Q_{b,g}$, Btu/h (W), is as follows:

$$Q_{b,g} = A_{b,g}U_{b,g}(T_{\text{base}} - T_o) \tag{6.67}$$

where $A_{b,g}$ = area of basement wall or floor below grade, ft^2 (m^2)
$U_{b,g}$ = overall heat-transfer coefficient of wall or floor and soil path, Btu/h·ft^2·°F (W/m^2·°C)

The values of $U_{b,g}$ are roughly given as follows:

	0 to 2 ft below grade	Lower than 2 ft
Uninsulated wall	0.35	0.15
Insulated wall	0.14	0.09
Basement floor	0.03	0.03

Refer to *ASHRAE Handbook 1989*, *Fundamentals*, for details.

The space heating load in the perimeter zone, including mainly transmission and infiltration losses, is sometimes expressed in a linear density $q_{h,\text{ft}}$, in Btu/h per linear foot of external wall, or Btu/h·ft (W/m).

Adjacent Unheated Spaces

Heat loss from the heated space to the adjacent unheated space Q_{un}, Btu/h (W), is usually assumed to be balanced by the heat transfer from the unheated space to the outdoor air, and this can be calculated approximately by the following formula:

$$Q_{\text{un}} = \sum_{i=1}^{n} A_iU_i(T_r - T_{\text{un}}) = (\sum_{j=1}^{m} A_jU_j + \dot{V}_{\text{inf}}\rho_o c_{pa})(T_{\text{un}} - T_o) \tag{6.68}$$

where A_i, U_i = area and overall heat-transfer coefficient of partitions between heated space and unheated space, ft^2 (m^2) and Btu/h·ft^2·°F (W/m^2·°C)
A_j, U_j = area and overall heat-transfer coefficient of building structures exposed to outdoor air in unheated space, ft^2 (m^2) and Btu/h·ft^2·°F (W/m^2·°C)

The temperature of the unheated space T_{un}, °F (°C), can be calculated as

$$T_{un} = \sum_{i=1}^{n} A_i U_i T_r + \frac{(60\dot{V}_{inf}\rho_o c_{pa} + \sum_{j=1}^{m} A_j U_j)T_o}{\sum_{i=1}^{n} A_i U_i + 60\dot{V}_{inf}\rho_o c_{pa} + \sum_{j=1}^{m} A_j U_j} \tag{6.69}$$

Latent Heat Loss and Heat Loss from Products

In Eq. (6.64), Q_l, Btu/h (W), represents the heat required to evaporate the liquid water to raise the relative humidity of the space air or to maintain a specific space relative humidity, i.e.,

$$Q_l = \dot{m}_w h_{fg,57} = [60\dot{V}_{oinf}\rho_o(w_r - w_o) - \dot{m}_p]h_{fg,57} \tag{6.70}$$

where $\dot{m}_w$ = mass flow of water evaporated, lb/h (kg/s)
$\dot{V}_{oinf}$ = volume flow rate of outdoor ventilation air and infiltrated air, cfm [m^3/(60 s)]
$\dot{m}_p$ = mass flow of water evaporated from minimum number of occupants that always stay in conditioned space when heating system is operated, lb/h (kg/s)

In Eq. (6.70), $h_{fg,57}$ indicates the latent heat of vaporization at a wet-bulb temperature of 57°F (13.9°C), that is, 72°F (22.2°C) dry-bulb temperature and a relative humidity of 40 percent. Its value can be taken as 1061 Btu/lb (2.47×10^6 J/kg).

For factories, heat added to the products or materials that enter the heated space within the occupied period Q_{mat}, Btu/h (W), should be considered part of the heating load and can be calculated as

$$Q_{mat} = \dot{m}_{mat} c_{pm}(T_r - T_o) \tag{6.71}$$

where $\dot{m}_{mat}$ = mass flow rate of cold products and cold material entering heated space, lb/h (kg/s)
c_{pm} = specific heat of product or material, Btu/lb·°F (J/kg·°C)

Infiltration

Infiltration can be considered to be 0.15 to 0.4 air changes per hour (ach) at winter design conditions only when (1) the exterior window is not well sealed and (2) there is a high wind velocity. The more sides that have windows in a room, the greater will be the infiltration. For hotels, motels, and high-rise domicile buildings, an infiltration rate of 0.038 cfm/ft^2 (0.193 L/s·m^2) of gross area of exterior windows is often used for computations for the perimeter zone.

As soon as the volume flow rate of infiltrated air $\dot{V}_{inf}$, cfm (m^3/min), is determined, the sensible heat loss from infiltration $Q_{inf,s}$, Btu/h (W), can be calculated as

$$Q_{inf,s} = \dot{V}_{inf}\rho_o c_{pa}(T_r - T_o) \tag{6.72}$$

where ρ_o = density of outdoor air, lb/ft^3 (kg/m^3).

Setback of Night Shutdown Operation

During a nighttime or unoccupied period, when the space temperature is set back lower than the indoor temperature during the operating period, it is necessary to warm up the conditioned space

the next morning before the arrival of the occupants in offices or other buildings. The warm-up or pickup load depends on the pickup temperature difference, the outdoor-indoor temperature difference, the construction of the building envelope or building shell, and the time required for warm-up. There are insufficient data to determine the oversizing factor of the capacity of the heating plant for morning warm-up. According to tests by Trehan et al. (1989), one can make a rough estimate of the energy required to warm up or raise a space temperature by 12°F (6.7°C) for a single- or two-story building with a roof-ceiling U value of 0.03 Btu/h · ft^2 · °F (0.17 W/m^2 · °C) and a U value for external walls of 0.08 Btu/h · ft^2 · °F (0.45 W/m^2 · °C) as follows:

Outdoor-indoor temperature difference, °F	Warm-up period, h	Oversizing factor, percent
35	1	40
55	2	40
55	1	100

As discussed in Sec. 5.16, heating pickup load during warm-up period depends on the difference between space temperature and occupied set point, and the amount of time prior to scheduled occupancy, and is often determined by computer software.

6.11 LOAD CALCULATION SOFTWARE

Introduction

Today, most of the load calculations are performed by personal computers. Among the widely adopted load calculation and energy analysis software, only Building Load Analysis and System Thermodynamics (BLAST) developed by the University of Illinois adopts the heat balance method. All the others are based on the transfer function or weighting factors method.

Load calculation software can also be divided into two categories. The first includes those developed by government or public institutions, such as the Department of Energy (DOE-2.0), National Bureau of Standards Load Program (NBSLP), and BLAST, which are "white box," or transparent to the user, and called public domain software. The second category consists of those programs developed by the private sector, such as TRACE 600, developed by The Trane Company; HAP E20-II, developed by Carrier Corporation; and HCC (loads) and ESP (energy), developed by Automatic Procedures for Engineering Consultants Inc. (APEC). The private-sector-developed software programs were based on the published literature of government, research, and public institutions such as ASHRAE. The most widely used, reliable, user-friendly, and continuously supported load and energy calculation programs in design are TRACE 600, HAP E20-II, and DOE-2.1E. BLAST is the most elaborate load calculation program developed in the United States and is usually considered a research tool. Most HVAC&R designers do care about the accuracy of the computational results of the software; however, the priority is the user-friendly inputs and outputs.

Trace 600—Structure and Basics

In this section we include the basics and inputs of the most widely used software program in load calculations, TRACE 600 written for DOS, in the United States. Recently, Windows 95-based TRACE 700 Load Design has become available. TRACE 600 Load and TRACE 700 Load Design are similar in structure, basics, and engineering capabilities. Most of the load and energy calculation software consists of four principal programs: loads, systems, plant, and economics (LSPE). TRACE 600 divides into five phases:

- *Load*. In the load phase, all external and internal heat gains are calculated. The heat gain profile is then converted to a cooling load profile.
- *Design*. In the design phase, based on the maximum block space sensible load, the volume flow and the size of the air system are determined from the psychrometric analysis. In design phase it also calculates the coil's load and selects the size of the coils.
- *System*. In the system simulation, the program predicts the load that is imposed on the equipment according to the space load profile and the type of air system selected.
- *Equipment.* In the equipment (plant) phase, the energy consumption of the fans, furnaces or boilers, and refrigeration systems is determined based on the hourly coil loads.
- *Economic*. During the economic analysis, utility cost is calculated from the energy use. When it is combined with installation cost and maintenance cost, a life cycle cost is available for comparison.

Only the load calculation program is covered in this section. The available documentation for TRACE 600 includes *User's Manual*, *Engineering Manual*, *Quick Reference*, *Cook Book*, *Software Bulletin—Getting Started*, etc.

The hardware requirements for TRACE 600 are as follows:

Personal computer	IBM AT 286 processor or higher
Hard disk	Program: 7.5 Mbytes hard disk space (3 M bytes for load calculation)
	Job: 3 to 10 Mbytes depending on the size of job
Ram requirement	640 kbytes
DOS Version 3.1 or higher	

The TRACE 600 data sheet can be created by using the input editor. The input file is organized in *card* format. In each input card, every input is shown by a given *field*. Sometimes, the value and unit of input are separated into two fields. Entering the inputs can be done either by using field mode or by using full-screen mode. In field mode, the user can access help and selection to determine and select a specified input. Field mode input is preferable. In full-screen mode, the user can view a full screen of raw input. Each line shows the inputs of a card and is prefixed by a two-character code. Each field in a card is separated by a forward slash.

Input data are subdivided into five groups: job, load, system, equipment, and economic. Many input cards and fields inside the input cards are optional, i.e., may be left blank.

Trace 600 Input—Load Methodology

In TRACE 600, card 10 lists various methods of cooling and heating load calculations for the user's selection. There are five cooling load calculating methods:

- *CEC-DOE2*. This method adopts the transfer function method for both heat gain and space cooling load calculations. The space load calculations adopt the precalculated weighting factors listed in DOE 2.1 *Engineering Manual*. This is a comparatively exact cooling load and energy calculation method, especially for air systems operated at nighttime shutdown mode. CEC-DOE2 needs more computational calculations. However, it is often not a primary problem when the computations are performed by a powerful PC.
- *CLTD-CLF*. This method also adopts the transfer function method for both heat gain and space cooling load calculations. Cooling load temperature difference (CLTD) and cooling load factor (CLF) tables are prepared based on the exact transfer function coefficients or weighting factors

from the TFM method. For many commercial buildings, when the air system is shut down during an unoccupied period, the external heat gains entering the space cannot be reasonably allocated over the cool-down period and the successive operating hours.

- *TETD-TA1 and TETD-TA2*. The TETD-TA1 method adopts the transfer function method to calculate heat gain, and TETD-TA2 uses an approximate TETD to calculate heat gain which adopts a decrement factor λ and a time lag δ to describe the amplitude and time-delay characteristics of the heat wave inside an exterior wall or a concrete roof slab. Both TETD-TA1 and TETD-TA2 use the time-averaging (TA) method to convert the heat gain to space cooling load. The TA technique lacks scientific support.
- *TETD-PO*. This method also adopts an approximate TETD to calculate the heat gain. It uses Post Office RMRG weighting factors to convert a heat gain of 100 percent radiative to space cooling load, and Post Office RMRX weighting factors to convert a heat gain which is not 100 percent radiative, such as heat gains from people, lights, and equipment.

There are six heating load calculation methods. Five of them—CEC-DOE2, CLTD-CLF, TETD-TA1, TETD-TA2, and TETD-PO—have already been described in the cooling load calculation methodology. The sixth heating load calculation method is called the UATD method. In the UATD method, heat losses are calculated based on the U value $\times$ area $\times$ temperature difference, which is the temperature difference of the outdoor and indoor design temperatures. Heat losses are also considered as instantaneous heating load.

For peak load calculation at winter design conditions, internal heat gains are not taken into account in the UATD method. However, for energy use calculation, internal heat gains need to be taken into account according to the load schedules. Otherwise the heating energy use becomes too conservative.

Trace 600 Input—Job

There are 11 job cards: Cards 01 to 05 are used to describe the name of the project, its location, the client of the project, the program user, and comments. Card 08, climatic information, lists the name of weather file, summer and winter clearness numbers, outdoor dry- and wet-bulb temperatures, ground reflectance, as well as the building orientation. Card 09, load simulation periods, covers the first and last month of cooling design, summer period, and daylight savings time. It also covers the peak cooling load hour.

In card 10, load simulation parameters (optional), the selection of cooling and heating load methodology has been described in preceding paragraphs. In addition, the input data cover airflow input and output units, percentage of wall load included in the return air, and room circulation rate when the transfer function method is used in the conversion of heat gain to space cooling load. Outdoor air dry-bulb temperature and humidity ratio are also required to determine the state of the supply air during psychrometric analysis.

The sensible and latent loads due to the ventilation outdoor air intake are a component of cooling coil load. They do not affect the supply airflow and the size of the supply fan if the mixture of outdoor and recirculating air is extracted by the supply fan.

Card 11, energy simulation parameters (optional), describes the first and last months of energy simulation, holiday and calendar type, and the conditioned floor area. It also determines the input data of calculation level whether it is at room, zone, or air system level.

Card 12, resource utilization factors (optional), covers the input data of energy utilization factors which indicate the inefficiency of producing and transfering energy, of electricity, gas, oil, steam, hot water, chilled water, and coal. Card 13, daylighting parameters (optional), describes the atmospheric moisture, atmospheric turbidity, or a measure of aerosols that affects daylighting, inside visible reflectivity, and geometry method, such as glass percentage (GLAS-PCT).

Trace 600 Input—External Loads

There are seven cards for external load input.

Card 19, load alternative description, will save separate files for each of the Load, Design, System, and Equipment alternatives.

Card 20, general room parameters, determines the room number, zone reference number, room description, floor length and width, construction type (a 2- to 12-in. (50- to 300-mm), light- or heavyweight concrete construction), plenum height, R value of the acoustic ceiling, floor-to-floor height, perimeter length, and duplicate floor and duplicate room/zone multipliers to save input data for a similar floor or room/zone.

Card 21, thermostat parameters, lists input data of room design cooling and heating dry-bulb temperatures and relative humidities; the highest and lowest room temperatures allowable to drift up during low occupancy or nonoccupancy; cooling and heating thermostat schedules; location of the thermostat; light, medium, or heavy room construction and the corresponding 2 to 8 h of time averaging; and carpet covering.

Card 22, roof parameters, describes the room number, alternate roof numbers, whether roof area is equal to floor area, roof length and width, roof U value, roof construction type, roof direction, roof tilting angle, and roof absorptivity α.

Card 23, skylight parameters (optional), covers the room number, roof number, length and width of skylight, number of skylights or percentage of glass of roof area, skylight U value, skylight shading coefficient, external and internal shading types, percentage of solar load to be picked up by return plenum air because a portion of skylight is exposed to the plenum air, visible light transmissivity, and the fraction of inside visible light being reflected on the inside surface of the glazing.

Card 24, wall parameters, lists the input data of room number; wall number; the length, height, U value, and construction type of the wall; wall direction; wall tilting angle; wall absorptivity; and ground reflectance.

Card 25, wall glass parameters, describes the room number, wall number, glass length and height, number of windows or percentage of glass (of gross wall area), glass U value, shading coefficient, external and internal shading type, percentage of solar load to return plenum air, visible light transmissivity, and the inside visible light being reflected on the inside surface of the glazing.

Trace 600 Input—Schedules

In TRACE 600, card 26 specifies the load and operating schedules of the internal loads, fans, reheating, and daylighting (optional). The operating schedule of internal loads and equipment affects the design load (maximum block load), especially the annual energy use.

1. For the design load calculation of internal loads include people, lighting, and miscellaneous equipment:

- TRACE 600 assumes that the internal loads are scheduled at 100 percent, 24 h/day during cooling design months operation and at 0 percent, 24 h/day during heating design months, a schedule of cooling only (CLGONLY).
- If the pickup, cool-down, or warm-up load is greater than the design load, the user should consider oversizing the cooling capacity and heating capacity to handle the pickup loads.
- For the energy use calulation of internal loads, choose or create a schedule at which the internal load varies 24 h/day.

2. For the outdoor ventilation air and infiltration loads:

- TRACE 600 assumes that the design outdoor ventilation air and infiltration are scheduled at 100 percent, 24 h/day for both cooling and heating design months.

- For the calculation of the energy use, only when the outdoor ventilation air system is separated from the main supply system is it then possible to choose and to create a schedule in which the airflow of outdoor ventilation air varies according to people's occupancy.
- During the calculation of energy use for a variable-air-volume (VAV) system, the airflow of the outdoor ventilation air may be varied based on its control systems into the following schedule: 100 percent, 24 h/day; varying according to supply volume flow of the air system; varying according to people's occupancy.
- For the calculation of energy use due to infiltration, choose and create a schedule in which the infiltration varies inversely with supply fan schedule, such as for hours fan schedule is 100 percent, infiltration is zero; then when fan schedule is 0 percent, the infiltration is 100 percent.

3. Operating schedules of fans that affect the design load and energy use calculations are as follows:

- At design load, TRACE 600 assumes that the main supply fan and auxiliary fan, if any, are scheduled at 100 percent, 24 h/day as the default value during cooling mode operation, and fan heat will be included in the cooling coil load. Similarly, at design load, main supply and auxiliary fans are scheduled at 0 percent, 24 h/day during heating mode operation, and no fan heat will be taken into account.
- In the energy use calculation, main supply and auxiliary fans in a constant-volume air system are scheduled at 100 percent, 24 h/day. Or choose or create a schedule at which fans will be cycling (on and off) depending on the space cooling and heating load.
- In the energy use calculation, main supply and auxiliary fans in a VAV system are scheduled depending on the space cooling and heating load.
- For a room exhaust fan operated at cooling design load, it is scheduled at 100 percent, 24 h/day as the default value. For the energy use calculation, choose or create an appropriate operating schedule.

4. The reheat minimum percentage determines the ratio of reheat minimum airflow for that room to the design supply airflow.

5. Daylighting schedule describes the availability of daylighting each hour. Choose an available schedule or create an appropriate daylighting schedule.

Trace 600 Input—Internal Loads

Four cards cover the input data of internal loads and airflows.

Card 27, people and lights, describes the room number, people's density and corresponding unit, people's sensible and latent loads, lighting heat gain and corresponding unit, type of lighting fixture, ballast factor, and daylighting reference points 1 and 2, to show whether there is daylighting control or one or two control areas. According to the types of lighting fixtures and the return airflow per unit floor area, the percentage of lighting load to return air should be determined more precisely. Refer to Sec. 6.6 for details.

Card 28, miscellaneous equipment (optional), covers the room number, miscellaneous equipment reference number, equipment description, energy consumption value and corresponding units, schedule and energy metering code, percentage of sensible load, percentage of miscellaneous equipment load to space and to air path, radiant fraction, and whether the plenum air, exhaust air, or return air path picks up the miscellaneous load.

Card 29, room airflows, covers the flow rates of outdoor ventilation air during cooling and heating operation and their corresponding units, the flow rates of infiltration air for cooling and heating and their corresponding units, and the amount of minimum reheat and its units.

Card 30, fan flow rates, describes the room number, airflow of the main and auxiliary cooling and heating supply fans and their corresponding units, and airflow of room exhaust fan and its unit.

Trace 600 Input—Partition and Shading Devices

Five cards cover the input data of partition and shading devices.

Card 31, partition parameters (optional), describes the room number, adjacent room number, partition number, and the length, height, *U* value, and construction type of the partition wall. The adjacent space temperature during cooling and heating periods can follow these profiles: constant, sine fit, prorate the adjacent space temperature against the outdoor air temperature, vary as the outdoor air temperature, or there is no heat transfer from the adjacent space through the partition.

Card 32, exposed floor parameters (optional), lists the room number, adjacent room number, exposed floor number, the perimeter length and loss coefficient of the slab on grade; the area, *U* value, and construction type of the exposed floor; and adjacent space temperature during cooling and heating periods whether it is constant, sine curve fit, or prorated curve fit.

Card 33, external shading devices (optional), describes its constructional type, glass height and glass width, the height above glass and the horizontal projection (projection out) of the overhang, the projection left and right, the left and right projection out, and the shading due to adjacent buildings. TRACE 600 does not allow for shading because of both adjacent building and overhang.

Card 34, internal shading devices, covers the types, overall *U* value, overall shading coefficient, location, operating schedule, and overall visible light transmittance of internal shading device; requires that below the minimum outdoor air dry-bulb temperature and above the maximum solar heat gain, the internal shading device is deployed; and sun control and glare control probability.

Card 35, daylight sensor (optional), describes the daylight sensor reference number, percentage of space affected, lighting set point in footcandles (luxes), type of control (continuous or stepped), minimum power or minimum light percentage, light control steps, manually operated control probability, height of the reference point and its distance from the glass, window sill height, ratio of the glass length seen by a sensor to the distance between the reference point and the glass, and skylight length and distance from the reference point.

Trace 600—Minimum Input Requirements, Run, and Outputs

For load calculation and energy analysis, the following are the minimum input requirements:

Card 08: climatic information, field 2: weather filename.

Card 20: general room parameters, field 1: room number.

Card 40: system type, field 1: system number, and field 2: system type.

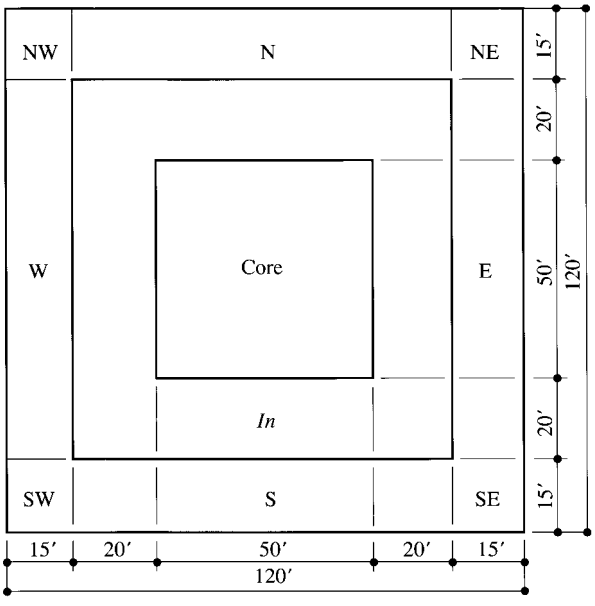
Card 41: zone assignment, field 1: system number, field 2: system serving these zones begins at, and field 3 : end at.

After the user has completed all the input required for the job, he or she must edit for errors, check for possible data loss, and run and save. Choose output from the TRACE 600 menu. Select the desired section of the output. Print the compressed report in a desired order.

Example 6.2. The plan of a typical floor in a multistory office building in New York City is shown in Fig. 6.9. The tenants and the partition walls of this floor are unknown during the design stage. The core part contains the restrooms, stairwells, and mechanical and electrical service rooms, and it has mechanical exhaust systems only. This building has the following construction characteristics:

Area of perimeter zone	6300 ft^2 (586 m^2)
Area of interior zone	5600 ft^2 (520 m^2)
Floor-to-floor height	13 ft (4 m)
Floor-to-ceiling height	9 ft (2.74 m)
Window:	
Percentage of glass/exterior wall	0.345
U value	0.67 Btu/h·ft^2·°F (3.8 W/m^2·°C)
Interior shading, SC	0.36
Exterior wall:	
Length	120 ft (36.6 m)
Height	13 ft (2.74 m)
U value	0.12 Btu/h·ft^2·°F (0.68 W/(m^2·°C)
Partition wall:	
Length	200 ft (61 m)
Height	9 ft (2.74 m)
U value	0.12 Btu/h·ft^2·°F (0.68 W/m^2·°C)

This typical floor has the following internal loads: The occupant density is 7 persons per 1000 ft^2 (93 m^2). For each person, there is 250 Btu/h (73 W) sensible heat gain, and 250 Btu/h (73 W) latent heat gain. The lighting load, including ballast allowance, is 1 W/ft^2 (10.7 W/m^2). The miscellaneous equipment load due to personal computers and appliances is 1 W/ft^2 (10.7 W/m^2).



In – interior
All dimensions in ft.

FIGURE 6.9 Typical plan of a high-rise office building.

The summer indoor design temperature is 75°F (23.9°C). If the infiltration load and the shading due to the adjacent building are ignored, calculate the maximum space cooling load during summer outdoor design conditions of this typical floor by using TRACE 600 Load program with the CLTD-CLD method.

Solution. The maximum space cooling loads for this typical floor, calculated by using TRACE 600 Load, with CLTD-CLF method on July 7, 5 P.M., are as follows:

	TRACE 600, Btu/h	Manually calculated in Example 7.2 of first edition, Btu/h
Envelope loads		
Glass solar	36,846	
Glass conduction	11,095	
Wall conduction	9,927	
Partition wall	1,296	
Ceiling load	4,226	
	63,390	
Internal loads		
Lights	36,147	
People	31,446	
Miscellaneous	37,994	
	105,587	
Grand total	168,977	198,770

Compare the TRACE 600 calculated space cooling load of 168,977 Btu/h (49,520 W) with the manually calculated space cooling load of the same typical floor with the same building and operating characteristics (in Example 7.2 of *Handbook of Air Conditioning and Refrigeration*, 1st ed.) of 198,770 Btu/h (58,240 W). The TRACE 600 calculated space cooling load is only 85 percent of the manually calculated value. This is due to a lower solar CLF and therefore lower solar cooling loads in the TRACE 600 calculation.

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CHAPTER 7
WATER SYSTEMS

7.1 FUNDAMENTALS 7.2
Types of Water System 7.2
Volume Flow and Temperature Difference 7.4
Water Velocity and Pressure Drop 7.5
7.2 WATER PIPING 7.7
Piping Material 7.7
Piping Dimensions 7.7
Pipe Joints 7.13
Working Pressure and Temperature 7.13
Expansion and Contraction 7.14
Piping Supports 7.15
Piping Insulation 7.15
7.3 VALVES, PIPE FITTINGS, AND ACCESSORIES 7.16
Types of Valves 7.16
Valve Connections and Ratings 7.17
Valve Materials 7.18
Piping Fittings and Water System Accessories 7.18
7.4 WATER SYSTEM PRESSURIZATION AND THE PRESENCE OF AIR 7.19
Water System Pressurization Control 7.19
Open Expansion Tank 7.20
Closed Expansion Tank 7.21
Size of Diaphragm Expansion Tank 7.21
Pump Location 7.23
Air in Water Systems 7.23
Penalties due to Presence of Air and Gas 7.24
Oxidation and Waterlogging 7.24
7.5 CORROSION AND DEPOSITS IN WATER SYSTEM 7.25
Corrosion 7.25
Water Impurities 7.25
Water Treatments 7.26
7.6 CLOSED WATER SYSTEM CHARACTERISTICS 7.27
System Characteristics 7.27
Changeover 7.28
7.7 CENTRIFUGAL PUMPS 7.30
Basic Terminology 7.30
Performance Curves 7.32
Net Positive Suction Head 7.33
Pump Selection 7.33
7.8 PUMP-PIPING SYSTEMS 7.34
System Curve 7.34
System Operating Point 7.34
Combination of Pump-Piping Systems 7.35
Modulation of Pump-Piping Systems 7.36
Pump Laws 7.37
Wire-to-Water Efficiency 7.37
7.9 OPERATING CHARACTERISTICS OF CHILLED WATER SYSTEM 7.38
Coil Load and Chilled Water Volume Flow 7.38
Chiller Plant 7.39
Variable Flow for Saving Energy 7.40
Water Systems in Commercial Buildings 7.40
7.10 PLANT-THROUGH-BUILDING LOOP 7.40
Bypass Throttling Flow 7.40
Distributed Pumping 7.41
Variable Flow 7.41
7.11 PLANT-BUILDING LOOP 7.43
System Description 7.43
Control Systems 7.43
System Characteristics 7.45
Sequence of Operations 7.46
Low ΔT between Chilled Water Supply and Return Temperatures 7.49
Variable-Speed Pumps Connected in Parallel 7.49
Use of Balancing Valves 7.49
Common Pipe and Thermal Contamination 7.51
7.12 PLANT-DISTRIBUTED PUMPING 7.52
7.13 CAMPUS-TYPE WATER SYSTEMS 7.53
Plant-Distribution-Building Loop 7.54
Plant-Distributed Building Loop 7.56
Multiple Sources-Distributed Building Loop 7.57
Chilled and Hot Water Distribution Pipes 7.58
7.14 COMPUTER-AIDED PIPING DESIGN AND DRAFTING 7.58
General Information 7.58
Computer-Aided Drafting Capabilities 7.58
Computer-Aided Design Capabilities 7.59
REFERENCES 7.60

7.1 FUNDAMENTALS

Types of Water System

Water systems that are part of an air conditioning system and that link the central plant, chiller/boiler, air-handling units (AHUs), and terminals may be classified into the following categories according to their use:

Chilled Water System. In a chilled water system, water is first cooled in the water chiller—the evaporator of a reciprocating, screw, or centrifugal refrigeration system located in a centralized plant—to a temperature of 40 to 50°F (4.4 to 10.0°C). It is then pumped to the water cooling coils in AHUs and terminals in which air is cooled and dehumidified. After flowing through the coils, the chilled water increases in temperature up to 60 to 65°F (15.6 to 18.3°C) and then returns to the chiller.

Chilled water is widely used as a cooling medium in central hydronic air conditioning systems. When the operating temperature is below 38°F (3.3°C), inhibited glycols, such as ethylene glycol or propylene glycol, may be added to water to create an aqueous solution with a lower freezing point.

Evaporative-Cooled Water System. In arid southwestern parts of the United States, evaporative-cooled water is often produced by an evaporative cooler to cool the air.

Hot Water Systems. These systems use hot water at temperatures between 450 and 150°F (232 and 66°C) for space and process heating purposes. Hot water systems are covered in greater detail in Chap. 8.

Dual-Temperature Water System. In a dual-temperature water system, chilled water or hot water is supplied to the coils in AHUs and terminals and is returned to the water chiller or boiler mainly through the following two distribution systems:

- Use supply and return main and branch pipes separately.
- Use the common supply and return mains, branch pipe, and coil for hot and chilled water supply and return.

The changeover from chilled water to hot water and vice versa in a building or a system depends mainly on the space requirements and the temperature of outdoor air. Hot water is often produced by a boiler; sometimes it comes from a heat recovery system, which is discussed in later chapters.

Condenser Water System. In a condenser water or cooling water system, the latent heat of condensation is removed from the refrigerant in the condenser by the condenser water. This condenser water either is from the cooling tower or is surface water taken from a lake, river, sea, or well. For an absorption refrigeration system, heat is also removed from the solution by cooling water in the absorber. The temperature of the condenser water depends mainly on the local climate.

Water systems also can be classified according to their operating characteristics into the following categories:

Closed System. In a closed system, chilled or hot water flowing through the coils, heaters, chillers, boilers, or other heat exchangers forms a closed recirculating loop, as shown in Fig. 7.1*a*. In a closed system, water is not exposed to the atmosphere during its flowing process. The purpose of recirculation is to save water and energy.

Open System. In an open system, the water is exposed to the atmosphere, as shown in Fig. 7.1*b*. For example, chilled water comes directly into contact with the cooled and dehumidified air in the air washer, and condenser water is exposed to atmosphere air in the cooling tower. Recirculation of water is used to save water and energy.

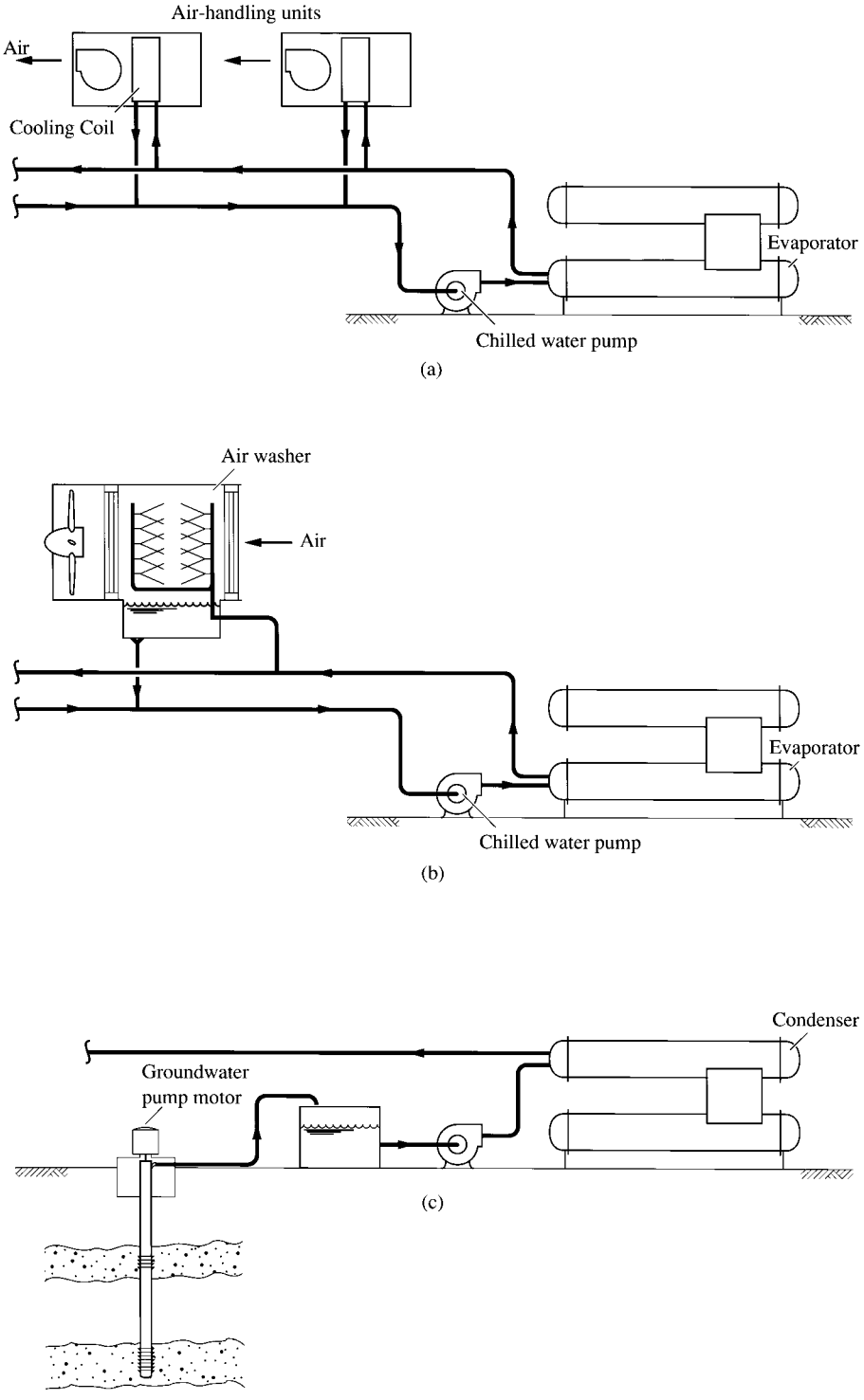


FIGURE 7.1 Types of water systems. (*a*) Closed system; (*b*) open system; (*c*) once-through system.

Open systems need more water treatments than closed systems because dust and impurities in the air may be transmitted to the water in open systems. A greater quantity of makeup water is required in open systems to compensate for evaporation, drift carryover, or blow-down operation.

Once-Through System. In a once-through system, water flows through the heat exchanger only once and does not recirculate, as shown in Fig. 7.1*c*. Lake, river, well, or seawater used as condenser cooling water represents a once-through system. Although the water cannot recirculate to the condenser because of its rise in temperature after absorbing the heat of condensation, it can still be used for other purposes, such as flushing water in a plumbing system after the necessary water treatments, to conserve water. In many locations, the law requires that well water be pumped back into the ground.

Volume Flow and Temperature Difference

The heating and cooling capacity of water when it flows through a heat exchanger Q_w, Btu/h (W), can be calculated as

$$\begin{aligned} Q_w &= \dot{V}_w \rho_w c_{pw} (T_{we} - T_{wl}) \\ &= 500 \dot{V}_{\text{gal}} (T_{we} - T_{wl}) \\ &= 500 \dot{V}_{\text{gal}} \Delta T \end{aligned} \tag{7.1}$$

where $\dot{V}_w$ = volume flow of water, ft^3/h (m^3/s)
$\dot{V}_{\text{gal}}$ = volume flow rate of water, gpm (L/s)
ρ_w = density of water, lb/ft^3 (kg/m^3)
c_{pw} = specific heat of water, Btu/lb · °F (J/kg · °C)
T_{we}, T_{wl} = temperature of water entering and leaving heat exchanger, °F (°C)
ΔT_w = temperature drop or rise of water after flowing through heat exchanger, °F (°C)

Here, the equivalent of pounds per hour is gpm = 60 min/h × 0.1337 ft^3/gal × 62.32 lb/ft^3 = 500.

Equation 7.1 also gives the relationship between ΔT_w and V_{gal} during the heat-transfer process.

The temperature of water leaving the water chiller should be no lower than 37°F (2.8°C) to prevent freezing. If the chilled water temperature is lower than 37°F (2.8°C), brine, ethylene glycol, or propylene glycol should be used. Brine is discussed in a later chapter. For a dual-temperature water system, the hot water temperature leaving the boiler often ranges from 100 to 150°F (37.8 to 65.6°C), and returns at a ΔT between 20 and 40°F (11.1 and 22.2°C).

For most dual-temperature water systems, the value of $\dot{V}_{\text{gal}}$ and the pipe size are determined based on the cooling capacity requirement for the coils and water coolers. This is because chilled water has a smaller ΔT_w than hot water does. Furthermore, the system cooling load is often higher than the system heating load. For a chilled water system to transport each refrigeration ton of cooling capacity, a ΔT_w of 8°F (4.4°C) requires a $\dot{V}_{\text{gal}}$ of 3 gpm (0.19 L/s), whereas for a ΔT_w of 24°F (13.3°C), $\dot{V}_{\text{gal}}$ is only 1.0 gpm (0.063 L/s).

The temperature of water entering the coil T_{we}, the temperature of water leaving the coil T_{wl}, and the difference between them $\Delta T_w = T_{wl} - T_{we}$ are closely related to the performance of a chilled water system, air system, and refrigeration system:

- Temperature T_{we} directly affects the power consumption in the compressor.
- The temperature differential ΔT_w is closely related to the volume flow of chilled water $\dot{V}_{\text{gal}}$ and thus the size of the water pipes and pumping power.
- Both T_{we} and ΔT_w influence the temperature and humidity ratio of air leaving the coil.

If the chilled water temperature leaving the water chiller and entering the coil is between 44 and 45°F (6.7 and 7.2°C), the off-coil temperature in the air system is usually around 55°F (12.8°C) for conventional comfort air conditioning systems. In low-temperature cold air distribution systems,

chilled water leaving the chiller may be as low as 34°F (1.1°C), and the off-coil temperature is often between 42 and 47°F, typically 44°F (5.6 and 8.3°C, typically 6.7°C).

The greater the value of ΔT_w for chilled water, the lower the amount of water flowing through the coil. Current practice is usually to use a value of ΔT_w between 10 and 18°F (5.6 and 10.0°C) for chilled water systems in buildings. Kelly and Chan (1999) noted a greater ΔT_w results at lower total power consumption in water pumps, cooling tower fans, and chillers. However, a greater ΔT_w means a larger coil and air-side pressure drop. For chilled water systems in a campus-type central plant, a value of ΔT_w between 16 and 24°F (8.9 and 13.3°C) is often used.

Water Velocity and Pressure Drop

The maximum water velocity in pipes is governed mainly by pipe erosion, noise, and water hammer. Erosion of water pipes is the result of the impingement of rapidly moving water containing air bubbles and impurities on the inner surface of the pipes and fittings. Solden and Siegel (1964) increased their feedwater velocity gradually from 8 ft/s (2.4 m/s) to an average of 35.6 ft/s (10.8 m/s). After 3 years, they found no evidence of erosion in the pipe or a connected check valve. Erosion occurs only if solid matter is contained in water flowing at high velocity. Velocity-dependent noise in pipes results from flow turbulence, cavitation, release of entrained air, and water hammer that results from the transient pressure impact on a sudden closed valve. Ball and Webster (1976) performed a series of tests on $\frac{3}{8}$-in. copper tubes with elbows. At a water velocity of 16.4 ft/s (5.0 m/s), the noise level was less than 53 dBA. Tests also showed that cold water at a speed up to 21 ft/s (6.4 m/s) did not cause cavitation. In copper and steel pipes, water hammer at a water velocity of 15 ft/s (4.6 m/s) exerted a pressure on 2-in.- (50-mm-) diameter pipes that was less than 50 percent of their design pressure.

Given the above results, excluding the energy cost for the pump power, the maximum water velocity in certain short sections of a water system may be raised to an upper limit of 11 ft/s (3.35 m/s) for a special purpose, such as enhancing the heat-transfer coefficients.

Normally, water flow in coils and heat exchangers becomes laminar and seriously impairs the heat-transfer characteristics only when its velocity drops to a value less than 2 ft/s (0.61 m/s) and its corresponding Reynolds number is reduced to about 10,000 (within the transition region). In evaporators, Redden (1996) found that at low tube water velocity at 1.15 ft/s (0.35 m/s) and low heat flux, instability of heat transfer occurred and caused the chilled water leaving temperature to fluctuate by 4°F (2.2°C). In condensers, condensing operation is not affected even if the condenser water velocity in the tube was about 1 ft/s (0.31 m/s). Water velocity should also be maintained at not less than 2 ft/s (0.61 m/s) in order to transport the entrained air to air vents.

When pipes are being sized, the optimum pressure drop ΔH_f, commonly expressed in feet (meters) of head loss of water per 100 ft of pipe length (Δp in pascals of pressure drop per meter length), is a compromise between energy costs and investments. At the same time, the age-corrosion of pipes should be considered.

Generally, the pressure drop for water pipes inside buildings ΔH_f is in a range of 1 ft/100 ft to 4 ft/100 ft (100 to 400 Pa/m), with a mean of 2.5 ft/100 ft (250 Pa/m) used most often. Because of a lower increase in installation cost for smaller-diameter pipes, it may be best to use a pressure drop lower than 2.5 ft/100 ft (250 Pa/m) when the pipe diameter is 2 in. or less.

Age corrosion results in an increase in the friction factor and a decrease in the effective diameter. The factors that contribute to age corrosion are sliming, caking of calcareous salts, and corrosion. Many scientists recommended an increase in friction loss of 15 to 20 percent, resulting in a design pressure drop of 2 ft/100 ft (200 Pa/m), for closed water systems; and a 75 to 90 percent increase in friction loss, or a design pressure drop of 1.35 ft/100 ft (135 Pa/m), for open water systems.

Figures 7.2, 7.3, and 7.4 show the pressure drop charts for steel, copper, and plastic pipes, respectively, for closed water systems. Each chart shows the volume flow $\dot{V}_{\text{gal}}$ (gpm), pressure drop ΔH_f (ft/100 ft), water velocity v_w (ft/s), and water pipe diameter D (in.). Given any two of these parameters, the other two can be determined. For instance, for a steel water pipe that has a water volume flow of 1000 gpm, if the pressure drop is 2 ft/100 ft, the diameter is 8 in. and the corresponding velocity is about 8 ft/s.

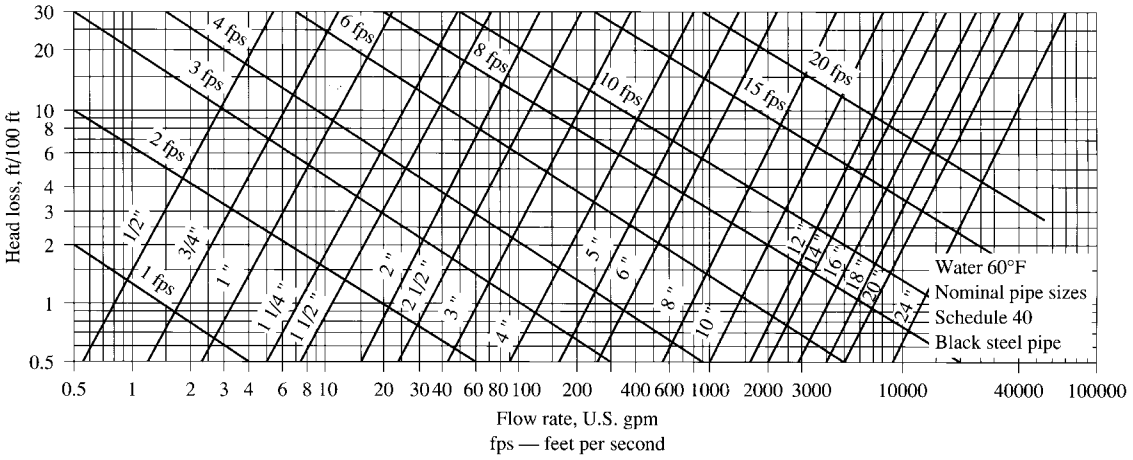


FIGURE 7.2 Friction chart for water in steel pipes (Schedule 40). (*Source: ASHRAE Handbook 1989 Fundamentals. Reprinted with permission.*)

It is a common practice to limit the water velocity to no more than 4 ft/s (1.2 m/s) for water pipes 2 in. (50 mm) or less in diameter in order to prevent an excessive ΔH_f. The pressure drop should not exceed 4 ft/100 ft (400 Pa/m) for water pipes of greater than 2-in. (50-mm) diameter.

An open water system or a closed water system that is connected with an open expansion tank, all the pressure differences between two points or levels, and pressure drops across a piece of equipment or a device are expressed in feet of water column or psi (head in meters of water column or pressure loss in kPa). The total or static pressure of water at a certain point in a water system is actually measured and expressed by that part of pressure which is greater or smaller than the atmospheric pressure, often called *gauge pressure*, in feet of water column gauge or psig (meters gauge or kPa · g). The relationships between the steady flow energy equation and the fluid pressure, and between the pressure loss and fluid head, are discussed in Sec.17.1.

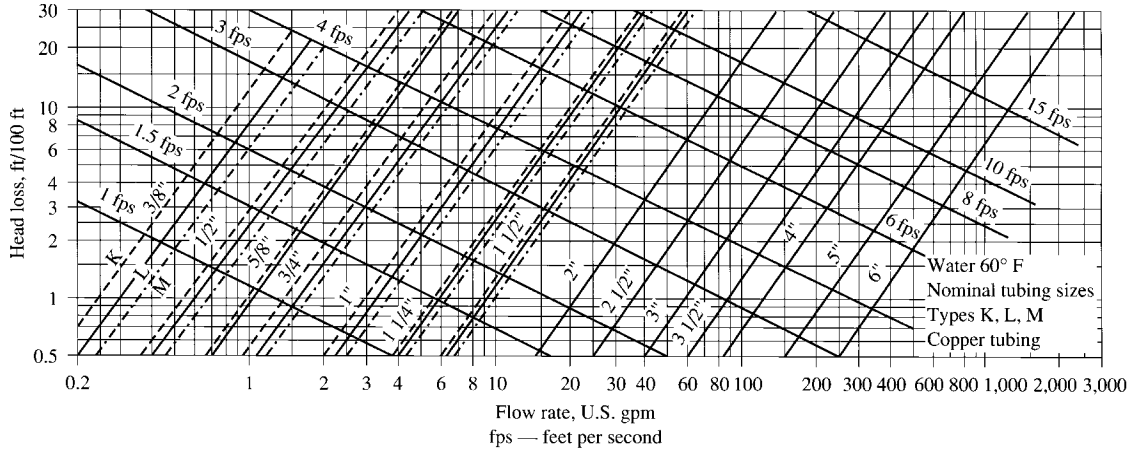


FIGURE 7.3 Friction chart for water in copper tubing (types K, L, and M). (*Source: ASHRAE Handbook 1989 Fundamentals. Reprinted with permission.*)

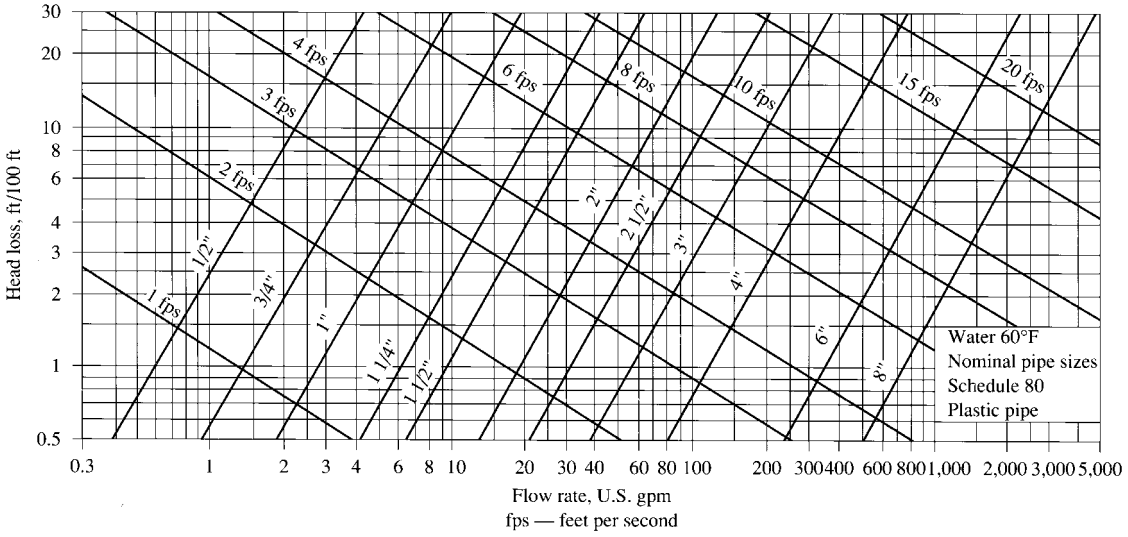


FIGURE 7.4 Friction chart for water in plastic pipes (Schedule 80). (*Source: ASHRAE Handbook 1989 Fundamentals. Reprinted with permission.*)

7.2 WATER PIPING

Piping Material

For water systems, the piping materials most widely used are steel, both black (plain) and galvanized (zinc-coated), in the form of either welded-seam steel pipe or seamless steel pipe; ductile iron and cast iron; hard copper; and polyvinyl chloride (PVC). The piping materials for various services are shown below:

Chilled water	Black and galvanized steel
Hot water	Black steel, hard copper
Cooling water and drains	Black steel, galvanized ductile iron, PVC

Copper, galvanized steel, galvanized ductile iron, and PVC pipes have better corrosion resistance than black steel pipes. Technical requirements, as well as local customs, determine the selection of piping materials.

Piping Dimensions

The steel pipe wall thicknesses currently used were standardized in 1930. The thickness ranges from Schedule 10, light wall, to Schedule 160, very heavy wall. Schedule 40 is the standard for a pipe with a diameter up to 10 in. (250 mm). For instance, a 2-in. (50-mm) standard pipe has an outside diameter of 2.375 in. (60.33 mm) and an inside diameter of 2.067 in. (52.50 mm). The nominal pipe size is only an approximate indication of pipe size, especially for pipes of small diameter. Table 7.1 lists the dimensions of commonly used steel pipes.

The outside diameter of extruded copper is standardized so that the outside diameter of the copper tubing is 1/8 in. (3.2 mm) larger than the nominal size used for soldered or brazed socket joints. As in the case with steel pipes, the result is that the inside diameters of copper tubes seldom equal the nominal sizes. Types K, L, M, and DWV designate the wall thickness of copper tubes: type K is the heaviest, and DWV is the lightest. Type L is generally used as the standard for pressure copper tubing. Type DWV is used for drainage at atmospheric pressure.

TABLE 7.1 Dimensions of Commonly Used Steel Pipes

Nominal size and pipe OD D, in.	Schedule number or weight*	Wall thickness t, in.	Inside diameter d, in.	Surface area		Cross-sectional		Weight of		Working pressure† ASTM A538B to 400°F		
				Outside, ft²/ft	Inside, ft²/ft	Metal area, in.²	Flow area, in.²	Pipe, lb/ft	Water, lb/ft	Mfr. process	Joint type	psig
$\frac{1}{4}$	40 ST	0.088	0.364	0.141	0.095	0.125	0.104	0.424	0.045	CW	Thrd	188
$D = 0.540$	80 XS	0.119	0.302	0.141	0.079	0.157	0.072	0.535	0.031	CW	Thrd	871
$\frac{3}{8}$	40 ST	0.091	0.493	0.177	0.129	0.167	0.191	0.567	0.083	CW	Thrd	203
$D = 0.675$	80 XS	0.126	0.423	0.177	0.111	0.217	0.141	0.738	0.061	CW	Thrd	820
$\frac{1}{2}$	40 ST	0.109	0.622	0.220	0.163	0.250	0.304	0.850	0.131	CW	Thrd	214
$D = 0.840$	80 XS	0.147	0.546	0.220	0.143	0.320	0.234	1.087	0.101	CW	Thrd	753
$\frac{3}{4}$	40 ST	0.113	0.824	0.275	0.216	0.333	0.533	1.13	0.231	CW	Thrd	217
$D = 1.050$	80 XS	0.154	0.742	0.275	0.194	0.433	0.432	1.47	0.187	CW	Thrd	681
1	40 ST	0.133	1.049	0.344	0.275	0.494	0.864	1.68	0.374	CW	Thrd	226
$D = 1.315$	80 XS	0.179	0.957	0.344	0.251	0.639	0.719	2.17	0.311	CW	Thrd	642
$1\frac{1}{4}$	40 ST	0.140	1.380	0.435	0.361	0.669	1.50	2.27	0.647	CW	Thrd	229
$D = 1.660$	80 XS	0.191	1.278	0.435	0.335	0.881	1.28	2.99	0.555	CW	Thrd	594
$1\frac{1}{2}$	40 ST	0.145	1.610	0.497	0.421	0.799	2.04	2.72	0.881	CW	Thrd	231
$D = 1.900$	80 XS	0.200	1.500	0.497	0.393	1.068	1.77	3.63	0.765	CW	Thrd	576
2	40 ST	0.154	2.067	0.622	0.541	1.07	3.36	3.65	1.45	CW	Thrd	230
$D = 2.375$	80 XS	0.218	1.939	0.622	0.508	1.48	2.95	5.02	1.28	CW	Thrd	551
$2\frac{1}{2}$	40 ST	0.203	2.469	0.753	0.646	1.70	4.79	5.79	2.07	CW	Weld	533
$D = 2.875$	80 XS	0.276	2.323	0.753	0.608	2.25	4.24	7.66	1.83	CW	Weld	835
3	40 ST	0.216	3.068	0.916	0.803	2.23	7.39	7.57	3.20	CW	Weld	482
$D = 3.500$	80 XS	0.300	2.900	0.916	0.759	3.02	6.60	10.25	2.86	CW	Weld	767
4	40 ST	0.237	4.026	1.178	1.054	3.17	12.73	10.78	5.51	CW	Weld	430
$D = 4.500$	80 XS	0.337	3.826	1.178	1.002	4.41	11.50	14.97	4.98	CW	Weld	695
6	40 ST	0.280	6.065	1.734	1.588	5.58	28.89	18.96	12.50	ERW	Weld	696
$D = 6.625$	80 XS	0.432	5.761	1.734	1.508	8.40	26.07	28.55	11.28	ERW	Weld	1209

TABLE 7.1 Dimensions of Commonly Used Steel Pipes (Continued)

Nominal size and pipe OD D, in.	Schedule number or weight*	Wall thickness t, in.	Inside diameter d, in.	Surface area		Cross-sectional		Weight of		Working pressure† ASTM A538B to 400°F		
				Outside, ft^2/ft	Inside, ft^2/ft	Metal area, $in.^2$	Flow area, $in.^2$	Pipe, lb/ft	Water, lb/ft	Mfr. process	Joint type	psig
8	30	0.277	8.071	2.258	2.113	7.26	51.16	24.68	22.14	ERW	Weld	526
$D = 8.625$	40 ST	0.322	7.981	2.258	2.089	8.40	50.03	28.53	21.65	ERW	Weld	643
	80 XS	0.500	7.625	2.258	1.996	12.76	45.66	43.35	19.76	ERW	Weld	1106
	30	0.307	10.136	2.814	2.654	10.07	80.69	34.21	34.92	ERW	Weld	485
10	40 ST	0.365	10.020	2.814	2.623	11.91	78.85	40.45	34.12	ERW	Weld	606
$D = 10.75$	XS	0.500	9.750	2.814	2.552	16.10	74.66	54.69	32.31	ERW	Weld	887
	80	0.593	9.564	2.814	2.504	18.92	71.84	64.28	31.09	ERW	Weld	1081
	30	0.330	12.090	3.338	3.165	12.88	114.8	43.74	49.68	ERW	Weld	449
12	ST	0.375	12.000	3.338	3.141	14.58	113.1	49.52	48.94	ERW	Weld	528
$D = 12.75$	40	0.406	11.938	3.338	3.125	15.74	111.9	53.48	48.44	ERW	Weld	583
	XS	0.500	11.750	3.338	3.076	19.24	108.4	65.37	46.92	ERW	Weld	748
	80	0.687	11.376	3.338	2.978	26.03	101.6	88.44	43.98	ERW	Weld	1076
	30 ST	0.375	13.250	3.665	3.469	16.05	137.9	54.53	59.67	ERW	Weld	481
14	40	0.437	13.126	3.665	3.436	18.62	135.3	63.25	58.56	ERW	Weld	580
$D = 14.00$	XS	0.500	13.000	3.665	3.403	21.21	132.7	72.04	57.44	ERW	Weld	681
	80	0.750	12.500	3.665	3.272	31.22	122.7	106.05	53.11	ERW	Weld	1081
16	30 ST	0.375	15.250	4.189	3.992	18.41	182.6	62.53	79.04	ERW	Weld	421
$D = 16.00$	40 XS	0.500	15.000	4.189	3.927	24.35	176.7	82.71	76.47	ERW	Weld	596
	ST	0.375	17.250	4.712	4.516	20.76	233.7	70.54	101.13	ERW	Weld	374
18	30	0.437	17.126	4.712	4.483	24.11	230.3	81.91	99.68	ERW	Weld	451
$D = 18.00$	XS	0.500	17.000	4.712	4.450	27.49	227.0	93.38	98.22	ERW	Weld	530
	40	0.562	16.876	4.712	4.418	30.79	223.7	104.59	96.80	ERW	Weld	607
20	ST	0.375	19.250	5.236	5.039	23.12	291.0	78.54	125.94	ERW	Weld	337
$D = 20.00$	30 XS	0.500	19.000	5.236	4.974	30.63	283.5	104.05	122.69	ERW	Weld	477
	40	0.593	18.814	5.236	4.925	36.15	278.0	122.82	120.30	ERW	Weld	581

*Numbers are schedule number per ASTM B36.10; ST = standard weight; XS = extra strong.

†Working pressures have been calculated per ASME/ANSI B31.9 using furnace butt weld (continuous weld, CW) pipe through 4 in. and electric resistance weld (ERW) thereafter. The allowance A has been taken as (a) 12.5 percent of t for mill tolerance on pipe wall thickness, plus (b) an arbitrary corrosion allowance of 0.025 in. for pipe sizes through NPS 2 and 0.065 in. from NPS $2\frac{1}{2}$ through 20 plus (c) a thread cutting allowance for sizes through NPS 2.

Because the pipe wall thickness of threaded standard weight pipe is so small after deducting the allowance A, the mechanical strength of the pipe is impaired. It is good practice to limit standard-weight threaded pipe pressures to 90 psig for steam and 125 psig for water.

***Source*:** *ASHRAE Handbook 1988, Equipment*. Reprinted with permission.

TABLE 7.2 Dimensions of Copper Tubes

Nominal diameter, in.	Type	Wall thickness t, in.	Outside diameter D, in.	Inside diameter d, in.	Surface area		Cross-sectional		Weight of		Working pressure* ASTMB 888 to 250°F	
					Outside, ft^2/ft	Inside, ft^2/ft	Metal area, $in.^2$	Flow area, $in.^2$	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
$\frac{1}{4}$	K	0.035	0.375	0.305	0.098	0.080	0.037	0.073	0.145	0.032	851	1596
	L	0.030	0.375	0.315	0.098	0.082	0.033	0.078	0.126	0.034	730	1368
$\frac{3}{8}$	K	0.049	0.500	0.402	0.131	0.105	0.069	0.127	0.269	0.055	894	1676
	L	0.035	0.500	0.430	0.131	0.113	0.051	0.145	0.198	0.063	638	1197
	M	0.025	0.500	0.450	0.131	0.008	0.037	0.159	0.145	0.069	456	855
$\frac{1}{2}$	K	0.049	0.625	0.527	0.164	0.138	0.089	0.218	0.344	0.094	715	1341
	L	0.040	0.625	0.545	0.164	0.143	0.074	0.233	0.285	0.101	584	1094
	M	0.028	0.625	0.569	0.164	0.149	0.053	0.254	0.203	0.110	409	766
$\frac{5}{8}$	K	0.049	0.750	0.652	0.196	0.171	0.108	0.334	0.418	0.144	596	1117
	L	0.042	0.750	0.666	0.196	0.174	0.093	0.348	0.362	0.151	511	958
$\frac{3}{4}$	K	0.065	0.875	0.745	0.229	0.195	0.165	0.436	0.641	0.189	677	1270
	L	0.045	0.875	0.785	0.229	0.206	0.117	0.484	0.455	0.209	469	879
	M	0.032	0.875	0.811	0.229	0.212	0.085	0.517	0.328	0.224	334	625
1	K	0.065	1.125	0.995	0.295	0.260	0.216	0.778	0.839	0.336	527	988
	L	0.050	1.125	1.025	0.295	0.268	0.169	0.825	0.654	0.357	405	760
	M	0.035	1.125	1.055	0.295	0.276	0.120	0.874	0.464	0.378	284	532
$1\frac{1}{4}$	K	0.065	1.375	1.245	0.360	0.326	0.268	1.217	1.037	0.527	431	808
	L	0.055	1.375	1.265	0.360	0.331	0.228	1.257	0.884	0.544	365	684
	M	0.042	1.375	1.291	0.360	0.338	0.176	1.309	0.682	0.566	279	522
	DWV	0.040	1.375	1.295	0.360	0.339	0.168	1.317	0.650	0.570	265	497
$1\frac{1}{2}$	K	0.072	1.625	1.481	0.425	0.388	0.351	1.723	1.361	0.745	404	758
	L	0.060	1.625	1.505	0.425	0.394	0.295	1.779	1.143	0.770	337	631
	M	0.049	1.625	1.527	0.425	0.400	0.243	1.831	0.940	0.792	275	516
	DWV	0.042	1.625	1.541	0.425	0.403	0.209	1.865	0.809	0.807	236	442
2	K	0.083	2.125	1.959	0.556	0.513	0.532	3.014	2.063	1.304	356	668
	L	0.070	2.125	1.985	0.556	0.520	0.452	3.095	1.751	1.339	300	573
	M	0.058	2.125	2.009	0.556	0.526	0.377	3.170	1.459	1.372	249	467
	DWV	0.042	2.125	2.041	0.556	0.534	0.275	3.272	1.065	1.416	180	338

TABLE 7.2 Dimensions of Copper Tubes (Continued)

Nominal diameter, in.	Type	Wall thickness t, in.	Outside diameter D, in.	Inside diameter d, in.	Surface area		Cross-sectional		Weight of		Working pressure* ASTMB 888 to 250°F	
					Outside, ft^2/ft	Inside, ft^2/ft	Metal area, $in.^2$	Flow area, $in.^2$	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
$2\frac{1}{2}$	K	0.095	2.625	2.435	0.687	0.637	0.755	4.657	2.926	2.015	330	619
	L	0.080	2.625	2.465	0.687	0.645	0.640	4.772	2.479	2.065	278	521
	M	0.065	2.625	2.495	0.687	0.653	0.523	4.889	2.026	2.116	226	423
3	K	0.109	3.125	2.907	0.818	0.761	1.033	6.637	4.002	2.872	318	596
	L	0.090	3.125	2.945	0.818	0.771	0.858	6.812	3.325	2.947	263	492
	M	0.072	3.125	2.981	0.818	0.780	0.691	6.979	2.676	3.020	210	394
	DWV	0.045	3.125	3.035	0.818	0.795	0.435	7.234	1.687	3.130	131	246
$3\frac{1}{2}$	K	0.120	3.625	3.385	0.949	0.886	1.321	8.999	5.120	3.894	302	566
	L	0.100	3.625	3.425	0.949	0.897	1.107	9.213	4.291	3.987	252	472
	M	0.083	3.625	3.459	0.949	0.906	0.924	9.397	3.579	4.066	209	392
4	K	0.134	4.125	3.857	1.080	1.010	1.680	11.684	6.510	5.056	296	555
	L	0.110	4.125	3.905	1.080	1.022	1.387	11.977	5.377	5.182	243	456
	M	0.095	4.125	3.935	1.080	1.030	1.203	12.161	4.661	5.262	210	394
	DWV	0.058	4.125	4.009	1.080	1.050	0.741	12.623	2.872	5.462	128	240
5	K	0.160	5.125	4.805	1.342	1.258	2.496	18.133	9.671	7.846	285	534
	L	0.125	5.125	4.875	1.342	1.276	1.963	18.665	7.609	8.077	222	417
	M	0.109	5.125	4.907	1.342	1.285	1.718	18.911	6.656	8.183	194	364
	DWV	0.072	5.125	4.981	1.342	1.304	1.143	19.486	4.429	8.432	128	240
6	K	0.192	6.125	5.741	1.603	1.503	3.579	25.886	13.867	11.201	286	536
	L	0.140	6.125	5.845	1.603	1.530	2.632	26.832	10.200	11.610	208	391
	M	0.122	6.125	5.881	1.603	1.540	2.301	27.164	8.916	11.754	182	341
	DWV	0.083	6.125	5.959	1.603	1.560	1.575	27.889	6.105	12.068	124	232
8	K	0.271	8.125	7.583	2.127	1.985	6.687	45.162	25.911	19.542	304	570
	L	0.200	8.125	7.725	2.127	2.022	4.979	46.869	19.295	20.280	224	421
	M	0.170	8.125	7.785	2.127	2.038	4.249	47.600	16.463	20.597	191	358
	DWV	0.109	8.125	7.907	2.127	2.070	2.745	49.104	10.637	21.247	122	229
10	K	0.338	10.125	9.449	2.651	2.474	10.392	70.123	40.271	30.342	304	571
	L	0.250	10.125	9.625	2.651	2.520	7.756	72.760	30.054	31.483	225	422
	M	0.212	10.125	9.701	2.651	2.540	6.602	73.913	25.584	31.982	191	358
12	K	0.405	12.125	11.315	3.174	2.962	14.912	100.554	57.784	43.510	305	571
	L	0.280	12.125	11.565	3.174	3.028	10.419	105.046	40.375	45.454	211	395
	M	0.254	12.125	11.617	3.174	3.041	9.473	105.993	36.706	45.863	191	358

*When using soldered or brazed fittings, the joint determines the limiting pressure.

Working pressures calculated using ASME B31.9 allowable stresses. A 5 percent mill tolerance has been used on the wall thickness. Higher tube ratings can be calculated using allowable stress for lower temperatures.

If soldered or brazed fittings are used on hard-drawn tubing, use the annealed ratings. Full-tube allowable pressures can be used with suitably rated flare or compression-type fittings.

***Source*:** *ASHRAE Handbook 1988, Equipment*. Reprinted with permission.

Copper tubes are also categorized as hard and soft copper. Soft pipes should be used in applications for which the pipe will be bent in the field. Table 7.2 lists the dimensions of copper tubes.

Thermoplastic plastic pipes are the most widely used plastic pipes in air conditioning. They are manufactured with dimensions that match steel pipe dimensions. The advantages of plastic pipes include resistance to corrosion, scaling, and the growth of algae and fungi. Plastic pipes have smooth surfaces and negligible age allowance. *Age allowance* is the allowance for corrosion and scaling for plastic pipes during their service life. Most plastic pipes are low in cost, especially compared with corrosion-resistant metal tubes.

The disadvantages of plastic pipes include the fact that their pressure ratings decrease rapidly when the water temperature rises above 100°F (37.8°C). PVC pipes are weaker than metal pipes and must usually be thicker than steel pipes if the same working pressure is to be maintained. Plastic pipes may experience expansion and contraction during temperature changes that is 4 times greater than that of steel. Precautions must be taken to protect plastic pipes from external damage and to account for its behavior during fire. Some local codes do not permit the use of some or all-plastic pipes. It is necessary to check with local authorities.

Pipe Joints

Steel pipes of small diameter (2 in. or 50 mm less) threaded through cast-iron fittings are the most widely used type of pipe joint. For steel pipes of diameter 2 in. (50 mm) and more, welded joints,

TABLE 7.3 Maximum Allowable Pressures at Corresponding Temperatures

						System	
				Fitting			
Application	Pipe material	Weight	Joint type	Class	Material	Temperature, °F	Maximum allowable pressure at temperature, psig
Recirculating water							
2 in. and smaller	Steel (CW)	Standard	Thread	125	Cast iron	250	125
	Copper, hard	Type L	95-5 solder	—	Wrought copper	250	150
	PVC	Sch. 80	Solvent	Sch. 80	PVC	75	350
	CPVC	Sch. 80	Solvent	Sch. 80	CPVC	150	150
	PB	SDR-11	Heat fusion	—	PB	160	115
			Insert crimp	—	Metal	160	115
2.5–12 in.	A53 B ERW steel	Standard	Weld	Standard	Wrought steel	250	400
			Flange	150	Wrought steel	250	250
			Flange	125	Cast iron	250	175
			Flange	250	Cast iron	250	400
			Groove	—	MI or ductile iron	230	300
	PB	SDR-11	Heat fusion		PB	160	115
Refrigerant							
	Copper, hard	Type L or K	Braze	—	Wrought copper	—	—
	A53 B SML steel	Standard	Weld		Wrought steel	—	—

Note: Maximum allowable working pressures have been derated in this table. Higher system pressures can be used for lower temperatures and smaller pipe sizes. Pipe, fittings, joints, and valves must all be considered.

Note: A53 ASTM Standard A53
PVC Polyvinyl chloride
CPVC Chlorinated polyvinyl chloride
PB Polybutylene

***Source*:** Abridged with permission from *ASHRAE Handbook 1988, Equipment*.

bolted flanges, and grooved ductile iron joined fittings are often used. Galvanized steel pipes are threaded together by galvanized cast iron or ductile iron fittings.

Copper pipes are usually joined by soldering and brazing socket end fittings. Plastic pipes are often joined by solvent welding, fusion welding, screw joints, or bolted flanges.

Vibrations from pumps, chillers, or cooling towers can be isolated or dampened by means of flexible pipe couplings. Arch connectors are usually constructed of nylon, dacron, or polyester and neoprene. They can accommodate deflections or dampen vibrations in all directions. Restraining rods and plates are required to prevent excessive stretching. A flexible metal hose connector includes a corrugated inner core with a braided cover. It is available with flanged or grooved end joints.

Working Pressure and Temperature

In a water system, the maximum allowable working pressure and temperature are not limited to the pipes only; joints or the pipe fittings, especially valves, may often be the weak links. Table 7.3 lists types of pipes, joint, and fittings and their maximum allowable working pressures for specified temperatures.

Expansion and Contraction

During temperature changes, all pipes expand and contract. The design of water pipes must take into consideration this expansion and contraction. Both the temperature change during the operating period and the possible temperature change between the operating and shutdown periods should

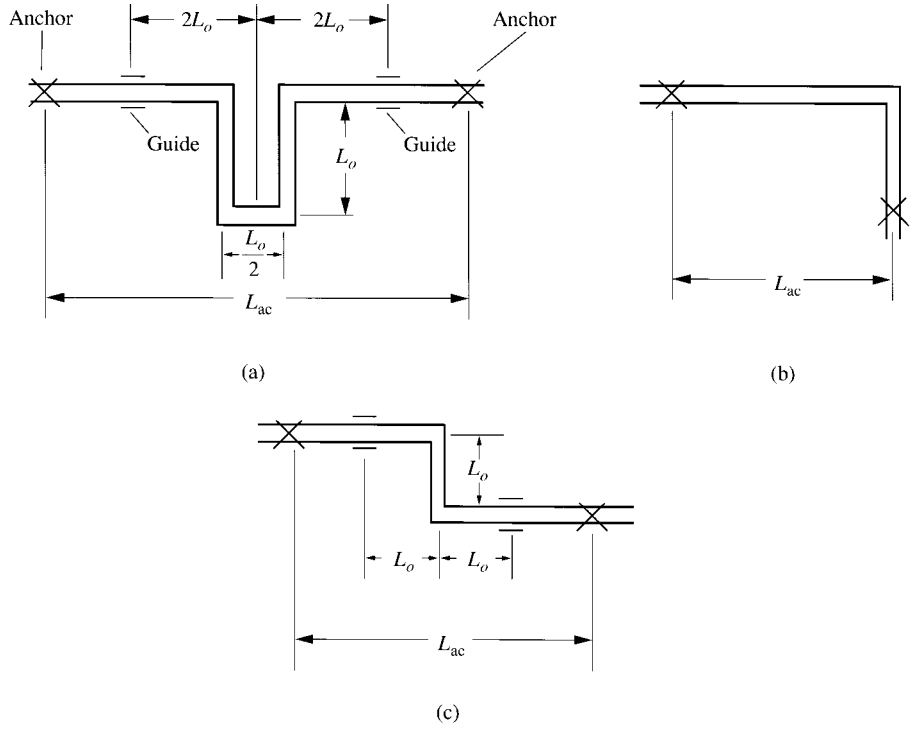


FIGURE 7.5 Expansion loops. (*a*) U bends; (*b*) L bends; (*c*) Z bends.

also be considered. For chilled and condenser water, which has a possible temperature change of 40 to 100°F (4.4 to 37.8°C), expansion and contraction cause considerable movement in a long run of piping. Unexpected expansion and contraction cause excess stress and possible failure of the pipe, pipe support, pipe joints, and fittings.

Expansion and contraction of hot and chilled water pipes can be better accommodated by using loops and bends. The commonly used bends are U bends, Z bends, and L bends, as shown in Fig. 7.5. Anchors are the points where the pipe is fixed so that it will expand or contract between them. Reaction forces at these anchors should be considered when the support is being designed. *ASHRAE Handbook 1992*, *HVAC Systems and Equipment*, gives the required calculations and data for determining these stresses. Guides are used so that the pipes expand laterally.

Empirical formulas are often used instead of detailed stress analyses to determine the dimension of the offset leg L_o [ft (m)]. Waller (1990) recommended the following formulas:

$$\begin{aligned} \text{U bends: } L_o &= 0.041D^{0.48}L_{ac}^{0.46}\,\Delta T \\ \text{Z bends: } L_o &= (0.13DL_{ac}\Delta T)^{0.5} \\ \text{L bends: } L_o &= (0.314DL_{ac}\Delta T)^{0.5} \end{aligned} \tag{7.2}$$

where D = diameter of pipe, in. (mm)
L_{ac} = distance between anchors, hundreds of ft (m)
ΔT = temperature difference, °F (°C)

If there is no room to accommodate U, Z, or L bends (such as in high-rise buildings or tunnels), mechanical expansion joints are used to compensate for movement during expansion. Packed expansion joints allow the pipe to slide to accommodate movement during expansion. Various types of packing are used to seal the sliding surfaces in order to prevent leakage. Another type of mechanical joint uses bellows or flexible metal hose to accommodate movement. These types of joints should be carefully installed to avoid distortion.

TABLE 7.4 Recommended Pipe Hanger Spacing, ft

Nominal pipe diameter, in.	Standard-weight steel pipe (water)	Copper tube (water)	Rod size, in.
$\frac{1}{2}$	7	5	$\frac{1}{4}$
$\frac{3}{4}$	7	5	$\frac{1}{4}$
1	7	6	$\frac{1}{4}$
$1\frac{1}{2}$	9	8	$\frac{3}{8}$
2	10	8	$\frac{3}{8}$
$2\frac{1}{2}$	11	9	$\frac{3}{8}$
3	12	10	$\frac{3}{8}$
4	14	12	$\frac{1}{2}$
6	17	14	$\frac{1}{2}$
8	19	16	$\frac{5}{8}$
10	20	18	$\frac{3}{4}$
12	23	19	$\frac{7}{8}$
14	25		1
16	27		1
18	28		$1\frac{1}{4}$
20	30		$1\frac{1}{4}$

Note: Spacing does not apply where concentrated loads are placed between supports such as flanges, valves, and specialties.

***Source*:** *ASHRAE Handbook 1988, Equipment*. Reprinted with permission.

Piping Supports

Types of piping support include hangers, which hang the pipe from above; supports, which usually use brackets to support the pipe from below; anchors to control the movement of the piping; and guides to guide the axial movement of the piping. Table 7.4 lists the recommended spacing of pipe hangers. Piping support members should be constructed based on the stress at their point of connection to the pipe as well as on the characteristics of the structural system. Pipe supports must have sufficient strength to support the pipe, including the water inside. Except for the anchors, they should also allow for expansion movement.

Pipes should be supported around the connections to the equipment so that the pipe's weight and expansion or contraction do not affect the equipment. For insulated pipes, heavy-gauge sheet-metal half-sleeves are used between the hangers and the insulation. Corrosion protection should also be carefully considered.

Piping Insulation

External pipe insulation should be provided when the inside water temperature $105°F < T_w < 60°F$ ($41°C < T_w < 15.6°C$) for the sake of energy saving, surface condensation, and high-temperature safety protection. The optimum thickness of the insulation of pipes depends mainly on the operating temperature of the inside water, the pipe diameter, and the types of service. There is a compromise between initial cost and energy cost. ASHRAE/IESNA Standard 90.1-1999 specifies the minimum pipe insulation thickness for water systems, as listed in Table 7.5. Insulation shall be protected from damage including that because of sunlight, moisture, equipment maintenance, and wind.

TABLE 7.5 Minimum Pipe Insulation Thickness*, in.

Fluid design operating temperature range, °F	Insulation conductivity		Nominal pipe or tube size, in.				
	Conductivity, Btu·in./h·ft²·°F	Mean rating temp. °F	<1	1 to <1½	1½ to <4	4 to <8	≥8
Heating systems (steam, steam condensate, and hot water)†‡							
>350	0.32–0.34	250	2.5	3.0	3.0	4.0	4.0
251–350	0.29–0.32	200	1.5	2.5	3.0	3.0	3.0
201–250	0.27–0.30	150	1.5	1.5	2.0	2.0	2.0
141–200	0.25–0.29	125	1.0	1.0	1.0	1.5	1.5
105–140	0.22–0.28	100	0.5	0.5	1.0	1.0	1.0
Domestic and service hot water systems							
105+	0.22–0.28	100	0.5	0.5	1.0	1.0	1.0
Cooling systems (chilled water, brine, and refrigerant)§							
40–60	0.22–0.28	100	0.5	0.5	1.0	1.0	1.0
<40	0.22–0.28	100	0.5	1.0	1.0	1.0	1.5

*For insulation outside the stated conductivity range, the minimum thickness T shall be determined as follows:

$$T = r[(1 + t/r)^{K/k} - 1]$$

where T = minimum insulation thickness (in.), r = actual outside radius of pipe (in.), t = insulation thickness listed in this table for applicable fluid temperature and pipe size, K = conductivity of alternate material at mean rating temperature indicated for the applicable fluid temperature (Btu·in.[h·ft²·°F]); and k = the upper value of the conductivity range listed in this table for the applicable fluid temperature.

†These thicknesses are based on energy efficiency considerations only. Additional insulation is sometimes required relative to safety issues/surface temperature.

‡Piping insulation is not required between the control valve and coil on run-outs when the control valve is located within 4 ft of the coil and the pipe size is 1 in. or less.

§These thicknesses are based on energy efficiency considerations only. Issues such as water vapor permeability or surface condensation sometimes require vapor retarders or additional insulation.

***Source*:** ASHRAE/IESNA Standard 90.1-1999. Reprinted with permission.

Insulation exposed to weather shall be suitable for outdoor service, such as, protected by aluminum, sheet metal, painted canvas, or plastic cover. Cellular foam insulation shall be protected as above or with a painted coating which itself is a water retardant and also provides shielding from solar radiation.

Insulation of chilled water piping or refrigerant suction piping shall include an exterior vapor retardant covering the insulation (unless the insulation is inherently vapor retardant). All penetrations and joints of the vapor retardant shall be sealed.

7.3 VALVES, PIPE FITTINGS, AND ACCESSORIES

Types of Valve

Valves are used to regulate or stop the water flow in pipes either manually or by means of automatic control systems. Valves used in automatic control systems are called *control valves*, discussed in Chap. 5. In this section, only manually operated valves, or simply valves, are discussed.

Hand-operated valves are used to stop or isolate flow, to regulate flow, to prevent reverse flow, and to regulate water pressure. The basic construction of a valve consists of the following (see Fig. 7.6): a disk to open or close the water flow; a valve body to seat the disk and provide the flow passage; a stem to lift or rotate the disk, with a handwheel or a handle and corresponding mechanism to make the task easier; and a bonnet to enclose the valve from the top.

Based on the shape of the valve disk, the valve body, or its function, commonly used valves can be classified into the following types:

Gate Valves. The disk of a gate valve is in the shape of a "gate" or wedge, as shown in Fig. 7.6*a*. When the wedge is raised at the open position, a gate valve does not add much flow resistance. The wedge can be either a solid wedge, which is most commonly used, or a split wedge, in which two disk halves being forced outward fit tightly against the body seat. Gate valves are used either fully opened or closed, an on/off arrangement. They are often used as isolating valves for pieces of equipment or key components, such as control valves, for service during maintenance and repair.

Globe Valves. They are so named because of the globular shape of the valve body, as shown in Fig. 7.6*b*. Globe valves have a round disk or plug-type disk seated against a round port. Water flow enters under the disk. Globe valves have high flow resistances. They can be opened or closed

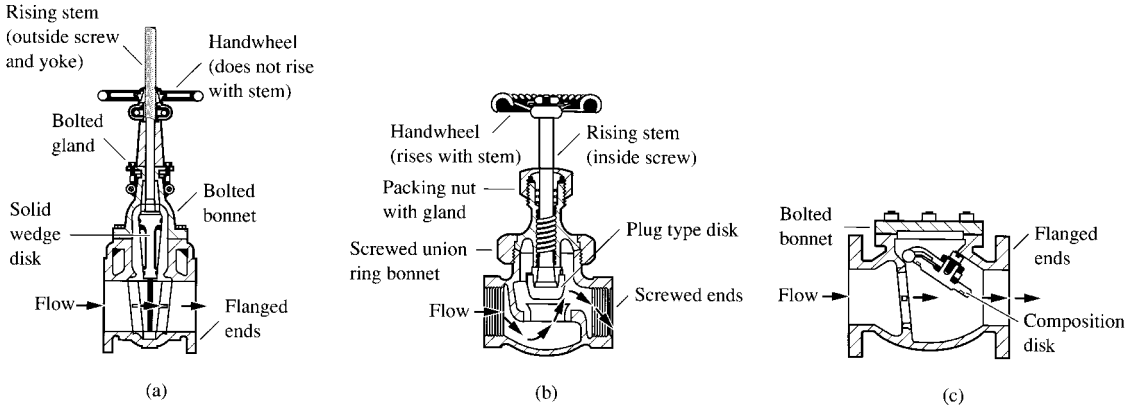


FIGURE 7.6 Types of valves. (*a*) Gate valve; (*b*) Globe valve; (*c*) Check valve, swing check.

substantially faster than gate valves. Angle valves are similar to globe valves in their seats and operation. The basic difference is that the valve body of an angle valve can also be used as a 90° elbow at that location.

Globe valves are used to throttle and to regulate the flow. They are sometimes called balancing valves. They are deliberately designed to restrict fluid flow, so they should not be used in applications for which full and unobstructed flow is often required.

Check Valves. Check valves, as their name suggests, are valves used to prevent, or check, reverse flow. There are basically two types of check valves: swing check and lift check. A swing check valve has a hinged disk, as shown in Fig. 7.6*c*. When the water flow reverses, water pressure pushes the disk and closes the valve. In a lift check valve, upward regular flow raises the disk and opens the valve, and reverse flow pushes the disk down to its seat and stops the backflow. A swing check valve has a lower flow resistance than a lift check valve.

Plug Valves. These valves use a tapered, cylindrical plug disk to fit the seat. They vary from fully open to fully closed positions within a quarter-turn. Plug valves may be used for throttling control during the balancing of a water system.

Ball Valves. These valves use a ball as the valve disk to open or close the valve. As with plug valves, they vary from fully open to fully closed positions within a quarter-turn. As with gate valves, ball valves are usually used for open/close service. They are less expensive than gate valves.

Butterfly Valves. A butterfly valve has a thin rotating disk. Like a ball or plug valve, it varies within a quarter-turn from fully open to fully closed. As described in Sec. 5.6, a butterfly valve exhibits low flow resistance when it is fully opened. The difference between a butterfly valve used for control purposes and a hand-operated butterfly valve is that the former has an actuator and can be operated automatically. Butterfly valves are lightweight, easy to operate and install, and lower in cost than gate valves. They are primarily used as fully open or fully closed, but they may be used for throttling purposes. Butterfly valves are gaining in popularity, especially in large pipes.

Balance Valves. These valves are used to balance the water flow in a water system. There are two kinds of balancing valves: manual balance valves and automatic balance valves. A globe valve can be used as a manual balance valve. A manual balance valve can also be a valve with integral pressure taps for flow measurement and a calibrated port to adjust the flow. An automatic balancing valve is also called an *automatic flow-limiting valve*. There is a moving element that adjusts the flow passage area according to the water pressure differential across the valve.

Pressure Relief Valves. These valves are safety valves used to prevent a system that is overpressurized from exceeding a predetermined limit. A pressure relief valve is held closed by a spring or rupture member and is automatically opened to relieve the water pressure when it rises above the system design working pressure.

Valve Connections and Ratings

The type of connection used between a valve and the pipes is usually consistent with the type of joint used in the pipe system. A water piping system with flanged joint requires a valve with flanged ends. The commonly used types of valve connection are as follows:

- *Threaded ends.* These connections are mainly used for small pipes with diameters from $\frac{1}{4}$ to 2 in. (6 to 50 mm). Threaded-end valves are usually inexpensive and simpler to install.

- *Flanged ends.* These connections are commonly used for larger pipes ($2\frac{1}{2}$ in. or 63 mm and above). Flanged ends are more easily separated when necessary.
- *Welded ends.* Steel valves, when used at higher pressure and temperature, are often connected with welded ends. Welded ends exhibit the fewest instances of leakage.
- *Grooved ends.* These connections use circumferential grooves in which a rubber gasket fits and are enclosed by iron couplings. Butterfly valves are often connected with grooved ends.
- *Soldered ends.* Bronze valves in copper piping systems use soldered ends. Tin-alloy soldering is the type of soldering commonly used. Lead soldering cannot be used in a potable water system because it will contaminate the water.

Valves are usually rated according to their ability to withstand pressure at a specified temperature. Metal valves have two different ratings, one for steam [working steam pressure (WSP)], which should correspond to its operating temperature, and the other for cold water, oil, or gas (WOG). The following are the commonly used ratings:

- 125 psig (862 kPa · g) WSP, 250 psig (1724 kPa · g) WOG
- 150 psig (1034 kPa · g) WSP, 300 psig (2068 kPa · g) WOG
- 300 psig (2069 kPa · g) WSP, 600 psig (4138 kPa · g) WOG

Here *psig* represents gauge pressure in pounds per square inch (kPa · g gauge pressure in kPa). As listed in Table 7.3, a wrought steel valve flange joint with a 150 psig (1034 kPa · g) rating can be used for a hot or chilled water system with a maximum allowable pressure of 250 psig (1724 kPa · g) at a temperature below 250°F (121°C), for pipes of diameter between 2.5 and 12 in. (63 and 300 mm).

Valve Materials

Valve materials are selected according to their ability to withstand working pressure and temperature, their resistance to corrosion, and their relative cost. The most commonly used materials for valves are as follows:

- *Bronze.* It has a good corrosive resistance and is easily machined, cast, or forged. Bronze is widely used for water valves up to a size of 3 in. (75 mm) because of its high cost. For valves above 3 in. (75 mm), bronze is still often used for sealing elements and stems because it is machinable and corrosion-resistant.
- *Cast iron and ductile iron.* These materials are used for pressure-containing parts, flanges, and glands in valves 2 in. (50 mm) and larger. Ductile iron has a higher tensile strength than cast iron.
- *Steel.* Forged or cast steel provides a higher tensile strength as well as toughness in the form of resistance to shock and vibration than do bronze, cast iron, and ductile iron. Steel is used in applications that require higher strength and toughness than bronze and ductile iron can provide.
- *Trim materials*. These include the elements and components that are easily worn as well as those parts that need to be resistant to corrosion, such as the disk, seating elements, and stem. Stainless steel, stellite (a kind of cobalt-chromium-tungsten alloy), and chromium-molybdenum steel are often used for trim material in valves.

Pipe Fittings and Water System Accessories

Water pipe fittings include elbows, tees, and valves. Water pipe elbows and tees are often made of cast iron, ductile iron, or steel. Pressure losses due to the water pipe fittings are usually expressed in terms of an equivalent length of straight pipe, for the sake of convenience. Table 7.6 allows calculation of the pressure losses for various types of piping fittings, in terms of an equivalent length of

TABLE 7.6 Pressure Losses for Pipe Fittings and Valves, Expressed in Terms of an Equivalent Length (in ft) of Straight Pipe

	Equivalent length, ft of pipe, for 90° elbows														
	Pipe size, in.														
Velocity, ft/s	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8	10	12
1	1.2	1.7	2.2	3.0	3.5	4.5	5.4	6.7	7.7	8.6	10.5	12.2	15.4	18.7	22.2
2	1.4	1.9	2.5	3.3	3.9	5.1	6.0	7.5	8.6	9.5	11.7	13.7	17.3	20.8	24.8
3	1.5	2.0	2.7	3.6	4.2	5.4	6.4	8.0	9.2	10.2	12.5	14.6	18.4	22.3	26.5
4	1.5	2.1	2.8	3.7	4.4	5.6	6.7	8.3	9.6	10.6	13.1	15.2	19.2	23.2	27.6
5	1.6	2.2	2.9	3.9	4.5	5.9	7.0	8.7	10.0	11.1	13.6	15.8	19.8	24.2	28.8
6	1.7	2.3	3.0	4.0	4.7	6.0	7.2	8.9	10.3	11.4	14.0	16.3	20.5	24.9	29.6
7	1.7	2.3	3.0	4.1	4.8	6.2	7.4	9.1	10.5	11.7	14.3	16.7	21.0	25.5	30.3
8	1.7	2.4	3.1	4.2	4.9	6.3	7.5	9.3	10.8	11.9	14.6	17.1	21.5	26.1	31.0
9	1.8	2.4	3.2	4.3	5.0	6.4	7.7	9.5	11.0	12.2	14.9	17.4	21.9	26.6	31.6
10	1.8	2.5	3.2	4.3	5.1	6.5	7.8	9.7	11.2	12.4	15.2	17.7	22.2	27.0	32.0

Iron and copper elbow equivalents

Fitting	Iron pipe	Copper tubing
Elbow, 90°	1.0	1.0
Elbow, 45°	0.7	0.7
Elbow, 90° long turn	0.5	0.5
Elbow, 90° welded	0.5	0.5
Reduced coupling	0.4	0.4
Open return bend	1.0	1.0
Angle radiator valve	2.0	3.0
Radiator or convector	3.0	4.0
Boiler or heater	3.0	4.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0

***Source*:** *ASHRAE Handbook 1997, Fundamentals*. Reprinted with permission.

straight pipe. The equivalent length for a fitting can be estimated by multiplying the elbow equivalent to that fitting by the equivalent length for a 90° elbow.

Water system accessories include drains, strainers, and air vents. Drains should be equipped at all low points of the system. Arrangements should be made so that a part of the system or individual components can be drained rather than draining the entire system. A condensate drain pipe is always required for cooling and dehumidifying coils. Galvanized steel is often used for this purpose. It is usually piped to a plumbing drain or other suitable location. A condensate drain pipe should be insulated so as to avoid surface condensation.

Water strainers are often installed before the pumps, control valves, or other components to protect them from dirt and impurities. Air vents are discussed in the next section.

7.4 WATER SYSTEM PRESSURIZATION AND THE PRESENCE OF AIR

Water System Pressurization Control

For an open water system, the maximum operating gauge pressure is the pressure at a specific point in the system where the positive pressure exerted by the water pumps, to overcome the pressure

drops across the equipment, components, fittings, and pipes plus the static head due to the vertical distance between the highest water level and that point, is at a maximum.

In a closed chilled or hot water system, a variation in the water temperature will cause an expansion of water that may raise the water pressure above the maximum allowable pressure. The purposes of system pressurization control for a closed water system are as follows:

- To limit the pressure of the water system to below its allowable working pressure
- To maintain a pressure higher than the minimum water pressure required to vent air
- To assist in providing a pressure higher than the net positive suction head (NPSH) at the pump suction to prevent cavitation
- To provide a point of known pressure in the system

Expansion tanks, pressure relief valves, pressure-reducing valves for makeup water, and corresponding controls are used to achieve water system pressurization control. There are two types of expansion tanks for closed water systems: open and closed.

Open Expansion Tank

An expansion tank is a device that allows for the expansion and contraction of water contained in a closed water system when the water temperature changes between two predetermined limits. Another function of an expansion tank is to provide a point of known pressure in a water system.

An open expansion tank is vented to the atmosphere and is located at least 3 ft (0.91 m) above the highest point of the water system, as shown in Fig. 7.7. Makeup water is supplied through a

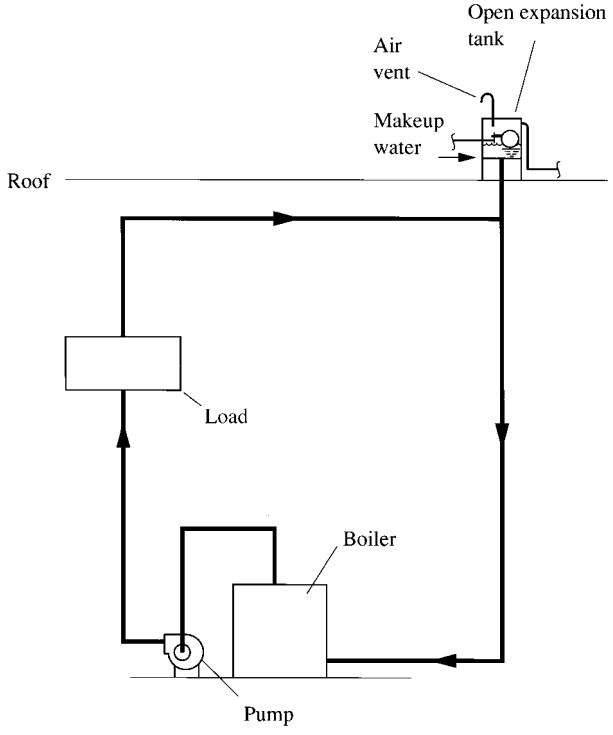


FIGURE 7.7 Open expansion tank.

float valve, and an internal overflow drain is always installed. A float valve is a globe or ball valve connected with a float ball to regulate the makeup water flow according to the liquid level in the tank. An open expansion tank is often connected to the suction side of the water pump to prevent the water pressure in the system from dropping below the atmospheric pressure. The pressure of the liquid level in the open tank is equal to the atmospheric pressure, which thus provides a reference point of known pressure to determine the water pressure at any point in the water system. The minimum tank volume should be at least 6 percent of the volume of water in the system V_s, ft^3 (m^3). An open expansion tank is simple, more stable in terms of system pressure characteristics, and low in cost. If it is installed indoors, it often needs a high ceiling. If it is installed outdoors, water must be prevented from freezing in the tank, air vent, or pipes connected to the tank when the outdoor temperature is below 32°F (0°C). Because the water surface in the tank is exposed to the atmosphere, oxygen is more easily absorbed into the water, which makes the tank less resistant to corrosion than a diaphragm tank (to be described later). Because of these disadvantages, an open expansion tank has only limited applications.

Closed Expansion Tank

A closed expansion tank is an airtight tank filled with air or other gases, as shown in Fig. 7.8. When the temperature of the water increases, the water volume expands. Excess water then enters the tank. The air in the tank is compressed, which raises the system pressure. When the water temperature drops, the water volume contracts, resulting in a reduction of the system pressure.

To reduce the amount of air dissolved in the water so as to prevent corrosion and prevent air noise, a diaphragm, or a bladder, is often installed in the closed expansion tank to separate the filled air and the water permanently. Such an expansion tank is called a diaphragm, or bladder, expansion tank. Thus, a closed expansion tank is either a plain closed expansion tank, which does not have a diaphragm to separate air and water, or a diaphragm tank.

For a water system with only one air-filled space, the junction between the closed expansion tank and the water system is a point of fixed pressure. At this point, water pressure remains constant whether or not the pump is operating because the filled air pressure depends on only the volume of water in the system. The pressure at this point can be determined according to the ideal gas law, as given by Eq. (2.1): $pv = RT_R$. The pressure in a closed expansion tank during the initial filling process or at the minimum operating pressure is called the *fill pressure* p_{fil}, psia. The fill pressure is often used as the reference pressure to determine the pressure characteristics of a water system.

Size of Diaphragm Expansion Tank

If a closed expansion tank with its filled volume of air is too small, the system pressure will easily exceed the maximum allowable pressure and cause water to discharge from the pressure relief valve, thus wasting water. If the closed tank is too large, when the water temperature drops, the system pressure may decrease to a level below the minimum allowable value and cause trouble in the air vent. Therefore, accurate sizing of a closed expansion tank is essential.

For diaphragm expansion tanks, the minimum volume of the water tank, V_t, gal (m^3), can be calculated by the following formula, recommended by *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*:

$$V_t = V_s\left(\frac{v_2/v_1 - 1 - 3\alpha(T_2 - T_1)}{1 - p_1/p_2}\right) \tag{7.3}$$

where T_1 = lower temperature, °F (°C)
T_2 = higher temperature, °F (°C)
V_s = volume of water in system, gal (m^3)
p_1 = absolute pressure at lower temperature, psia (kPa abs.)

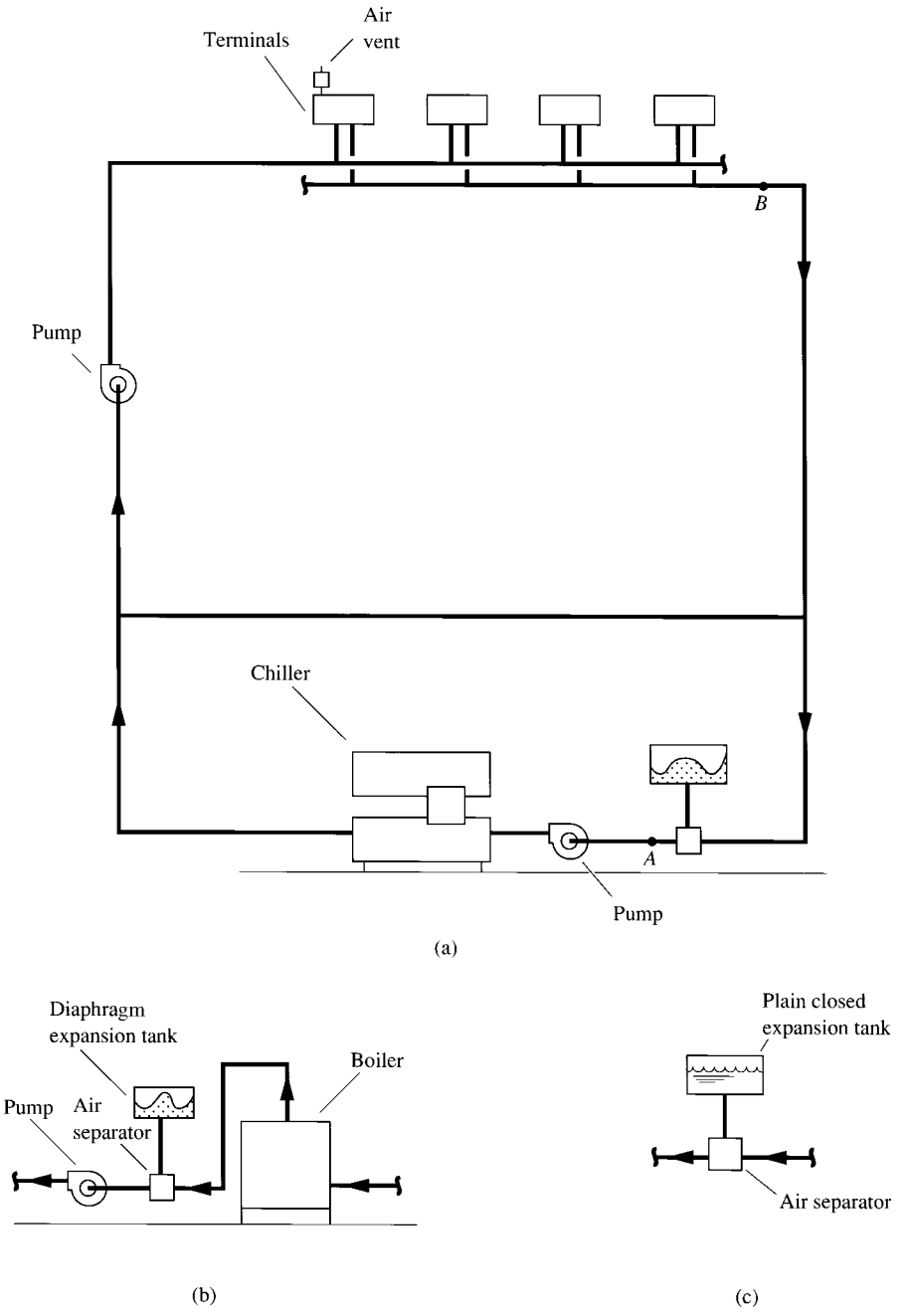


FIGURE 7.8 Closed expansion tank for a water system. (*a*) Diaphragm expansion tank in a chilled water system. (*b*) Diaphragm expansion tank in a hot water system. (*c*) Plain closed expansion tank.

p_2 = absolute pressure at higher temperature, psia (kPa abs.)
v_1, v_2 = specific volume of water at lower and higher temperature, respectively, ft^3/lb (m^3/kg)
α = linear coefficient of thermal expansion; for steel, $\alpha = 6.5 \times 10^{-6}$ in./in · °F (1.2×10^{-5} per °C); for copper, $\alpha = 9.5 \times 10^{-6}$ in./in. · °F (1.7×10^{-5} per °C)

In a chilled water system, the higher temperature T_2 is the highest anticipated ambient temperature when the chilled water system shuts down during summer. The lower temperature in a heating system is often the ambient temperature at fill conditions (for example, 50°F or 10°C).

Pump Location

The location of the pump in a water system that uses a diaphragm expansion tank should be arranged so that the pressure at any point in the water system is greater than the atmospheric pressure. In such an arrangement, air does not leak into the system, and the required net positive suction head (NPSH) can be maintained at the suction inlet of the water pump. NPSH is discussed in detail in Sec. 7.7.

A water pump location commonly used for hot water systems with diaphragm expansion tanks is just after the expansion tank and the boiler, as shown in Fig. 7.8*b*. In this arrangement, the pressure at the pump suction is the sum of the water pressure and the fill pressure. In another often-used arrangement, the diaphragm expansion tank is moved to the highest point of the water system, and the pump is still located after the boiler. In a chilled water system, the location of the chilled water pump is usually before the water chiller, and the diaphragm expansion tank is usually connected to the suction side of the water pump.

Air in Water Systems

In a closed recirculated water system, air and nitrogen are present in the following forms: dissolved in water, free air or gas bubbles, or pockets of air or gas. The behavior of air or gas dissolved in liquids is governed and described by Henry's equation. Henry's equation states that the amount of gas dissolved in a liquid at constant temperature is directly proportional to the absolute pressure of that gas acting on the liquid, or

$$x = \frac{p}{H} \tag{7.4}$$

where x = amount of dissolved gas in solution, percent by volume
p = partial pressure of that gas, psia
H = Henry's constant; changes with temperature

The lower the water temperature and the higher the total pressure of the water and dissolved gas, the greater the maximum amount of dissolved gas at that pressure and temperature.

When the dissolved air or gas in water reaches its maximum amount at that pressure and temperature, the water becomes saturated. Any excess air or gas, as well as the coexisting water vapor, can exist only in the form of free bubbles or air pockets. A water velocity greater than 1.5 ft/s (0.45 m/s) can carry air bubbles along with water. When water is in contact with air at an air-water interface, such as the filled airspace in a plain closed expansion tank, the concentration gradient causes air to diffuse into the water until the water is saturated at that pressure and temperature. An equilibrium forms between air and water within a certain time. At specific conditions, 24 h may be required to reach equilibrium.

The oxygen in air that is dissolved in water is unstable. It reacts with steel pipes to form oxides and corrosion. Therefore, after air has been dissolved in water for a long enough time, only nitrogen remains as a dissolved gas circulating with the water.

Penalties due to Presence of Air and Gas

The presence of air and gas in a water system causes the following penalties for a closed water system with a plain closed expansion tank:

- Presence of air in the terminal and heat exchanger, which reduces the heat-transfer surface
- Corrosion due to the oxygen reacting with the pipes
- Waterlogging in plain closed expansion tanks
- Unstable system pressure
- Poor pump performance due to gas bubbles
- Noise problems

There are two sources of air and gas in a water system. One is the air-water interface in a plain closed expansion tank or in an open expansion tank, and the other is the dissolved air in a city water supply.

Oxidation and Waterlogging

Consider a chilled water system that uses a plain closed expansion tank without a diaphragm, as shown in Fig. 7.8. This expansion tank is located in a basement, with a water pressure of 90 psig (620 kPa · g) and a temperature of 60°F (15.6°C) at point *A*. At such a temperature and pressure, the solubility of air in water is about 14.2 percent. The chilled water flows through the water pump, the chiller, and the riser and is supplied to the upper-level terminals. During this transport process, part of the oxygen dissolved in the water reacts with the steel pipes to form oxides and corrosion. At upper-level point *B*, the water pressure is only 10 psig (69 kPa · g) at a chilled water temperature of about 60°F (15.6°C). At this point, the solubility of air in water is only about 3.3 percent. The difference in solubility between point *A* and *B* is $14.2 - 3.3 = 10.9$ percent. This portion of air, containing a higher percentage of nitrogen because of the formation of oxides, is no longer dissolved in the chilled water, but is released from the water and forms free air, gas bubbles, or pockets. Some of the air pockets are vented through air vents at the terminals, or high points of the water system. The chilled water returns to point *A* again and absorbs air from the air-water interface in the plain closed expansion tank, creating an air solubility in water of about 14.2 percent. Of course, the actual process is more complicated because of the formation of oxides and the presence of water vapor.

Such a chilled water recirculating process causes the following problems:

- Oxidation occurs because of the reaction between dissolved oxygen and steel pipes, causing corrosion during the chilled water transport and recirculating process.
- The air pockets vented at high levels originally come from the filled air in the plain closed expansion tank; after a period of recirculation of the chilled water, part of the air charge is removed to the upper levels and vented. The tank finally waterlogs and must be charged with compressed air again. Waterlogging also results in an unstable system pressure because the amount of filled air in the plain closed expansion tank does not remain constant. Oxidation and water logging also exist in hot water systems, but the problems are not as pronounced as in a chilled water system.

Oxidation and waterlogging can be prevented or reduced by installing a diaphragm expansion tank instead of a plain closed expansion tank. Air vents, either manual or automatic, should be installed at the highest point of the water system and on coils and terminals at higher levels if a water velocity of not less than 2 ft/s (0.61 m/s) is maintained in the pipes, in order to transport the entrained air bubbles to these air vents.

In a closed chilled water system using a diaphragm expansion tank, there is no air-water interface in the tank. The 3.3 percent of dissolved air, or about 2.6 percent of dissolved nitrogen, in

water returning from point *B* to *A* cannot absorb more air again from the diaphragm tank. If there is no fresh city water supply to the water system, then after a period of water recirculation the only dissolved air in water will be the 2.6 percent nitrogen. No further oxidation occurs after the initial dissolved oxygen has reacted with the steel pipe. Waterlogging does not occur either.

Because of the above concerns, a closed water system should have a diaphragm or bladder expansion tank. An open expansion tank at high levels causes fewer problems than a plain closed expansion tank. A diaphragm tank may be smaller than an equivalent plain tank. An air eliminator or air separator is usually required for large water systems using a diaphragm tank to separate dissolved air from water when the water system is charged with a considerable amount of city water.

7.5 CORROSION AND DEPOSITS IN WATER SYSTEM

Corrosion

Corrosion is a destructive process that acts on a metal or alloy. It is caused by a chemical or electrochemical reaction of a metal. Galvanic corrosion is the result of contact between two dissimilar metals in an electrolyte. The corrosion process involves a flow of electricity between two areas of a metal surface in a solution that conducts the electric current. One area acts as the anode and releases electrons, whereas the other area acts as the cathode, which accepts electrons and forms negative ions. Corrosion, or the formation of metal ions by means of oxidation and disintegration of metal, occurs only at the anodes. In iron and steel, ferrous ions react with oxygen to form ferric hydroxide (the rust).

Moisture encourages the formation of an electrolyte, which is one of the basic elements that give rise to corrosion. Oxygen accelerates the corrosion of ferrous metals by means of a reaction with hydrogen produced at the cathode. This creates the reaction at the anode. Some alloys, such as those of stainless steel and aluminum, develop protective oxide films to prevent further corrosion.

For iron and steel, solutions such as those containing mineral acids accelerate the corrosion, and solutions such as those containing alkalies retard it. Because the corrosion reaction at the cathode depends on the concentration of hydrogen ions, the more acidic the solution, the higher the concentration of hydrogen ions and the greater the corrosion reaction. Alkaline solutions have a much higher concentration of hydroxyl ions than hydrogen, and as such the ions decrease the corrosion rate.

Water Impurities

In hot and chilled water systems, the problems associated with water mainly concern water's dissolved impurities, which cause corrosion and scale, and the control of algae, bacteria, and fungi. Typical samples of dissolved impurities in public water supplies are listed in Table 7.7.

Calcium hardness, sulfates, and silica all contribute to the formation of scale. Scale is the deposit formed by the precipitation of water-insoluble constituents on a metal surface. Chlorides cause corrosion. Iron may form deposits on a surface through precipitation. All these increase the fouling factor of water.

In addition to dissolved solids, unpurified water may contain suspended solids, which may be either organic or inorganic. Organic constituents may be in the form of colloidal solutions. At high water velocities, hard suspended solids may abrade pipes and equipment. Particles that settle at the bottom may accelerate corrosion.

In open water systems, bacteria, algae, and fungi cause many operating problems. The possibility of bacteria existing in the cooling tower and causing Legionnaires' disease necessitates microbiological control.

TABLE 7.7 Analyses of Typical Public Water Supplies

Substance	Unit	Location or area*,† (1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Silica	SiO_2	2	6	12	37	10	9	22	14	—
Iron	Fe_2	0	0	0	1	0	0	0	2	—
Calcium	Ca	6	5	36	62	92	96	3	155	400
Magnesium	Mg	1	2	8	18	34	27	2	46	1,300
Sodium	Na	2	6	7	44	8	183	215	78	11,000
Potassium	K	1	1	1	—	1	18	10	3	400
Bicarbonate	HCO_3	14	13	119	202	339	334	549	210	150
Sulfate	SO_4	10	2	22	135	84	121	11	389	2,700
Chloride	Cl	2	10	13	13	10	280	22	117	19,000
Nitrate	NO_3	1	—	0	2	13	0	1	3	—
Dissolved solids		31	66	165	426	434	983	564	948	35,000
Carbonate hardness	$CaCO_3$	12	11	98	165	287	274	8	172	125
Noncarbonate hardness	$CaSO_4$	5	7	18	40	58	54	0	295	5,900

*All values are ppm of the unit cited to nearest whole number.
†Numbers indicate location or area as follows:
(1) Catskill supply, New York City
(2) Swamp water (colored), Black Creek, Middleburg, FL
(3) Niagara River (filtered), Niagara Falls, NY
(4) Missouri River (untreated), average
(5) Well waters, public supply, Dayton, OH, 30 to 60 ft (9 to 18 m)
(6) Well water, Maywood, IL, 2090 ft
(7) Well water, Smithfield, VA, 330 ft
(8) Well water, Roswell, NM
(9) Ocean water, average
***Source*:** *ASHRAE Handbook 1987, HVAC Systems and Applications*. Reprinted with permission.

Water Treatments

Scale and Corrosion Control. One effective corrosion control method is to reduce the oxygen composition in water systems. In past years, acids and chromates were the chemical compounds commonly used to eliminate or to reduce scale and corrosion. On January 3, 1990, however, "Proposed Prohibition of Hexavalent Chromium Chemicals in Comfort Cooling Towers" was posted by the Environmental Protection Agency because chromates are suspected carcinogens and disposal problems are associated with these chemicals.

There has been a significant improvement in water treatment chemistry in recent years. Currently used chemical compounds include crystal modifiers and sequestering chemicals. Crystal modifiers cause a change in the crystal formation of scale. As a result, scale ions cannot interlace with ions of other scale-forming elements. Another important characteristic of these crystal modifiers and sequestering chemicals is that they can be applied to water systems that have a wide range of pH values. (The pH value indicates the acidity or alkalinity of a solution. It is the negative logarithm of the hydrogen ion concentration of a solution.) Even if the chemical is over- or underfed, it will not cause operating problems. Crystal modifiers and sequestering chemicals create fewer environmental problems.

Microbiological Control. The growth of bacteria, algae, and fungi is usually treated by biocides to prevent the formation of an insulating layer on the heat-transfer surface, which would promote corrosion and restrict water flow. Chlorine and chlorine compounds have been effectively and widely used. Bromine has the same oxidizing power as chlorine and is effective over a wide pH range. Biocide chemicals are detrimental to the environment if they are used in excess, however.

Biostat is a new type of chemical used in algae growth control. It prevents the algae spores from maturing, which is an approach different from that of a biocide.

Blow-down or bleed-off operation is an effective water treatment. It should be considered as important as treatments that use chemicals.

Chemical Feeding. Improper chemical feeding causes operating problems. A water treatment program with underfed chemicals results in an ineffective treatment, whereas an overfed program not only increases the operating cost but also may cause environmental problems. Generally, a continuous feeding of very small amounts of chemicals often provides effective and economical water treatment.

7.6 CLOSED WATER SYSTEM CHARACTERISTICS

System Characteristics

Closed water systems, including hot, chilled, and dual-temperature systems, can be characterized as follows:

Constant or Variable Flow. A constant-flow water system is a system for which the volume flow at any cross-sectional plane in the supply or return mains remains constant during the operating period. Three-way mixing valves are used to modulate the water flow rates to the coils. In a variable-flow system, all or part of the volume flow varies when the system load changes during the operating period. Two-way valves are used to modulate the water flow rates to the coils or terminals.

Direct Return or Reverse Return. In a direct-return water system, the various branch piping circuits, such as *ABGHA* and *ABCFGHA*, are not equal in length (see Fig. 7.9*a*). Careful balance is often required to establish the design flow rates for a building loop when a direct-return distribution loop is used as described in later sections. In a reverse-return system, the piping lengths for each branch circuit, including the main and branch pipes, are almost equal (see Fig. 7.9*b*).

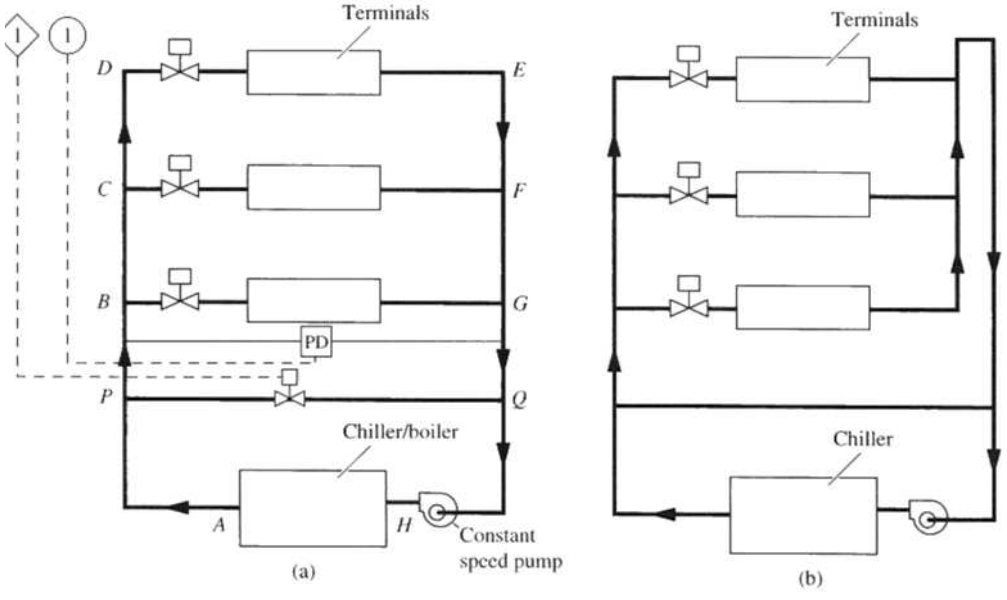


FIGURE 7.9 Direct-return and reverse–return water systems. (*a*) Direct-return; (*b*) reverse-return

Two Pipe or Four Pipe. In a dual-temperature water system, the water piping from the boiler or chiller to the coils and the terminals, or to various zones in a building, can be either a two-pipe system, with a supply main and a return main, as shown in Fig. 7.10*a*; or a four-pipe system, with a hot water supply main, a hot water return main, a chilled water supply main, and a chilled water return main, as shown in Fig. 7.10*b*. For a two-pipe system, it is impossible to heat and cool two different coils or terminals in the same zone simultaneously. Changeover from summer cooling mode operation to winter heating mode operation is required. A four-pipe system does not need changeover operation. Chilled and hot water can be supplied to the coils or terminals simultaneously. However, a four-pipe system requires a greater installation cost.

Several decades earlier, there was also a three-pipe system with a hot water supply main, a chilled water supply main, and a common return main. ASHRAE/IESNA Standard 90.1-1999 clearly specifies that hydronic systems that use a common return system for both hot water and chilled water shall not be used. This is because of the energy loss during the mixing of the hot and chilled water.

Changeover

In a dual-temperature two-pipe system, changeover refers to when the operation of one zone or the entire water system in a building changes from heating mode to cooling mode, or vice versa. During changeover, the water supplied to the terminals is changed from hot water to chilled water, or vice versa. The changeover temperature T_{co}, °F (°C), is the outdoor temperature at which the space sensible cooling load can be absorbed and removed by the combined effect of the conditioned outdoor air, the primary air, and the space transmission and infiltration loss. Such a relationship can be expressed as:

$$T_{co} = T_r - \frac{\Sigma Q_{ris} + \Sigma Q_{res} - K\dot{V}_{so}(T_r - T_{so})}{q_{tl}} \qquad (7.5)$$

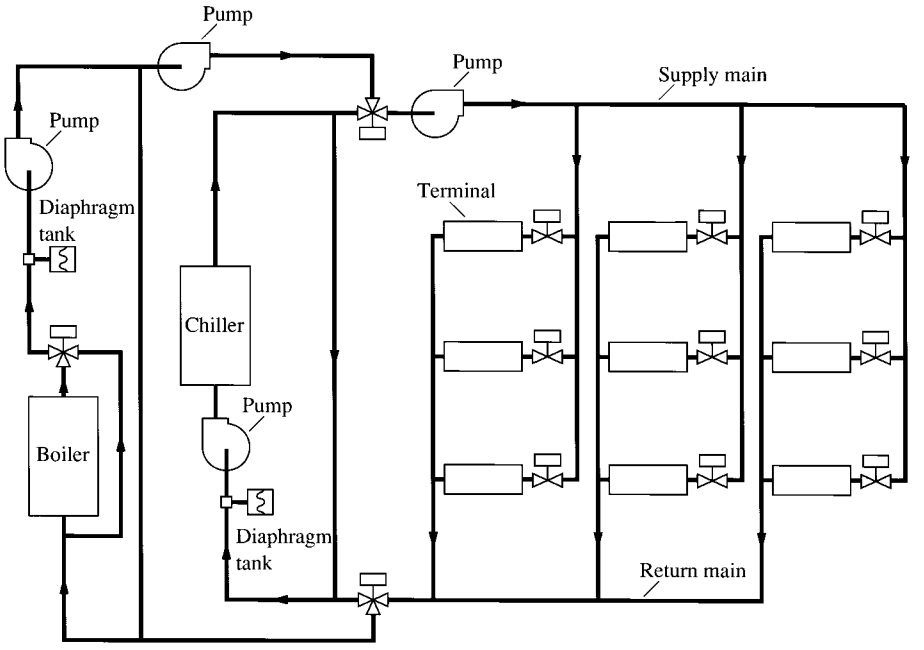
$$K = 60\rho_{so}c_{pa}$$

where T_r = space temperature, °F (°C)
ΣQ_{ris} = sum of internal sensible loads from electric lights, occupants, and appliances, Btu/h (W)
ΣQ_{res} = sum of external sensible loads through building shell, Btu/h (W)
$\dot{V}_{so}, \rho_{so}$ = volume flow rate and density of conditioned outdoor air, cfm (m^3/min) and lb/ft^3 (kg/m^3)
c_{pa} = specific heat of air, Btu/lb·°F (J/kg·°C)
T_{so} = supply temperature of outdoor air or primary air, °F (°C)
q_{tl} = transmission and infiltration losses per 1°F of outdoor-indoor temperature difference, Btu/h·°F (W/°C)

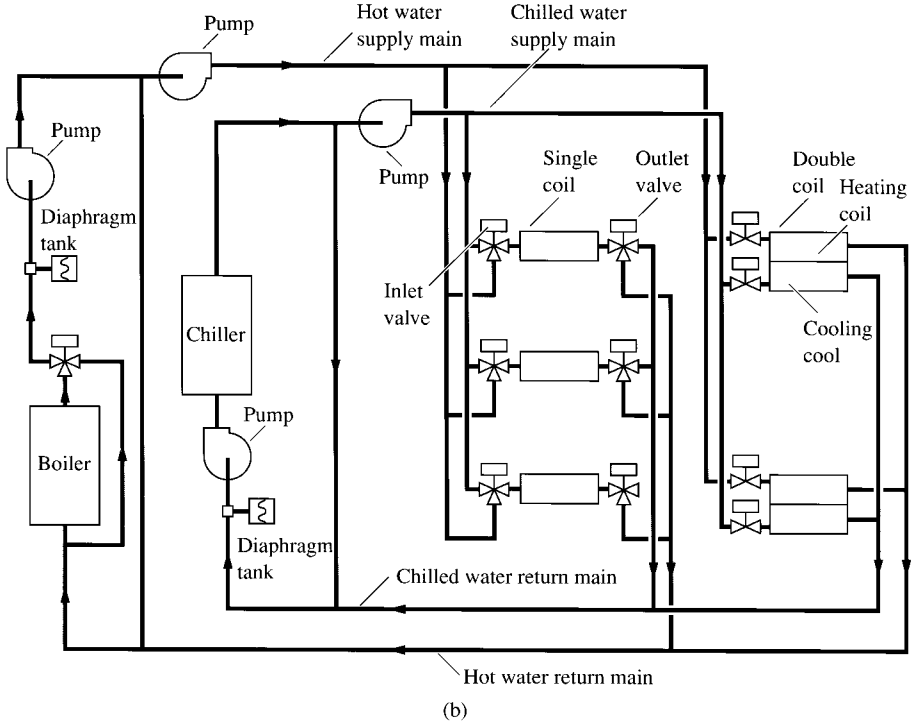
Changeover usually takes from 3 to 8 h to complete. The greater the size of the water system, the longer the changeover period. To prevent more than one changeover per day, the changeover temperature T_{co} may have a tolerance of ± 2°F (1.1°C).

Changeover may cause a sudden flow of a large amount of hot water into the chiller or of chilled water into the boiler. Such a rapid change in temperature imposes a thermal shock on the chiller or boiler and may damage the equipment. For chillers, the temperature of water entering the chiller should be no higher than 80°F (26.7°C) to prevent excessive refrigerant pressure in the evaporator. For boilers, a temperature control system bypasses most of the low-temperature water until the water temperature can be gradually increased.

Changeover may be performed either manually or automatically. Manual changeover is simple but may be inconvenient during periods when daily changeover is required. With sufficient safety controls, automatic changeover reduces the operating duties significantly. A compromise is



(a)



(b)

FIGURE 7.10 Multiple-zone, dual-temperature water systems. (*a*) Two-pipe system; (*b*) four-pipe system.

a semiautomatic changeover system in which the changeover temperature is set by a manual switch.

Outdoor reset control is often used to vary the supply water temperature T_{ws} [°F(°C)], in response to the outdoor temperature T_o for a hot water system. Typically, T_{ws} is 130°F (54.4°C) at the winter design temperature and drops linearly to 80°F (26.7°C) at the changeover temperature.

ASHRAE/IESNA Standard 90.1-1999 specifies that two-pipe changeover systems are acceptable when the following requirements are met:

- The designed deadband width between changeover from one mode to the other is of at least 15°F (8.3°C) outdoor temperature.
- System controls will allow operation in one mode for at least 4 h before changing over to another mode.
- At the changeover point, reset controls allow heating and cooling supply temperatures to be no more than 30°F (16.7°C) apart.

7.7 CENTRIFUGAL PUMPS

Centrifugal pumps are the most widely used pumps for transporting chilled water, hot water, and condenser water in HVAC&R systems because of their high efficiency and reliable operation. Centrifugal pumps accelerate liquid and convert the velocity of the liquid to static head. A typical centrifugal pump consists of an impeller rotating inside a spiral casing, a shaft, mechanical seals and bearings on both ends of the shaft, suction inlets, and a discharge outlet, as shown in Fig. 7.11. The impeller can be single-stage or multistage. The vanes of the impeller are usually backward-curved.

The pump is usually described as standard-fitted or bronze-fitted. In a standard-fitted construction, the impeller is made of gray iron, and in a bronze-fitted construction, the impeller is made of bronze. For both constructions, the shaft is made of stainless steel or alloy steel, and the casing is made of cast iron.

Three types of centrifugal pumps are often used in water systems in HVAC&R systems: double-suction horizontal split-case, frame-mounted end suction, and vertical in-line pumps, as shown in the upper part of Fig. 7.12. Double-suction horizontal split-case centrifugal pumps are the most widely used pumps in large central hydronic air conditioning systems.

Basic Terminology

Volume flow rate $\dot{V}_p$ [gpm (m^3/s)] is the capacity handled by a centrifugal pump. On a cross-sectional plane perpendicular to fluid flow in a water system, the *static head* H_s [ft (m)] is the pressure expressed in feet (meters) of water column that is exerted on the surrounding fluid and enclosure. On a cross-sectional plane, *velocity head* H_v [ft (m)] can be calculated as

$$H_v = \frac{V_o^2}{2g} \tag{7.6}$$

where v_o = velocity of water flow at pump outlet, ft/s (m/s)
g = gravitational acceleration, 32.2 ft/s^2 (9.81 m/s^2)

Total head H_t [ft (m)] is the sum of static head and velocity head, i.e.,

$$H_t = H_s + H_v \tag{7.7}$$

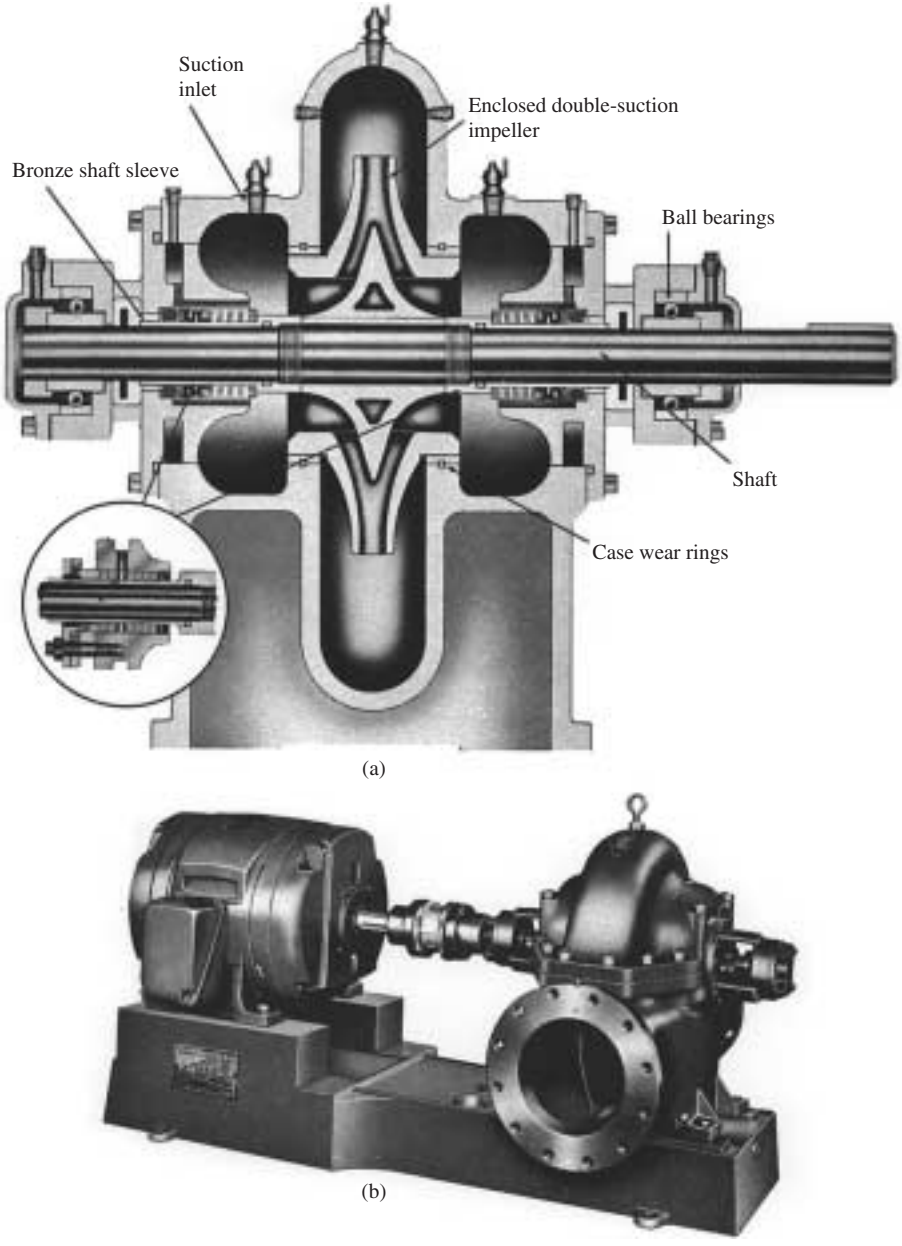


FIGURE 7.11 A double-suction, horizontal split-case, single-stage centrifugal pump. (a) Sectional view; (b) centrifugal pump and motor assembly.

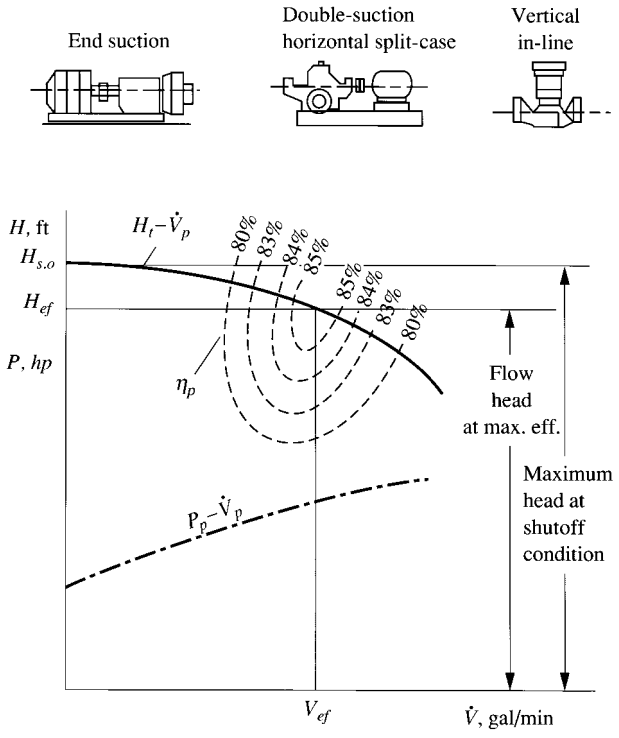


FIGURE 7.12 Performance curves for centrifugal pumps.

Net static head ΔH_s [ft (m)] is the head difference between the discharge static head H_{dis} and suction static head H_{suc}, both in feet (meter), as shown in Fig. 7.13.

Pump power P_p (hp) is the power input on the pump shaft; and *pump efficiency* η_p is the ratio of the energy output from water to the power input on the pump shaft, and both can be calculated as

$$P_p = \frac{\dot{V}_p H_t g_s}{3960 \eta_p}$$
$$\eta_p = \frac{\dot{V}_p H_t g_s}{3960 P_p} \tag{7.8}$$

where g_s = specific gravity, i.e., the ratio of the mass of liquid handled by the pump to the mass of water at 39°F (4°C).

Performance Curves

Pump performance is often illustrated by a head-capacity H_t-$\dot{V}_p$ curve and a power-capacity P_p-$\dot{V}_p$ curve, as shown in Fig. 7.12. The head-capacity curve illustrates the performance of a centrifugal pump from maximum volume flow to the shutoff point. If the total head at shutoff point H_{so} is 1.1 to 1.2 times the total head at the point of maximum efficiency H_{ef}, the pump is said to have a *flat* head-capacity curve. If $H_{so} > 1.1H_{\text{ef}}$ to $1.2H_{\text{ef}}$, it is a steep-curve pump.

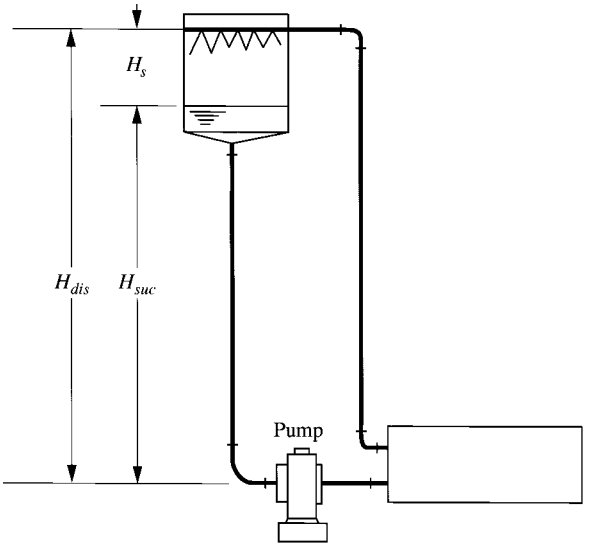


FIGURE 7.13 Net static head.

Net Positive Suction Head

The lowest absolute water pressure at the suction inlet of the centrifugal pump (shown in Fig. 7.11) must exceed the saturated vapor pressure at the corresponding water temperature. If the absolute pressure is lower than the saturated vapor pressure, the water evaporates and a vapor pocket forms between the vanes in the impeller. As the pressure increases along the water flow, the vapor pocket collapses and may damage the pump. This phenomenon is called *cavitation*.

The sum of the velocity head at the suction inlet and the head loss (due to friction and turbulence) between the suction inlet and the point of lowest pressure inside the impeller is called the *net positive suction head required* (NPSHR), in feet (meters). This factor is determined by the pump manufacturer for a given centrifugal pump. For a specific water system using a centrifugal pump, the *net positive suction head available* (NPSHA), in feet (meters), can be calculated as

$$\text{NPSHA} = H_{\text{at}} + H_{\text{suc}} - H_f - 2.31p_{\text{vap}} \tag{7.9}$$

where H_{at} = atmopheric pressure, usually expressed as 34 ft (10.3 m or 101 kPa) of water column
H_{suc} = static suction head, ft (m)
ΔH_f = head loss due to friction and dynamic losses of suction pipework and fittings, ft (m)
p_{vap} = saturated water vapor pressure corresponding to water temperature at suction inlet, psia (kPa abs.)

NPSHA must be greater than NPSHR to prevent cavitation.

Pump Selection

First, the selected pump must satisfy the volume flow and total head requirements and should operate near maximum efficiency most of the time. Second, for comfort air conditioning systems, quiet operation is an important consideration. A noise generated in a water system is very difficult to isolate and

remove. In most cases, the lowest-speed pump that can provide the required $\dot{V}_p$ and pump head ΔH_t is the quietest and often the most economical choice. Third, today a variable-speed pump is often cost-effective for a variable-flow system. However, when a constant-speed pump is used to serve a variable-flow system that operates with minor changes of head, a flat-curve pump should be selected.

7.8 PUMP-PIPING SYSTEMS

System Curve

When a pump is connected with a pipe system, it forms a pump-piping system. A water system may consist of one pump-piping system or a combination of several pump-piping systems.

The speed of a variable-speed pump in a variable-flow water system is often controlled by a pressure-differential transmitter installed at the end of the supply main, with a set point normally between 15 and 20 ft WC (4.5 and 6 m WC). This represents the head loss resulting from the control valve, pipe fittings, and pipe friction between the supply and return mains at the farthest branch circuit from the variable-speed pump. Therefore, the head losses of a pump-piping system can be divided into two parts:

- Constant, or fixed, head loss ΔH_{fix}, which remains constant as the water flow varies. Its magnitude is equal to the set point of the pressure-differential transmitter ΔH_{set}, or the head difference between the suction and the discharge levels of the pump in open systems ΔH_{sd} [ft WC (m WC)].
- Variable head loss ΔH_{var}, which varies as the water flow changes. Its magnitude is the sum of the head losses caused by pipe friction ΔH_{pipe}, pipe fittings ΔH_{fit}, equipment ΔH_{eq} (such as the pressure drop through the evaporator, condenser, and coils), and components ΔH_{cp}, all in ft WC (m WC), that is,

$$\Delta H_{\text{var}} = \Delta H_{\text{pipe}} + \Delta H_{\text{fit}} + \Delta H_{\text{eq}} + \Delta H_{\text{cp}} \tag{7.10}$$

Head losses ΔH_{fix} and ΔH_{var} are shown in Fig. 7.14. The relationship between the pressure loss Δp [ft WC (kPa)]; flow head ΔH_{var} [ft WC (m WC)]; flow resistance of the water system R_{var} [ft WC/(gpm)^2 (m WC $\cdot$ s^2/m^6)]; and water volume flow rate $\dot{V}_w$ [gpm (m^3/s)], can be expressed as

$$\begin{aligned} \Delta H_{\text{var}} &= \Delta p \frac{g_c}{\rho_w g} \\ \Delta p &= R_{\text{var}} \dot{V}_w^2 \\ \Delta H_{\text{var}} &= R'_{\text{var}} \dot{V}_w^2 \end{aligned} \tag{7.11}$$

where ρ_w = density of water, lb/ft^3 (kg/m^3)
g = gravitational acceleration, ft/s^2 (m/s^2)
g_c = dimensional constant, 32.2 lb$_m$ $\cdot$ ft/lb$_f$ $\cdot$ s^2

The curve that indicates the relationship between the flow head, flow resistance, and water volume flow rate is called the *system curve* of a pump-piping system, or a water system.

System Operating Point

The intersection of the pump performance curve and the water system curve is the system operating point of this variable-flow water system, as shown by point P in Fig. 7.14. Its volume flow rate is represented by $\dot{V}_P$ [gpm (m^3/s)], and its total head is $\Delta H_P = \Delta H_{\text{fix}} + \Delta H_{\text{var}}$ [ft WC (m WC)].

Usually, the calculated system head loss is overestimated, and the selected pump is oversized with a higher pump head, so that the actual system operation point is at point P'. Therefore, for a

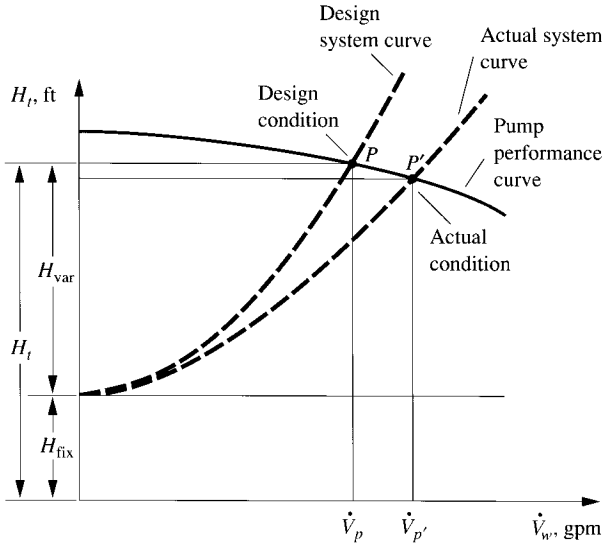


FIGURE 7.14 Water system curve and system operating point.

variable-flow water system installed with a constant-speed pump, the design system operating point is preferably located to the left of the region of pump maximum efficiency, because the system operating point of an oversized pump moves into or nearer to the region of pump maximum efficiency.

Combination of Pump-Piping Systems

When two pump-piping systems 1 and 2 are connected in series as shown in Fig. 7.15*a*, the volume flow rate of the combined pump-piping system, $\dot{V}_{com}$ [gpm (m^3/s)] is

$$\dot{V}_{com} = \dot{V}_1 = \dot{V}_2 \tag{7.12}$$

where $\dot{V}_1$ and $\dot{V}_2$ are the volume flow rate of pump-piping systems 1 and 2, gpm (m^3/s). The total head lift of the combined system ΔH_{com} [ft WC (m WC)] is

$$\Delta H_{com} = \Delta H_1 + \Delta H_2 \tag{7.13}$$

where ΔH_1 and ΔH_2 are the head of pump-piping systems 1 and 2, ft WC (m WC).

It is simpler to use one system curve to represent the whole system, i.e., to use a combined system curve. The system operating point of the combined pump-piping system is illustrated by point P with a volume flow of $\dot{V}_P$ and head of ΔH_P. The purpose of connecting pump-piping systems in series is to increase the system head.

When a pump-piping system has parallel-connected water pumps, its volume flow rate $\dot{V}$ [gpm (m^3/s)] is the sum of the volume flow rates of the constituent pumps $\dot{V}_1$, $\dot{V}_2$, etc. The head of each constituent pump and the head of the combined pump-piping system are equal. It is more convenient to draw a combined pump curve and one system curve to determine their intersection, the system operating point P, as shown in Fig. 7.15*b*. The purpose of equipping a water system with parallel-connected water pumps is to increase its volume flow rate.

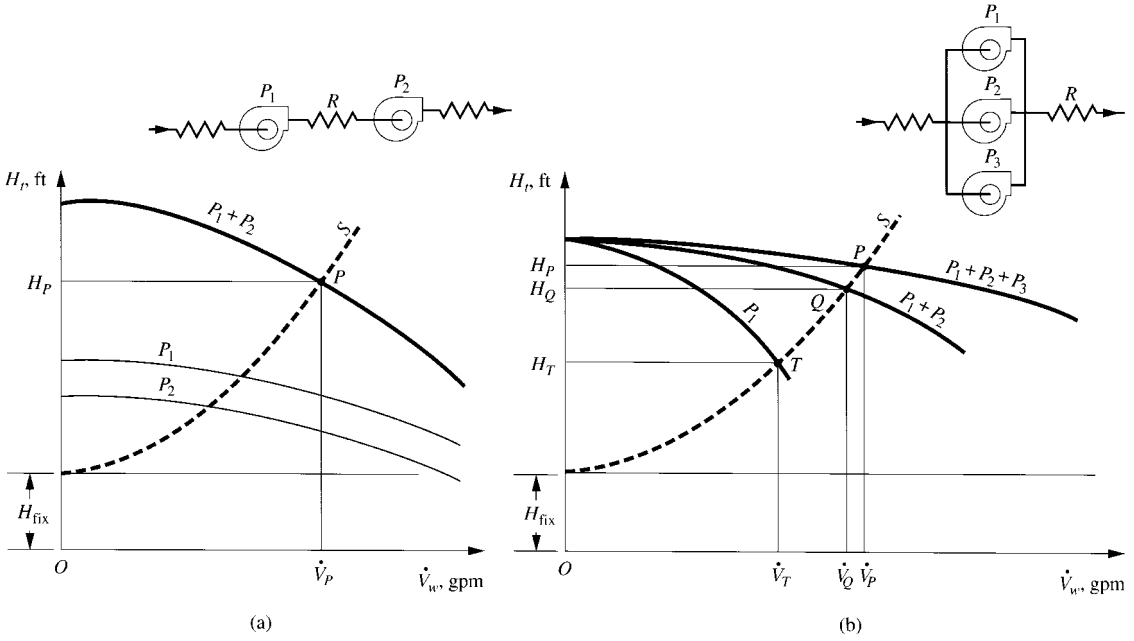


FIGURE 7.15 Combination of pump-piping systems. (*a*) Two pump-piping systems connected in series; (*b*) three parallel-connected pumps.

Modulation of Pump-Piping Systems

Modulation of the volume flow rate of a pump-piping system can be done by means of the following:

- Throttle the volume flow by using a valve. As the valve closes its opening, the flow resistance of the pump-piping system increases. A new system curve is formed, which results in having a new system operating point that moves along the pump curve to the left-hand side of the original curve, with a lower volume flow rate and higher total head, as shown in Fig. 7.16*a*. Such behavior is known as *riding on the curve*. Using the valve to modulate the volume flow rate of a pump-piping system always wastes energy because of the head loss across the valves ΔH_{val} in Fig. 7.16*a*.
- Turn water pumps on or off in sequence for pump-piping systems that have multiple pumps in a parallel connection. Modulation of the volume flow rate by means of turning water pumps on and off often results in a sudden drop or increase in volume flow rate and head, as shown by system operating points *P, Q*, and *T* in Fig. 7.16*b*.
- Vary the pump speed to modulate the volume flow and the head of a pump-piping system. When the speed of the pump is varied from n_1 to n_2 and then to n_3, new pump curves P_2 and P_3 are formed, as shown in Fig. 7.16*b*. The system operating point will move from point *P* to *Q* and then to *T* along the system curve, with a lower volume flow rate, head, and input pump power. The system curve becomes the modulating curve and approaches $\Delta H_{fix} = \Delta H_{set}$ when the volume flow rate is zero. Here ΔH_{set} is the set point of the pressure-differential transmitter, ft WC (m WC). Varying the pump speed requires the lowest pump power input in comparison with other modulation methods.

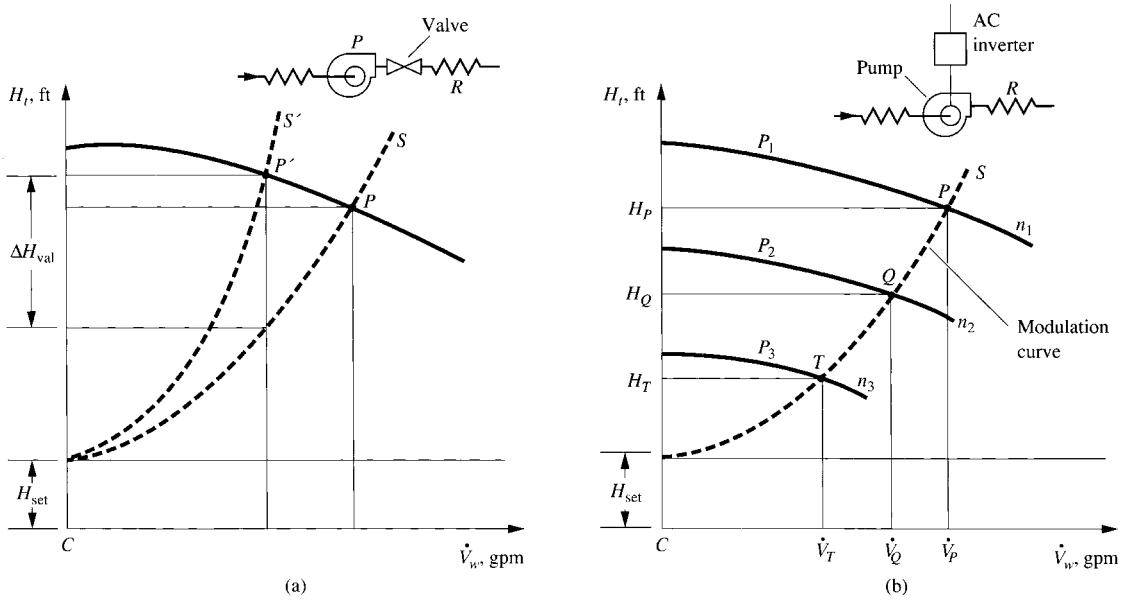


FIGURE 7.16 Modulation of pump-piping systems. (*a*) Using a valve; (*b*) varying the pump speed.

Pump Laws

The performance of geometrically and dynamically similar pump-piping systems 1 and 2 can be expressed as follows:

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{D_2^3 n_2}{D_1^3 n_1} \tag{7.14a}$$

$$\frac{\Delta H_{t2}}{\Delta H_{t1}} = \frac{n_2^2}{n_1^2} \tag{7.14b}$$

$$\frac{P_2}{P_1} = \frac{n_2^3}{n_1^3} \tag{7.14c}$$

where $\dot{V}$ = volume flow rate of pump-piping system, gpm (m^3/s)
ΔH_t = total head lift, ft WC (m WC)
P = pump power input at shaft, hp (kW)
D = outside diameter of pump impeller, ft (m)
n = speed of pump impeller, rpm

Equations (7.14*a*) through (7.14*c*) are known as the pump laws. They are similar to the fan laws and are discussed in detail in Chap. 17.

Wire-to-Water Efficiency

A pump may be directly driven by a motor, or it may be driven by a motor and belts. When the energy cost of a water system is evaluated, the pump total efficiency η_p, the motor efficiency η_{mot}, and the efficiency of the variable-speed drives η_{dr} should all be considered.

The wire-to-water efficiency of a water system η_{ww}, expressed either in dimensionless form or as a percentage, is defined as the ratio of energy output from water to the energy input to the electric wire connected to the motor. It can be calculated as

$$\eta_{ww} = \eta_p \eta_{\text{dr}} \eta_{\text{mot}} \tag{7.15}$$

The total efficiency of the centrifugal pump η_p can be obtained from the pump manufacturer or calculated from Eq. (7.8). The pump efficiency η_p depends on the type and size of pump as well as the percentage of design volume flow rate during operation. Pump efficiency usually varies from 0.7 to 0.85 at the design volume flow rate. Drive efficiency η_{dr} indicates the efficiency of a direct drive, belt drive, and various types of variable-speed drives. For direct drive, $\eta_{\text{dr}} = 1$. Among variable-speed drives, an adjustable-frequency alternating-current (ac) drive has the highest drive efficiency. For a 25-hp (18.7-kW) motor, η_{dr} often varies from 0.96 at design flow to 0.94 at 30 percent design flow to 0.80 at 20 percent design flow. Motor efficiency η_{mot} depends on the type and size of motor. It normally varies from 0.91 for a 10-hp (7.5-kW) high-efficiency motor to 0.96 for a 250-hp (187-kW) motor as listed in Table 6.2.

7.9 OPERATING CHARACTERISTICS OF CHILLED WATER SYSTEM

Many chilled and hot water systems used in commercial central hydronic air conditioning systems often have their central plant located in the basement, rooftop, or equipment floors of the building. The hot/chilled water from the boiler/chiller in the central plant is then supplied to the coils and terminals of various zones in one building or in adjacent buildings by means of supply main pipes. Water returns from the coils and terminals to the central plant via the return mains.

Coil Load and Chilled Water Volume Flow

In AHUs or fan coils, two-way control valves are currently widely used to modulate the water volume flow rate so as to maintain a predetermined air discharge temperature or space temperature at reduced system loads. Coils, especially oversized coils, operate at design load usually less than 5 percent of their total operating time. For a typical coil, nearly 60 percent of the operating time may correspond to a coil load of 35 to 65 percent of the design value.

During part-load operation, the required fraction of design volume flow rate of chilled water flowing through a coil $\dot{V}_w$ is not equal to the fraction of design sensible coil load Q_{cs} Btu/h (W) which is the sensible heat transfer from the coil to the conditioned air, as shown in Fig. 7.17*a*. In Fig. 7.17*a*, $\dot{V}_{w,d}$ indicates the design chilled water volume flow rate, gpm (m^3/min), and $Q_{cs,d}$ the design sensible coil load, Btu/h (W). This is because of the characteristics of sensible heat transfer described by

$$Q_{cs} = A_o U_o \Delta T_m \tag{7.16}$$

where A_o = outer surface area of coil, ft^2 (m^2)
U_o = overall heat-transfer coefficient based on outer area, Btu/h · ft^2 · °F (W/m^2 · °C)
ΔT_m = logarithmic temperature between conditioned air and chilled water, °F (°C)

When the volume flow rate of chilled water $\dot{V}_w$ is reduced, the decrease in the product of $A_o U_o \Delta T_m$ is not the same as the reduction in the chilled water volume flow rate $\dot{V}_w$. When $\dot{V}_w$ drops, the outer surface area A_o remains the same and U_o is slightly reduced. Only a considerable rise in chilled water temperature across the coil $\Delta T_{w,c} = T_{wl} - T_{we}$, as shown in Fig. 7.17*b*, can reduce ΔT_m suffi-

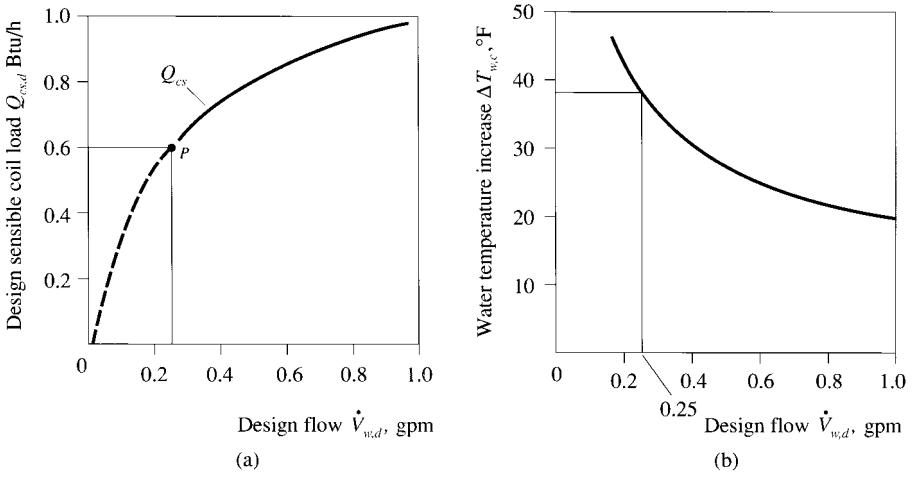


FIGURE 7.17 Relationship between fraction of design volume flow rate $\dot{V}_{w,d}$, coil load Q_{cs}, and water temperature rise $\Delta T_{w,c}$. (a) Q_{cs} versus $\dot{V}_w$; (b) $\Delta T_{w,c}$ versus $\dot{V}_{w,d}$.

ciently to match the reduction of Q_{cs}. Figure 7.17 is obtained for entering water and entering air temperatures that remain constant at various fractions of the design flow.

Theoretically, when the sensible coil load Q_{cs} is reduced to 0.6 of the design value, the chilled water volume flow rate should be decreased to about 0.25 of the design volume flow rate to match the reduction of Q_{cs}. Meanwhile, the power input at the shaft of the variable-speed pump is only about 8 percent of its design brake horsepower. There is a tremendous savings in pump power for a variable-flow system compared to a constant-flow system.

A two-way control valve for the coil must be carefully selected. First, an equal-percentage contour valve should be used. As described in Sec. 5.6 and shown in Figs. 5.16 and 7.17*a*, when a coil is equipped with an equal-percentage valve, the sensible coil load is directly related to the valve stem travel or control output signal through the percentage water flow rate and thus provides better control quality. Second, the control valve closes its opening to provide both the pressure drop for the modulated required flow and an additional pressure drop for the coils nearer to the pump for water flow balance if it is a direct-return piping system.

Chiller Plant

For chilled water systems, a central plant is often installed with multiple chillers, typically two to four chillers. Multiple chillers are usually connected in parallel. Each chiller is often installed with a chilled water pump that has the same volume flow rate as the water chiller. In such an arrangement, it is more convenient to turn the chillers on or off in sequence. ASHRAE/IESNA Standard 90.1-1999 specifies that when a chilled water plant is equipped with more than one chiller, provisions shall be made so that the chilled water flow in the chiller plant can be automatically reduced when a chiller is shut down. Chillers must provide adequate chilled water flow and cooling capacity that the AHUs and fan coils require.

Usually, a fairly constant-volume flow in the evaporator of the water chiller is preferable to avoid an extremely high temperature drop in the chiller and to prevent water from freezing at a reduced flow during part-load operation. A constant flow of chilled water in chillers is also beneficial to the capacity control of multiple chillers.

Variable Flow for Saving Energy

ASHRAE/IESNA Standard 90.1-1999 specifies that water systems having a total pump system power exceeding 10 hp (7.5 kW) shall include control valves to modulate or step open and close as a function of load and shall be designed for variable flow for energy savings. Water systems should be able to reduce system flow to 50 percent of design flow or less. Individual pumps serving variable flow systems having a pump head exceeding 100 ft (30 m) and motors exceeding 50 hp (37 kW) shall have controls and devices (such as variable speed control) that will reduce pump motor demand of no more than 30 percent of design wattage at 50 percent of design water flow.

Water Systems in Commercial Buildings

The following types of water systems are currently used in commercial buildings in the United States:

- Plant-through-building loop: bypass throttling flow
- Plant-through-building loop: distributed pumping
- Plant-building loop
- Plant-distributed pumping loop
- Plant-distribution-building loop
- Plant-distributed building loop
- Multiple sources-distributed building loop

Since the 1960s, one of the old, energy-inefficient water systems, a constant-flow system using three-way valves which has constant flow in its chiller/boiler and its supply and return mains, is gradually being replaced by a water system that is equipped with cheaper and more effective two-way valves and uses energy-efficient variable flow for distribution. Constant-flow systems using three-way valves are not discussed here.

7.10 PLANT-THROUGH-BUILDING LOOP

In a plant-through-building loop system, water is transported only by plant (chiller/boiler) pump(s) or by distributed pump(s). Plant-through-building loops can be classified into three categories: bypass throttling flow, distributed pumping, and variable flow.

Bypass Throttling Flow

A plant-through-building loop water system using bypass throttling flow is one of the older chilled/hot water systems that has been adopted in commercial buildings since the use of two-way control valves. For each chiller/boiler, a corresponding plant constant-speed water pump is equipped as shown in Fig. 7.9*a*. The chilled or hot water is supplied to the coils and terminals through the supply and return mains and branches, and is then returned to the chiller/boiler. There is a crossover bridge that connects the supply and return mains at junctions *P* and *Q*. A bypass two-way control valve is often installed on the crossover bridge. A pressure-differential transmitter and a pressure relief valve are used to maintain a set pressure differential across the supply and return mains by modulating the opening of the bypass two-way control valve when the system pressure

tends to increase during part-load operation. A portion of the water is throttled in the control valve and flows through the bypass crossover. It is then combined with water from the return main and returns to the chiller/boiler. A constant flow (or approximately constant flow) is maintained in the chiller/boiler.

A plant-through-building loop using bypass throttling control cannot save pumping energy if the set point of the pressure-differential transmitter is fixed at the value of pressure drop across the crossover bridge at design load during part-load operation. In comparison with a plant-building loop water system, it is simpler, and lower in first cost and needs a smaller pump room space. However, a plant-building loop (primary-secondary loop, which is discussed in a later section) water system saves far more pump energy than a plant-through-building loop using bypass throttling control.

Plant-through-building loop using bypass throttling control still has applications in small projects and especially in retrofit where space may not be available for a plant-building loop system.

Distributed Pumping

A plant-through-building loop water system using distributed pumping is often used for hot water systems. Hot water supplied from a boiler is divided into several distributed-piping loops, as shown in Fig. 8.7. For each of the distributed-piping loop in a control zone, there is either a variable-speed on-line hot water circulating pump or a constant-speed on-line pump with a two-way control valve, finned-tube baseboard heaters, supply and return pipes, accessories, and controls. Hot water flows through the distributed-piping loops and then returns to the boiler for heating again. The amount of hot water flowing through the boiler is reduced within a limit during part-load operation.

A space temperature sensor sends a signal to a direct digital control (DDC) unit controller and modulates the amount of hot water extracted from the supply header by means of a variable-speed on-line pump, or a two-way control valve and an on-line pump to maintain a preset zone temperature. A temperature sensor is also mounted at the water outlet of the boiler to maintain a preset hot water leaving temperature through the modulation of the burner's capacity by a DDC unit controller.

A plant-through-building loop with distributed-pumping water system is a simple, energy-efficient system.

Variable Flow

A plant-through-building loop using variable flow, as shown in Fig. 7.18, is an ideal chilled water system with variable flow in chiller, supply main, coils, return main, and variable-speed pump(s) during part-load operation. For each chiller, there is often a corresponding variable-speed pump. Chilled water is supplied to various control valves and coils in branches through supply and return mains. It is then extracted by the pump(s) and returned to the chillers for cooling again.

Three related direct digital controls are equipped for this system:

- A control of chilled water temperature leaving the chiller
- An air discharge temperature control by modulation of the control valve and thus the water flow rate entering the coil
- A pressure-differential control using pressure-differential transmitter with a DDC unit controller that modulates the variable-speed pump to maintain a preset pressure differential between supply and return mains

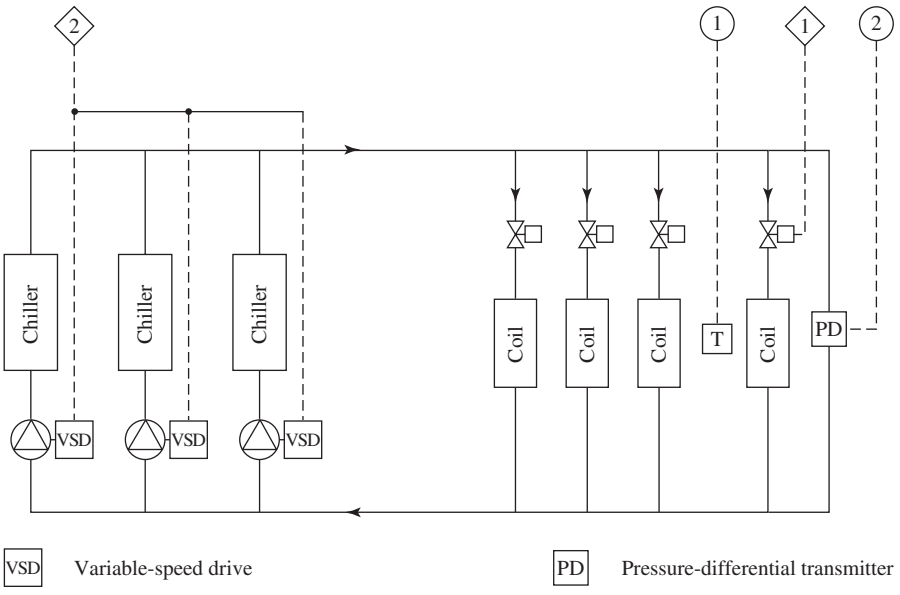


FIGURE 7.18 A chilled water system of plant-through-building loop using variable flow.

Today, although many chiller manufacturers allow a reduction of chilled water flow of 30 to 40 percent of the design volume flow rate, multiple variable-speed pumps must run at approximately the same speed at the same pump head. If $Q_{cs}/Q_{cs,d}$ drops to 0.5, the system flow ratio $\dot{V}_w/\dot{V}_{w,d}$ needs to reduce to only about 0.28 (refer to Table 7.8). This may cause operating troubles, control problems, and system instability.

Until the middle of 1997, there hasn't been a single project actually using plant-through-building loop with variable-flow chilled water system reported in HVAC&R publications in the United States and operated successfully.

TABLE 7.8 Magnitudes of $\dot{V}_{bg}/\dot{V}_{bg,d}$ and Calculated Results for Example 7.1

	$Q_{cs}/Q_{cs,d}$										
	0.3		0.4	0.5	0.6	0.65		0.7	0.8	0.9	1.0
	1 chiller	2 chillers				2 chillers	3 chillers				
$\dot{V}_{bg}/\dot{V}_{bg,d}$	0.17	0.17	0.22	0.275	0.34	0.38	0.38	0.45	0.60	0.8	1.0
$\Delta T_{w,c}$, °F	36	36	36	36	35.2	34.2	34.2	31.1	26.7	22.5	20
$\dot{V}_{pt}$, gpm	350	700	700	700	700	700	1050	1050	1050	1050	1050
$\dot{V}_{cn}$, gpm	180	530	480	425	360	320	670	600	450	250	0
T_{ee}, °F	57.5	48.8	51.3	54.2	57.1	58.6	52.4	53.3	55.3	57.1	60

7.11 PLANT-BUILDING LOOP

System Description

Plant-building loop water systems, also called primary-secondary loop (or circuit) water systems, are the widely adopted water systems for large new and retrofit commercial HVAC&R installations in the United States today. A plant-building loop chilled water, hot water, or dual-temperature water system consists of two piping loops:

- *Plant loop (primary loop).* In a plant loop, there are chiller(s)/boiler(s), circulating water pumps, diaphragm expansion tank, corresponding pipes and fittings, and control systems, as shown by loop *ABFG* in Fig. 7.19. A constant volume flow rate is maintained in the evaporator of each chiller. For a refrigeration plant equipped with multiple chillers, the chilled water volume flow rate in the plant loop will vary when a chiller and its associated chiller pump are turned on or off.
- *Building loop (secondary loop).* In a building loop, there are coils, terminals, probably variable-speed water pumps, two-way control valves and control systems, and corresponding pipes, fittings, and accessories, as shown by loop *BCDEC′F* in Fig. 7.19. The water flow in the building loop is varied as the coil load is changed from the design load to part-load.

A short common pipe, sometimes also called a bypass, connects these two loops and combines them into a plant-building loop.

Control Systems

For a plant-building loop water system, there are four related specific control systems:

Coil Discharge Air Temperature Control. A sensor and a DDC system or unit controller are used for each coil to modulate the two-way control valve and the water flow into the coil. The discharge temperature after the coil can be maintained within predetermined limits.

Water Leaving Chiller Temperature Control. In a chiller, chilled water temperature leaving the chiller is always maintained at a preset value within a specified period by varying the refrigerant flow in the chiller. In a boiler, the leaving temperature of hot water is maintained at a predetermined value by varying the fuel flow to the burner.

During part-load, the chilled water temperature leaving the chiller is reset to a higher value, such as between 3 and 10°F (1.7 and 5.6°C) according to system loads or outdoor temperature both to reduce the pressure lift between evaporating and condensing pressure and to save compressing power. ASHRAE/IESNA Standard 90.1-1999 specifies that a chilled water system with a design capacity exceeding 300,000 Btu/h (87,900 W) supplying chilled water to comfort air conditioning systems shall be equipped with controls that automatically reset supply chilled water temperature according to building loads (including return chilled water temperature) or outdoor temperature.

Staging Control. Chillers are turned on and off in sequence depending on the required system cooling capacity Q_{sc} Btu/h (W), or the sum of the coils' loads. The required system cooling capacity can be found by measuring the product of the temperature difference across the supply and return mains, as shown by temperature sensors T8 and T7 and the water volume flow rate by the flowmeter F2 in Fig. 7.19. If the produced refrigeration capacity Q_{rf}, Btu/h (W), measured by the product of chilled water supply and return temperature differential (T6 and T5) and the flowmeter (F1) is less than Q_{sc}, a DDC system controller turns on a chiller. If $Q_{rf} - Q_{sc}$ is greater than the refrigeration capacity of a chiller Q_{rfc}, the system controller turns off a chiller. Chillers should not be staged on or off based on the chilled water volume flow rate flowing through the common pipe.

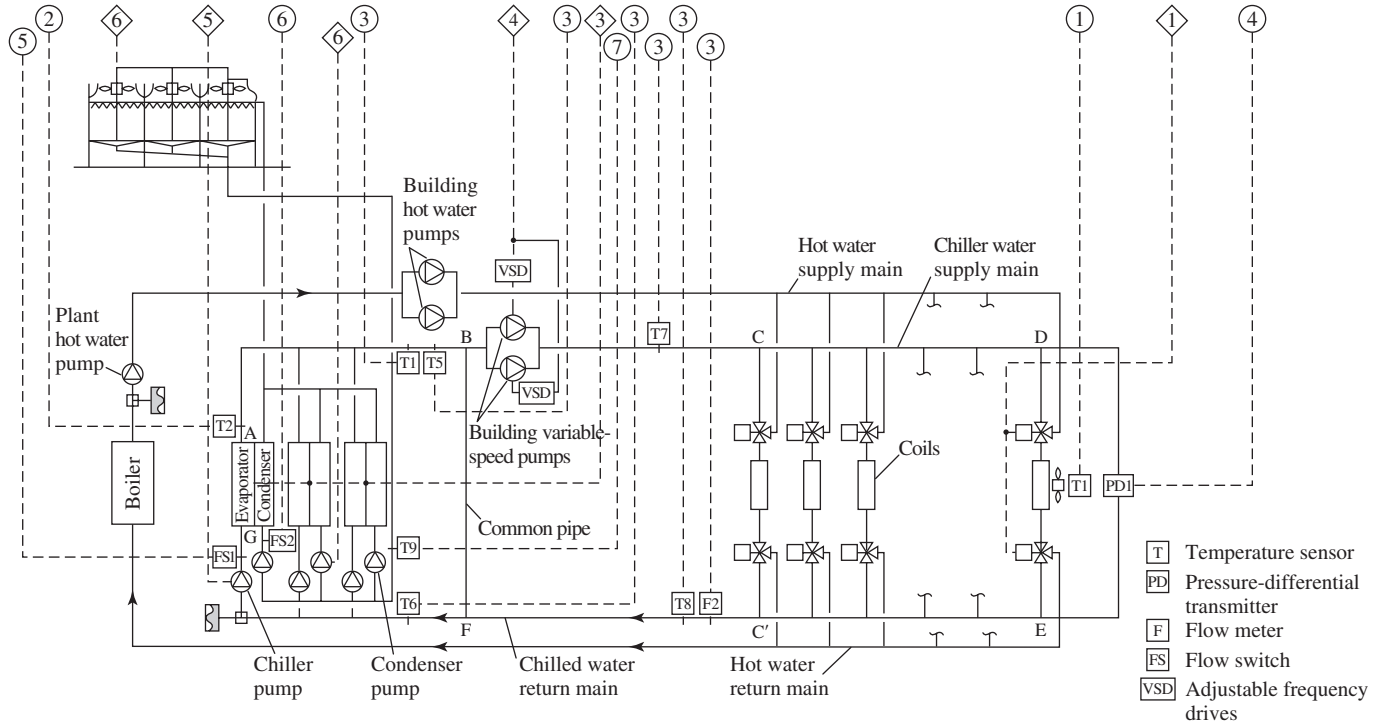


FIGURE 7.19 A dual-temperature water system with plant-building loop.

Pressure-Differential Control. These controls are used to maintain the minimum required pressure differential between the supply and return mains at a specific location, as shown by PD1 in Fig. 7.19. If only one differential-pressure transmitter is installed for chilled or hot water supply and return mains, it is usually located at the end of the supply main farthest from the building pump discharge. If multiple differential-pressure transmitters are installed, they are often located at places remote from the building pump discharge, with a low signal selector to ensure that any coil in the building loop has an adequate pressure differential between the supply and return mains.

The set point of the differential-pressure transmitter should be equal to or slightly greater than the sum of the pressure drops of the control valve, coil, pipe fittings, and piping friction of the branch circuit between the supply and return mains. A low set point cannot ensure adequate water flow through the coils. A high set point consumes more pump power at a reduced flow. A set point of 15 to 25 ft (4.5 to 7.5 m) of head loss may be suitable.

System Characteristics

For a plant-building loop chilled water system, when the volume flow rate of the chilled water in the building loop is at its design value $\dot{V}_{bg,d}$, the volume flow rate in the plant loop $\dot{V}_{pl}$ is equal to that in the building loop $\dot{V}_{bg,d}$ theoretically, all in gpm (m^3/min). In actual practice, $\dot{V}_{pl}$ is slightly (less than 3 percent) higher than $\dot{V}_{bg,d}$ to guarantee a sufficient chilled water supply to the building loop.

At design load, chilled water leaving the chiller(s) at point *A* flows through the junction of the common pipe, plant loop, and building loop (point *B*), is extracted by the variable-speed building pump; and is supplied to the coils. From the coils, chilled water returns through another junction of the building loop, common pipe, and plant loop (point *F*). There is only a very small amount of bypass chilled water in the common pipe flows in the direction from point *B* to *F*. The chilled water return from the coils is then combined with the bypass water from the common pipe and is extracted by the chiller pump(s) and enters the chiller(s) for cooling again.

When the coils' load drops during part-load operation, and the water volume flow rate $\dot{V}_{bg}$ reduces in the building loop because the control valves have been partially closed, $\dot{V}_{pl}$ is now considerably greater than $\dot{V}_{bg}$. Chilled water then divides into two flows at junction *B:* water at the reduced volume flow rate is extracted by the variable-speed building pump in the building loop and is supplied to the coils; the remaining water bypasses the building loop by flowing through the common pipe, is extracted by the chiller pump(s), and returns to the chiller(s).

For a water system that includes a plant-building loop with a common pipe between the two loops, Carlson (1968) states the following rule: One pumped circuit affects the operation of the other to a degree dependent on the flow and pressure drop in piping common to both circuits. The lower the pressure drop in the common pipe, the greater the degree of isolation between the plant and building loops. The head-volume flow characteristics of these loops act as two separate systems.

A plant-building loop has the following characteristics:

- It provides variable flow at the building loop with separate building pump(s) and constant flow through the evaporator of the chiller and thus saves pumping power during periods of reduced flow in the building loop. According to Rishel (1983), the annual pump energy consumption of a plant-building loop with variable flow in a building loop that uses a variable-speed building pump is about 35 percent that of a plant-through-building loop constant-flow system using three-way control valves.
- It separates the building loop from the plant loop and makes the design, operation, and control of both loops simpler and more stable.
- Based on the principles of continuity of mass and energy balance, if differences in the density of chilled water are ignored at junctions *B* and *F*, the sum of the volume flow rates of chilled water entering the junction must be equal to the sum of volume flow rates of water leaving that junction. Also, for an adiabatic mixing process, the total enthalpy of chilled water entering the

junction must be equal to the total enthalpy of water leaving the junction. At junction *B* or *F*, chilled water has the same water pressure and temperature.

Sequence of Operations

Consider a chilled water system in a dual-temperature water system that is in a plant-building loop, as shown in Fig. 7.19. There are three chillers in the plant loop, each of which is equipped with a constant-speed chiller pump. In the building loop, there are two variable-speed building pumps connected in parallel. One is a standby pump. Chilled water is forced through the water cooling coils in AHUs that serve various zones in the building. For simplicity, assume that the latent coil load remains constant when the coil load varies. Based on the data and information from Ellis and McKew (1996), for such a chilled water system, the sequence of operations of the DDC system is as follows:

1. When the system controller of the water system is in the off position, the chiller pump is off, condenser pump is off, building pump is off, and the cooling tower fan is off.

2. If the system controller is turned on, then the chiller's on/off switch in the unit controller is placed in the on position; and interlock signals are sent to three chiller pumps, one variable-speed building pump, and three condenser pumps and start all these pumps. The variable-speed building pump is always started from zero speed and increases gradually for safety and energy saving. As the chilled water flow swiches confirm that all the pumps are delivering sufficient water flow, the compressor of the leading chiller (first chiller) is turned on.

3. Temperature sensor T2 tends to maintain the set point of the chilled water leaving chiller temperature often at 45°F (7.2°C) by means of refrigerant flow control through multiple on/off compressors, modulation of inlet vanes, or variable-speed compressor motor (details are discussed in later chapters). Temperature sensors T7 and T8 and flowmeter F2 measure the required system cooling capacity Q_{sc}; and sensors T5 and T6 and flowmeter F1 measure the produced refrigeration capacity Q_{rf}. If $Q_{sc} > Q_{rf}$, chiller is staged on in sequence, until $Q_{rf} \geq Q_{sc}$.

4. Condenser water temperature sensor *T* measures the water temperature entering the condenser so that it will not be lower than a limit recommended by the manufacturer for normal operation.

5. When the coils' control valves in AHUs close, the chilled water flow drops below the design flow. As the pressure-differential transmittter DP1 senses that the pressure differential between chilled water supply and return mains increases to a value which exceeds the set point, such as 15 ft (4.5 m), the system controller modulates the variable-speed drive (VSD) and reduces the speed of the variable-speed pump to maintain a 15-ft (4.5-m) pressure differential.

6. At the design system load, three chillers shall provide nearly their maximum cooling capacity, and the veriable-speed building pump shall provide maximum flow through pump speed control. All two-way valves shall be nearly opened fully. A constant chilled water flow is maintained in the evaporator of each chiller. Cooling tower fan shall be continuously operated at full speed.

7. During part-load operation as the sum of the coils load (system load) decreases, the two-way valves close their openings to reduce the chilled water flowing through the coils. At a specific fraction of design sensible coil load $Q_{cs}/Q_{cs,d}$, there is a corresponding water volume flow rate in the building loop, expressed as a fraction of design flow $\dot{V}_{bg}/\dot{V}_{bg,d}$, that offsets this coil load. The building variable-speed pump should operate at this building volume flow rate $\dot{V}_{bg}$ [gpm (m^3/min)] with a head sufficient to overcome the head loss in the building loop through the modulation of the variable-speed pump.

The supply and return temperature differential of the building loop, or the mean chilled water temperature rise across the coils ΔT_{wc} [°F°C] depends on the fraction of the design sensible coil load $Q_{cs}/Q_{cs,d}$ and the fraction of the design volume flow rate through the coils $\dot{V}_{bg}/\dot{V}_{bg,d}$. The smaller the value of $\dot{V}_{bg}/\dot{V}_{bg,d}$, the greater the temperature rise $\Delta T_{w,c}$. At part load ($Q_{cs}/Q_{cs,d} < 1$), $\Delta T_{w,c}$ is always greater than that at the design load, as shown in Fig. 7.20*d*.

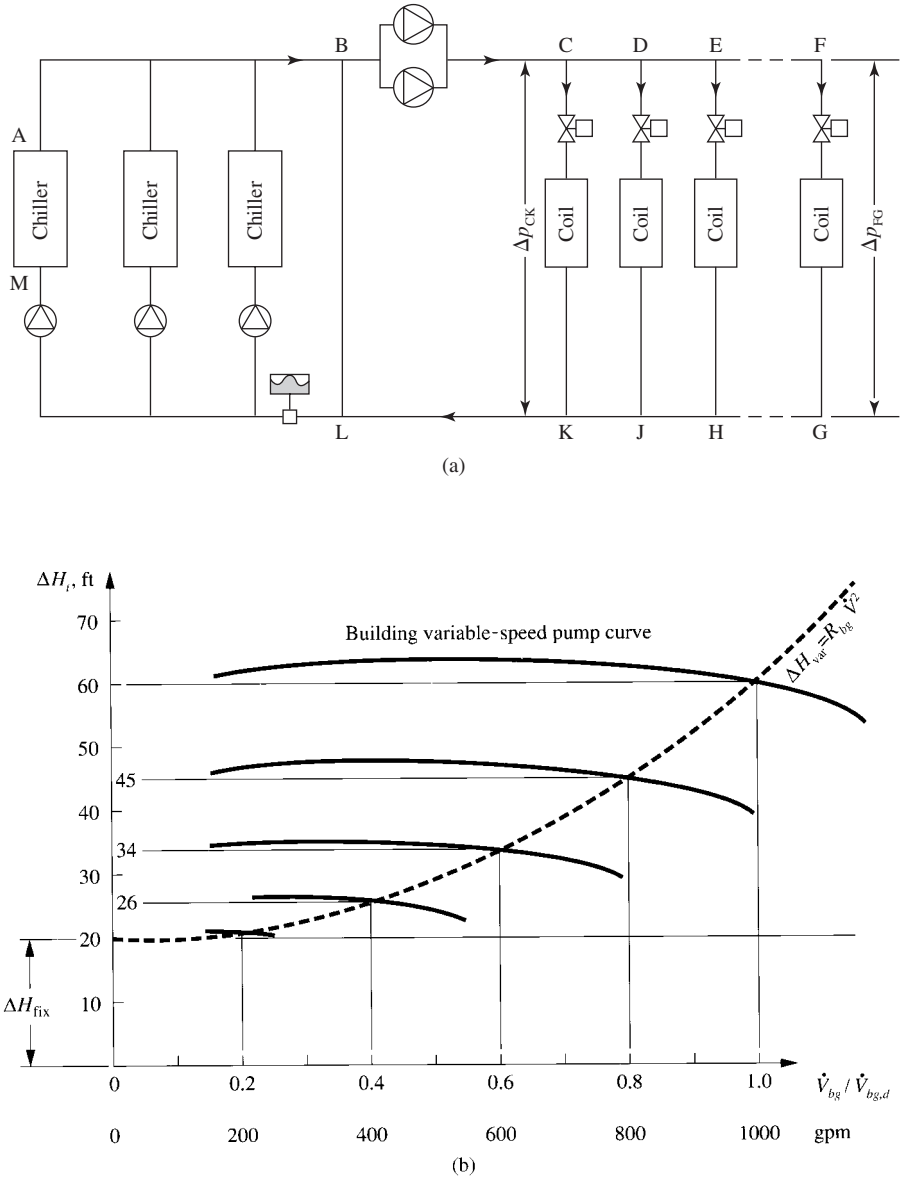
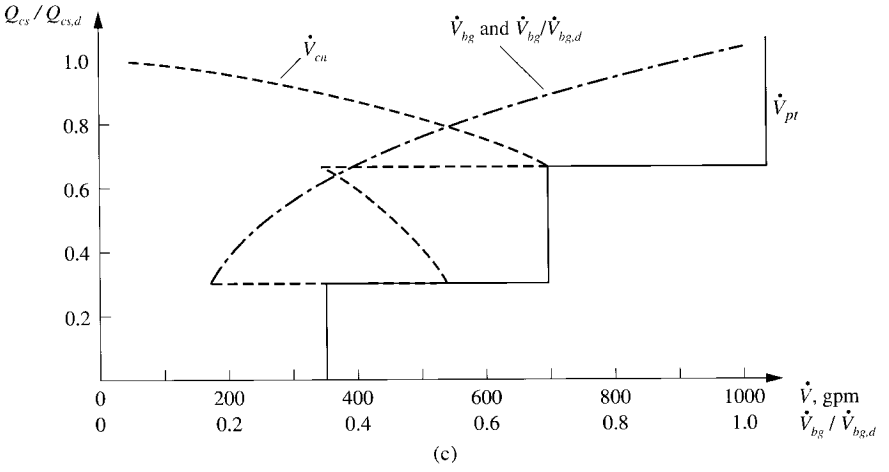
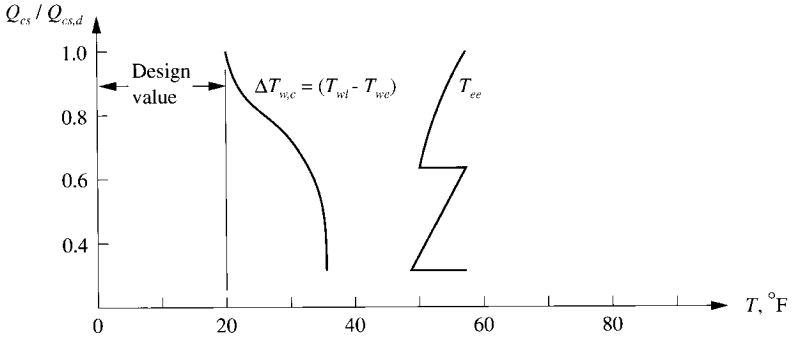


FIGURE 7.20 System performance curves for plant-building loop. (*a*) Schematic diagram; (*b*) head of building variable-speed pump at various volume flow rates; (*c*) $\dot{V}_{bg}$ versus $Q_{cs}/Q_{cs,d}$ and $\dot{V}_{bg}/\dot{V}_{bg,d}$ versus $Q_{cs}/Q_{cs,d}$; (*d*) $\Delta T_{w,c}$ versus $Q_{cs}/Q_{cs,d}$.



(c)



(d)

FIGURE 7.20 *(Continued)*

8. During part-load operation, temperature sensors T5, T6, T7, and T8 and flowmeters F1 and F2 measure the readings which give the produced cooling capacity Q_{rf} and required system cooling capacity Q_{sc}. If $Q_{rf} - Q_{sc} < Q_{rf,c}$ (one chiller's cooling capacity, Btu/h), none of the chillers is staging off. When $Q_{rf} - Q_s \geq Q_{rf,c}$, one of the chillers is then turned off until $Q_{rf} - Q_{sc} < Q_{sc,c}$. To turn off a chiller, the compressor is turned off first, then the chiller pump, condenser pump, and cooling tower fans corresponding to that chiller.

9. During part-load operation, a constant flow of chilled water is still maintained in the evaporator of each turned-on chiller. However, the volume flow rate in the plant loop $\dot{V}_{pt}$ depends on the number of operating chillers and their associated chiller pumps. The staging on or off of the chillers and their associated pump causes a variation of chilled water volume flow rate in the plant loop.

The difference between the volume flow rate of chilled water in the plant loop $\dot{V}_{pt}$ and the volume flow rate in the building loop $\dot{V}_{pt} - \dot{V}_{bg}$ gives the volume flow rate of chilled water in the com-

mon pipe $\dot{V}_{cn}$, that is, $\dot{V}_{cn} = \dot{V}_{pt} - \dot{V}_{bg}$. At part-load operation, there is always a bypass flow of chilled water from the plant loop returning to the chiller via the common pipe.

10. During part-load operation, the set point of the chilled water leaving temperature is often reset to a higher value according to either the outdoor temperature or the reduction of system load.

11. During part-load operation, as the two-way control valves close, the chilled water pressure in the supply main of the building loop tends to increase. The pressure-differential transmitter PD1 senses this increase and reduces the speed of the variable-speed building pump by means of a variable-speed drive to maintain a constant 15-ft (4.5-m) pressure differential between supply and return mains. When the system load increases, the two-way control valves open wider and PD1 senses the drop of the pressure differential, increases the speed of the pump, and still maintains a required 15-ft (4.5-m) pressure differential.

12. When the water system is shut down, the system controller should be in the off position. First, the compressor(s) are turned off, then the variable-speed building pump is off, condenser pump(s) are off, chiller pump(s) are off, and cooling tower fan(s) are off. The speed of the variable-speed pump is gradually reduced to zero first, and then the pump is turned off.

Low ΔT between Chilled Water Supply and Return Temperatures

Many chilled water systems suffer a lower actual ΔT between chilled water supply and return temperatures compared to the design value. There is also argument that a primary-secondary control scheme that depends on system flow to gauge system load is virtually blind to load variation.

First, from Eq. (7.1), $\Delta T = Q_{sc}/(500\dot{V}_{gal})$, a lower ΔT is due to an overestimated coil load (system load) Q_{sc}, or an underestimated water volume flow rate $\dot{V}_{gal}$, or both. Second, a plant-building (primary-secondary) loop should measure system load Q_{sc}, which is the product $\Delta T\dot{V}_{gal}$ and is not the only water flow to stage on and off the chillers. Third, the space load and coil load of many projects are often overestimated, the equipment is oversized, and pump head is often calculated on the safe side. All these result in a far greater actual flow and cause a low ΔT. Fourth, the cleanliness of the coils including the air-side cleanliness has an influential effect on low ΔT. Finally, for a chiller plant equipped with three chillers, if the design $\Delta T = 20°F$ (11.1°C), when one of the chillers is operated at 50 percent of the design load, ΔT will be lower, to about 16°F (8.9°C) only. A chilled water system using a plant-building loop is not an essential factor causing a low ΔT between the supply and return mains.

Variable-Speed Pumps Connected in Parallel

In a chilled water system, if two or more variable-speed pumps are connected in parallel, all the variable-speed pumps must generate the same head. The purpose of variable-speed pumps connected in parallel is to increase the volume flow rate. Their flow is additive.

For identical variable-speed pumps connected in parallel, the best overall efficiency is often obtained if the pumps are operated at identical speeds. Parallel-connected variable-speed pumps should be reduced or increased to approximately the same speed. If two parallel-connected identical variable-speed pumps are operated at different speeds, or a large pump with higher head is connected to a small pump with lower head, then it is possible that the lower-speed pump or pump with lower head sometimes may contribute negative effects since they must operate at the same head. Different speed pumps or different sized pumps are hardly operated at higher efficiency at the same time. The performance of two or more variable-speed pump-piping systems connected in parallel is further complicated in that only variable-speed pumps are connected in parallel.

Use of Balancing Valves

Equal-percentage two-way control valves are widely used to modulate the flow rate of chilled water flowing through the coils during part-load operation. Because of the lower installation cost and since often there is only limited space available inside the ceiling plenum, the direct-return piping system is often the best choice for the chilled water system in a multistory building. For a variable-flow building loop using a direct-return piping arrangement, the argument concerns whether a balance valve is necessary for each branch pipe to balance the water flow according to its requirement, such as for branches *CK*, *DJ*, *EH,* and *FG*, as shown in Fig. 7.20*a*. If there are no balancing valves installed in the branch pipes, after the control valve balances the water flow at design load, can it still effectively adjust the amount of chilled water entering the coil as required in part-load operation. This depends mainly on the type of control valve, the control mode adopted, the variation in pressure drop between various branches, and the difference in main pipe pressure drop between the farthest and the nearest branch regardless of whether equal-percentage two-way control valve with modulation control (such as proportional plus integral control) or two-way control valve with two-position on/off control (including pulse-width-adjusted two-position control) is used.

For an equal-percentage two-way control valve with modulation control for many AHUs, consider a plant-building loop in a chilled water system, as shown in Fig. 7.20*a*. At the design load, the chilled water volume flow rate through branch *FG* is 80 gpm (0.30 m^3/min), the corresponding pressure drop of its fully opened equal-percentage two-way valve is 7.5 ft WC (3.3 psi or 2.3 m WC), and the pressure drop across the farthest branch *FG* is 20 ft WC (8.7 psi or 6.1 m WC). From Eq. (5.8), the flow coefficient

$$C_v = \frac{\dot{V}}{\sqrt{\Delta p_{vv}}} = \frac{80}{\sqrt{3.3}} = 44$$

Usually, the difference between the pressure drop across the farthest branch from the building pump *FG* and the pressure drop across the nearest branch *CK* is often within 60 ft WC (26 psi or 18 m WC). At design load, for a fully opened two-way control valve in branch *CK*, even if all this 26 psi (18 m head loss) has been added, the chilled water volume flow rate is then

$$\dot{V} = C_v\sqrt{\Delta p_{vv}} = 44\sqrt{3.3 + 26} = 238 \text{ gpm (15 L/s or 0.90 m}^3\text{/min)}$$

From Fig. 5.16, for a typical equal-percentage two-way valve, when the percentage of the valve stem travel lies between 80 and 100 percent, the relationship between the percentage of full-range travel of valve stem z and the percentage of water flow rate when the valve is fully opened $\dot{V}_v$ is

$$\dot{V}_v = Ke^{kz} = 0.004e^{5.5} \tag{7.17}$$

where k = proportional constant
K = flow parameter affected by size of valve

Since $80/238 = 0.336$, substituting into Eq. (7.17) gives

$$0.336 = 0.004e^{5.5z}$$

And the percentage of full-range travel of valve stem $z = 0.805$. That is, an equal-percentage two-way control valve in branch *CK* will close its opening from 100 percent fully open to 80.5 percent for water flow balance at design load. Avery et al. (1990) emphasized that "If properly selected valves (those with equal percentage ports and with the correct actuators) are used, 20 percent or less of the stroke will be used to balance the flow. The rest of the stroke will still be available to modulate the flow within the design limits." Rishel (1997) also stated that manual balance valves and automatic balance valves should not be used on variable-volume, direct-return, modulating type, coil control valve, chilled water systems.

Therefore, an equal-percentage, two-way control valve, direct-return VAV system can balance the water flow in a direct-return chilled water system, and at least 80 percent of its stroke is still

available to perform the control actions at part-load if the variations in pressure drop of various branches are small and the difference in main pipe pressure drop between the farthest and nearest branches is within 60 ft WC (18 m WC).

Common Pipe and Thermal Contamination

For a plant-building loop in a chilled water system, if there is a backflow of a portion of the return chilled water from the building loop that enters the common pipe at junction *L* and is mixed with the supply chilled water from the plant loop at junction *B*, as shown in Fig. 7.20*a*, then the thermal contamination of building return chilled water occurs. Lizardos (1995) suggested that the length of the common pipe (expressed as the number of diameters of the common pipe) should be as follows:

Chilled water velocity in return main, ft/s (m/s)	Minimum length of common pipe
< 5 (1.5)	3 diameters, or > 2 ft (0.6 m)
> 5 (1.5)	10 diameters

The diameter of the common pipe should be equal to or greater than the diameter of the return main of the building loop.

Example 7.1. Consider a chilled water system using a plant-building loop. The design chilled water flow rate is 1000 gpm (63.1 L/s) with a chilled water temperature rise across the coils of $\Delta T_{w,c} = 20°F$ (11.1°C). There are three chillers in the central plant, each equipped with a constant-speed chiller pump that provides 350 gpm (22 L/s) at 40-ft (12-m) total head. In the building loop, there are two variable-speed building pumps, each with a volume flow rate of 1000 gpm (63.1 L/s) at 60-ft (18-m) head. One of these building pumps is a standby pump. At design conditions, chilled water leaves the chiller at a temperature $T_{el} = 40°F$ (4.4°C) and returns to the chiller at 60°F (15.6°C). Chilled water leaving the chiller is controlled at 40°F (4.4°C) for both design and part-load operation. Once the fouling and inefficiency of the coils have been taken into account, the fractions of design volume flow rate $\dot{V}_{bg}/\dot{V}_{bg,d}$ required to absorb the coil load at various fractions of the sensible load $Q_{cs}/Q_{cs,d}$ are listed in Table 7.8.

When the system load drops, plant chiller 1 will turn off when $Q_{cs}/Q_{cs.d}$ equals 0.65 and chiller 2 will turn off when $Q_{cs}/Q_{cs,d}$ equals 0.30. When the system load increases, chiller 2 turns on at $Q_{cs}/Q_{cs,d} = 0.35$, and chiller 1 turns on at $Q_{cs}/Q_{cs,d} = 0.7$. Plant chiller 3 operates continuously.

Calculate the following based on the chillers' on/off schedule at various fractions of the sensible coil load:

1. Mean chilled water temperature rise across the coil
2. Water flow in the common pipe
3. Temperature of water returning to the water chiller

Solution

1. From the given information in Table 7.8, and Eq. (7.1), the mean water temperature rise across the coil for $Q_{s,c}/Q_{sc,d} = 0.9$ is

$$\Delta T_{w,c} = 20 Q_{cs} \dot{V}_{bg,d}/Q_{cs,d}\dot{V}_{bg} = \frac{20 \times 0.9}{0.8} = 22.5°F\ (12.5°C)$$

Values of $\Delta T_{w,c}$ at other values $Q_{sc}/Q_{sc,d}$ can be similarly calculated and are listed in Table 7.8.

2. For $Q_{cs}/Q_{cs,d} = 0.9$, because all three chillers are operating, the water volume flow rate in the plant loop $\dot{V}_{pt} = 3(350) = 1050$ gpm. The water volume flow rate in the building loop is

$$\dot{V}_{\text{bg}} = \frac{1000\dot{V}_{\text{bg}}}{\dot{V}_{\text{bg},d}} = 1000(0.8) = 800 \text{ gpm } (50.5 \text{ L/s})$$

Then, the water volume flow rate in the common pipe is

$$\dot{V}_{cn} = \dot{V}_{pt} - \dot{V}_{\text{bg}} = 1050 - 800 = 250 \text{ gpm } (15.8 \text{ L/s})$$

When $Q_{cs}/Q_{cs,d} = 0.65$, chiller 1 is turned off. Just before the chiller is turned off, the volume flow rate in the common pipe is

$$\dot{V}_{cn} = 1050 - 0.38(1000) = 670 \text{ gpm } (42.3 \text{ L/s})$$

Immediately after chiller 1 is turned off

$$\dot{V}_{cn} = 2(350) - 0.38(1000) = 320 \text{ gpm } (20.2 \text{ L/s})$$

Values of $\dot{V}_{cn}$ for other values of $Q_{cs}/Q_{cs,d}$ can be similarly calculated and are listed in Table 7.8.

3. For $Q_{cs}/Q_{cs,d} = 0.9$, after the adiabatic mixing of water from the building loop and common pipe, the temperature of chilled water returning to the water chiller can be calculated as

$$T_{ee} = \frac{250}{1050}(40) + \frac{0.8(1000)(40 + 22.5)}{1050} = 57.1°\text{F } (13.9°\text{C})$$

For $Q_{cs}/Q_{cs,d} = 0.65$, just before chiller 1 is turned off,

$$T_{ee} = \frac{700}{1050}(40) + \frac{0.35(1000)(40 + 34.2)}{1050} = 51.4°\text{F } (10.8°\text{C})$$

Immediately after chiller 1 is turned off,

$$T_{ee} = \frac{350}{700}(40) + \frac{0.35(1000)(40 + 37.1)}{700} = 58.6°\text{F } (14.8°\text{C})$$

Other chilled water temperatures upon entering the chiller can be similarly calculated and are listed in Table 7.8.

7.12 PLANT-DISTRIBUTED PUMPING

A water system using plant-distributed pumping consists of two loops: a plant loop and a distributed pumping loop connected by a bypass (common pipe), as shown in Fig. 7.21. As in the plant-building loop, the plant loop comprises chiller(s)/boiler(s), constant-speed plant pumps, piping, and controls. A constant flow is maintained in the evaporator of each chiller/boiler.

In each of the distributed pumping loops connected to an associated air-handling unit, there is a corresponding variable-speed distributed pump, a coil, two isolating valves, a drain valve, and other accessories. The discharge air temperature of the AHU is controlled by the DDC unit through the modulation of the water flow rate by a variable-speed drive (VSD) and the asssociated variable-speed pump. There is no two-way control valve, and no pressure-differential transmitter is equipped to maintain a fixed pressure differential between the supply and return mains at the farthest branch.

In summer, as the AHU is operated at part-load operation for cooling, if a temperature sensor senses the discharge air temperature drops below the preset limit, a DDC unit controller modulates

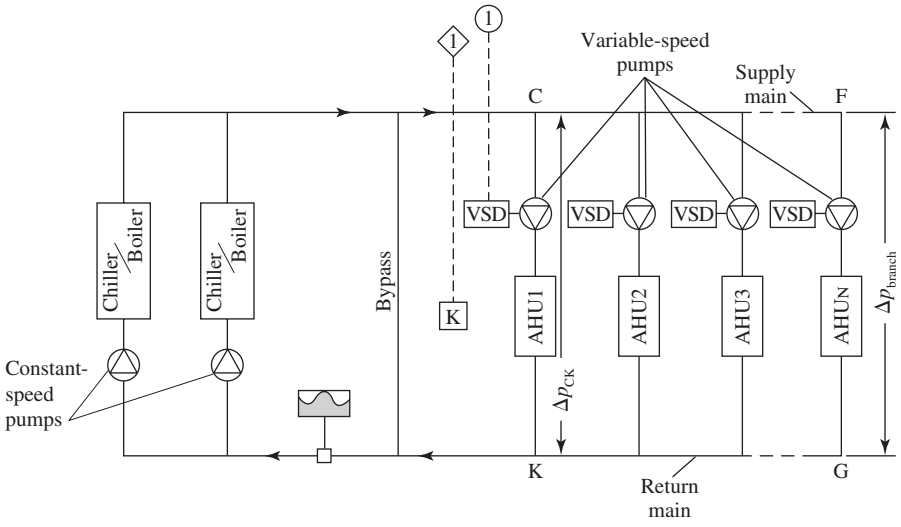


FIGURE 7.21 Schematic diagram of a plant-distributed pumping loop .

the VSD and the associated variable-speed pump to reduce the amount of chilled water flowing through the coil of the AHU, to maintain an approximately constant discharge air temperature. In the plant-distributed pumping loop, a portion of chilled water supply from the plant loop will return to the chiller by means of the bypass. During part-load operation for winter heating, the discharge air temperature increases, and the controller modulates the VSD and the variable-speed pump to reduce the amount of hot water flowing through the coil of the AHU.

Compared to a plant-building loop, a plant-distributed pumping loop has the following advantages:

- A variable-speed pump replaces the two-way control valve and balancing valve.
- No pressure-differential transmitter and control is required.
- When a distributed pumping loop is nearer to the plant (chiller/boiler), less distributing energy to transport water in the mains is needed.

The disadvantages include higher first cost and that more maintenance is required.

Distributed pumping is suitable to apply for the water system that serves large coils in AHUs and where the AHU must be installed inside a fan room to avoid inconvenient maintenance and noise.

7.13 CAMPUS-TYPE WATER SYSTEMS

Chilled water or chilled and hot water is often supplied to many buildings separated from a central plant in universities, medical centers, and airports. The benefits of using a campus-type central plant chilled water system instead of individual building installations are cost savings, minimal environmental impact (e.g., from cooling towers), effective operation and maintenance, and reliability.

The following are three types of currently used campus-type water systems: plant-distribution-building loop, plant-distributed building loop, and multiple sources-distributed building loop.

Plant-Distribution-Building Loop

System Description. Many recently developed campus-type central plant chilled water systems use a plant-distribution-building loop, as shown in Fig. 7.22*a*. As in a plant-building loop, constant flow is maintained in the evaporator of each chiller in the plant loop. Each chiller also has its own constant-speed chiller pump. Chilled water leaves the chiller at a temperature of 40 to 42°F (4.4 to 5.6°C). It is then extracted by the distribution pumps and forced to the supply main of the distribution loop.

Chilled water in the supply and return mains of the distribution loop operates under variable flow. Multiple variable-speed pumps are often used to transport chilled water at a volume flow rate slightly higher than the sum of the volume flow rates required in the building loops. At each building entrance, the variable-speed building pump extracts the chilled water and supplies it to the coils in AHUs and terminals in various zones by means of building supply mains. Chilled water is then returned to the water chillers through building return mains, a distribution-loop return main, and chiller pumps.

The system performances of the plant loop and building loop are similar to those in a plant-building loop.

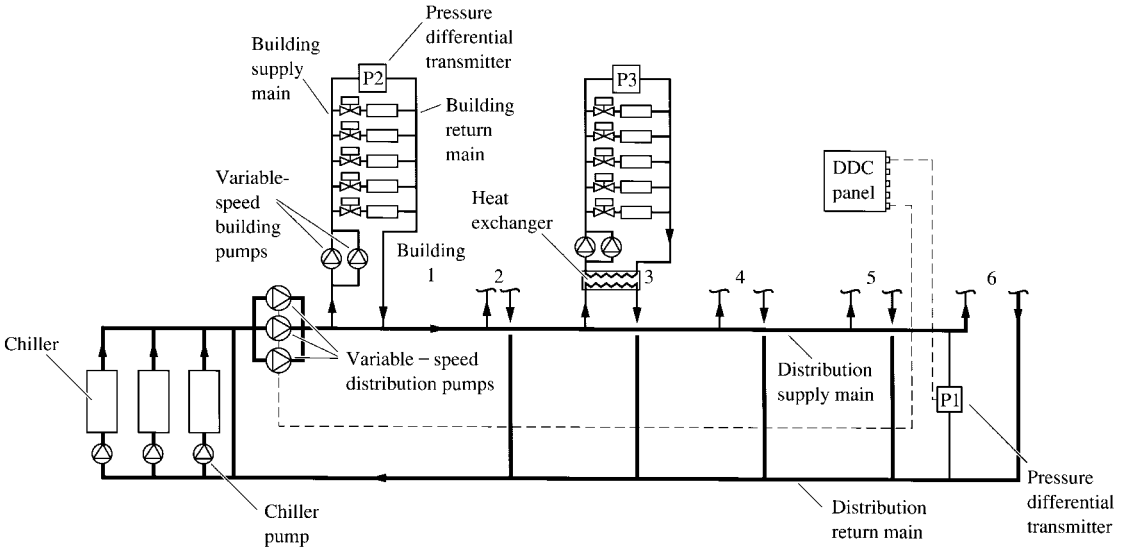
Pressure Gradient of Distribution Loop. A campus-type chilled water central plant may transport several thousand gallons of water per minute to the farthest building at a distance that may be several thousand feet away from the plant. The pressure gradient of the distribution supply and return mains due to the pipe friction and fitting losses causes uneven pressure differentials among the supply and return mains [$\Delta H_{s,r}$ (ft WC or m WC)] of buildings along the distribution loop, as shown in Fig. 7.22*b*. Buildings nearer to the central plant have a greater $\Delta H_{s,r}$ than buildings farther from the plant. Along the distribution loop, a smaller pressure gradient results in a lower pumping power but a larger diameter of chilled water pipe. A more even $\Delta H_{s,r}$ does not impose excessive pressure drop across the control valves of coils.

Using a lower pressure drop ΔH_f is an effective means of reducing the pressure gradient and pressure differential $\Delta H_{s,r}$ and saves energy. For a distribution loop, a value of ΔH_f between 0.5 and 1 ft/100 ft (0.5 and 1 m/100 m) pipe length, sometimes even lower, may be used. Low values of $\Delta H_{s,r}$ can be offset at the coil control valves without affecting the coil's proper operation when there is no building pump in the building loop. A life-cycle cost analysis should be conducted to determine the optimum ΔH_f.

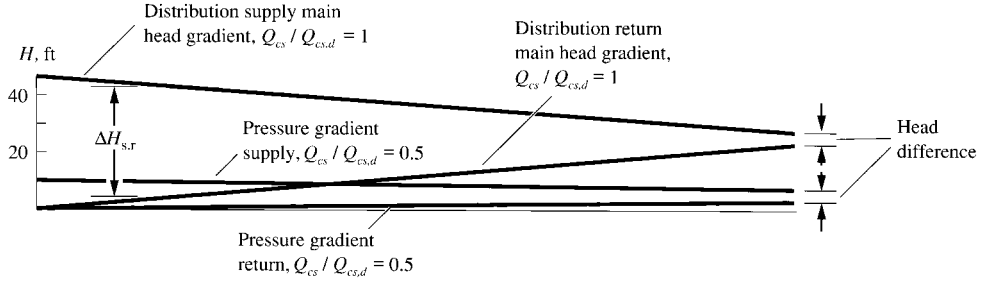
Using two-way distribution from the central plant, with two supply and return distribution loops, may reduce the pipe distance and the diameter of the supply and return mains. Such a distribution loop layout depends on the location of the central plant and the air conditioned buildings as well as the cost analyses of various alternatives.

Using a reverse-return piping arrangement instead of a direct-return one does even the pressure differentials $\Delta H_{s,r}$ along the supply and return mains of the distribution loop. However, having an additional pipe length equal to that of the return main significantly increases the piping investment. A simpler and cheaper way is to install a pressure throttling valve at each building entrance to offset the excess $\Delta H_{s,r}$. Usually, direct return is used for a distribution loop.

Variable-Speed Building Pumps. The function of a variable-speed building pump is (1) to provide variable-flow and corresponding head to overcome the pressure drop of the building loop at design and reduced coil loads and (2) to provide different magnitudes of head for the building loop according to the needs of various types of buildings. When only variable-speed distribution pump(s) are used instead of both variable-speed distribution pump(s) and variable-speed building pump(s), it must provide sufficient pump head to overcome the pressure drop of the building



(a)



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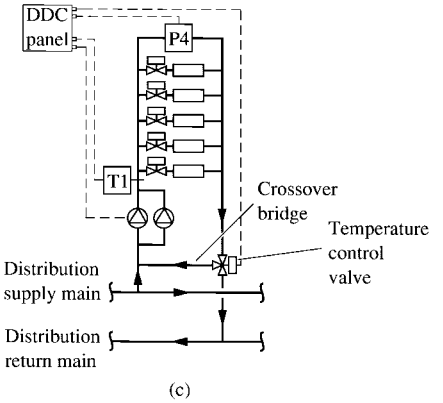

(c)

FIGURE 7.22 Chilled water system using plant-distribution-building loop. (*a*) Schematic diagram; (*b*) pressure gradient for distribution loop; (*c*) crossover bridge with temperature control valve.

loops. However, the pressure characteristics of the supply and return mains of the distribution loop at reduced flows make it difficult to satisfy various load profiles in different buildings during part-load operation. Having a variable-speed building pump for each building also saves more pump energy. Therefore, the use of variable-speed pumps for both distribution and building loops is preferable.

Control of Variable-Speed Distribution Pump. Two types of controls can be used to modulate a variable-speed distribution pump to transport the required chilled water volume flow to various buildings:

- A differential-pressure transmitter may be located near the farthest end of the distribution supply main, as shown in Fig. 7.22*a*. Theoretically, the head of the building pump should extract the exact required amount of chilled water corresponding to the sum of the coil loads in the building loop, force it through the coils, and discharge it to the distribution return main. Therefore, a set point for the pressure differential of about 5 ft (1.5 m) may be appropriate. This type of control is widely used.
- A DDC system measures the total water flow that returns from each building by means of flowmeters and modulates the variable-speed distribution pump to supply exactly the required amount to various building loops. This type of control is more precise but more expensive.

Building Entrance. Chilled water is usually supplied directly from the distribution supply main to the building supply main. A pressure throttling valve may be used to offset the excess pressure differential $\Delta H_{s,r}$ along the distribution loop.

Although using a heat exchanger at the building entrance entirely isolates the chilled water in the distribution loop from the chilled water in the building loop, a temperature increase of about 3 to 7°F (1.7 to 3.9°C) is required for a chilled water heat exchanger. Because chilled water has a supply and return temperature differential of only about 15 to 20°F (8.3 to 11.1°C), a heat exchanger is seldom used at a building or zone entrance for a chilled water system. Because a hot water system has a greater supply and return temperature differential, a heat exchanger is sometimes used at the building entrance for a hot water system.

A chilled water building loop may be divided into various zones based on different height levels within the building. In such an arrangement, the coils in the lower floors of the building loop will not suffer a high static pressure because the low-level water loop is often isolated from the high-level water loop by means of a heat exchanger at the zone entrance.

If a building requires a chilled water supply temperature higher than that given by the distribution supply main, a crossover bridge with a temperature control valve can be arranged for this purpose, as shown in Fig. 7.22*c*.

The return temperature from the coils in a building loop is affected by the cleanliness of the coil, including air-side cleanliness, and the control system in the building loop. The building's variable-speed pump is often controlled by the pressure-differential transmitter located at the end of the building supply main, as shown in Fig. 7.22*a* and *c*.

Plant-Distributed Building Loop

A plant-distributed building loop water system has nearly the same configuration as a plant-distribution-building loop system except that there is no distribution pump in the distribution loop. Constant-speed chiller/boiler pumps in the plant loop supply water to the beginning of the supply main of the distribution loop, point *S*, and extract water from the end of the return main of the distribution loop, point *R*. Various variable-speed building pumps in the building loops extract water from the distribution main point *S*. They also overcome the pressure loss of the distribution supply main piping up to the building entrance, the pressure loss of the building loop Δp_{bg} including building supply and return mains, coils, two-way control valve, and fittings; and the pressure loss of the distribution return main piping from the building outlet to point *R*.

Compared to a plant-distribution-building loop, a plant-distributed building loop has the following advantages:

- It saves the installation cost of variable-speed distribution pumps, related controls, and pump room space.
- If the pressure differential between the distribution supply and return mains at the building entrance $\Delta H_{s,r}$ is greater than the pressure loss of the building loop Δp_{bg} either at design load or at part-load, then a plant-distributed building loop can save more pumping energy.

On the other hand, a plant-distributed building loop requires variable-speed building pumps of higher head which means more attention to pump noise control in the building's mechanical room.

Multiple Sources-Distributed Building Loop

Many conversions and retrofits of campus-type chilled water systems require existing chilled water plants in addition to the developed central plant. In these cases, there are buildings with chilled water sources (chillers); buildings with chilled water coils and loads; and buildings with chilled water sources and loads connected to the same plant-distributed building loop, as shown in Fig. 7.23*a*. A

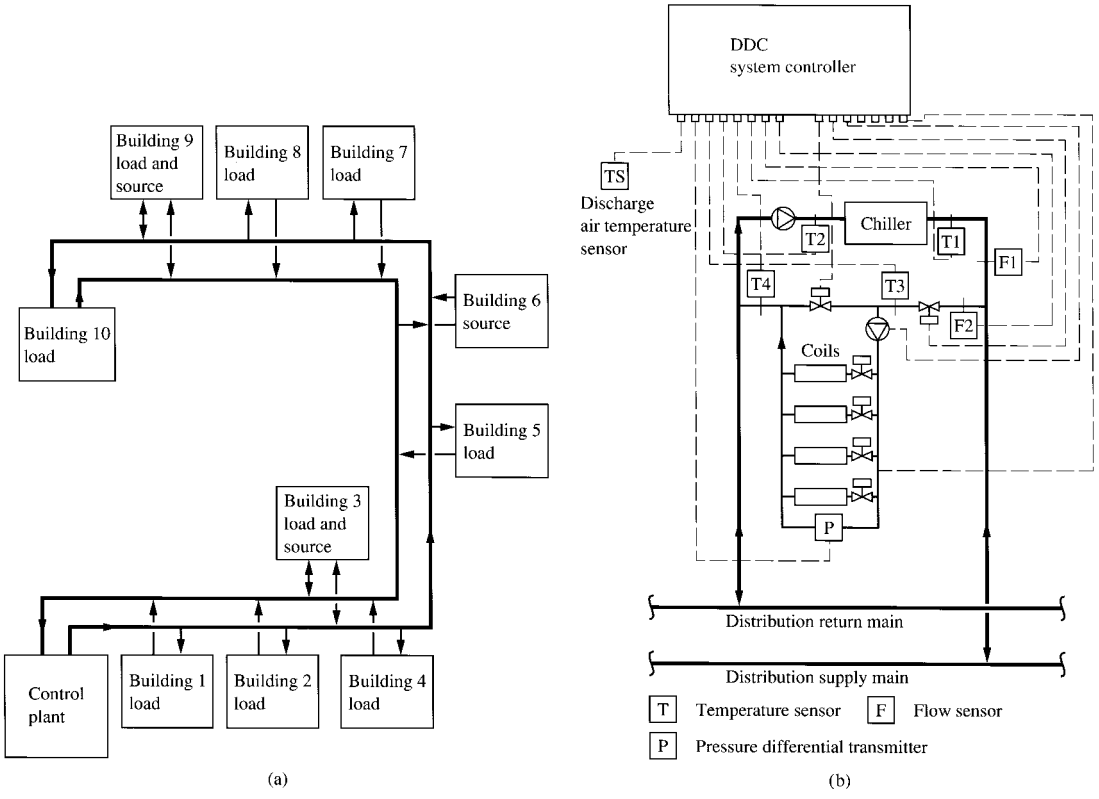

FIGURE 7.23 A multiple sources-distributed building loop. (*a*) Schematic diagram; (*b*) building with sources and loads.

DDC system controller is used for each type of building. There is a central microprocessor for the whole chilled water system. The optimization program may proceed as follows:

1. Measure the chilled water temperature across each load and source, as well as its rate of water flow.
2. Add all the loads.
3. Turn on the most efficient source first, including the demand and downtime, according to available sources. A chiller's running capacity should match the required loads. There should be a time delay to start or stop the chiller.
4. Use trending and expert system control strategies to predict load changes according to past experience and outdoor conditions.

Chilled and Hot Water Distribution Pipes

Chilled and hot water distribution pipes are large pipes mounted in underground accessible tunnels or trenches. They are well insulated, although underground return chilled water mains for which return temperature $T_{ret} > 60°F$ (15.6°C) may not be insulated, depending on a detailed cost analysis. Factory-made conduits consist of inner steel pipe, insulation, airspace, and outer conduit; or steel pipe, with and without insulation, and outer casing. Expansion loops or couplings should be included, and a good drainage system is important to protect the insulating quality. Please refer to *ASHRAE Handbook 1996, HVAC Systems and Equipment*, chapter 11, for details on design and installation.

7.14 COMPUTER-AIDED PIPING DESIGN AND DRAFTING

General Information

According to "Selecting Piping System Software" by Amistadi (1994), three currently widely used piping design and drafting computer programs were reviewed: University of Kentucky's KYCAD/KYPIPE, Trane's Water Piping Design, and Softdesk's Piping. The Trane and Softdesk computer programs require AutoCAD as a base product whereas the University of Kentucky software has its own integrated CAD system.

The University of Kentucky programs have extensive hydraulic modeling capacities including transient analysis. They are intended for mechanical engineers and are used to design and analyze large, complex water systems. Trane's and Softdesk's software programs are limited to steady-state imcompressible fluids. They are intended for contractors, mechanical engineers, and drafters. Kentucky's and Softdesk's programs support both inch-pound and metric units, while Trane's package supports only inch-pound units. The University of Kentucky software requires 2.0 Mbytes of disk space and 2.0 Mbytes RAM. Softdesk's software requires 10 Mbytes disk space and 8 Mbytes RAM. Trane's software requires the most disk space—10 Mbytes and 12 Mbytes RAM. The University of Kentucky program is MS-DOS applications, whereas Trane's and Softdesk's are available for Windows platforms.

Computer-Aided Drafting Capabilities

Trane's piping software allows designers to create a schematic two-dimensional (2D) piping system in AutoCAD and to link to the computer programs of piping size calculations for use in piping system design. Softdesk's software is intended for drafting in 2D or 3D graphics. Design information databases and engineering analysis software programs are linked to the drawing for support. The

University of Kentucky program is aimed for the design and diagnosis of large, complicated water systems.

Softdesk's software provides all the graphical components, such as pipes, valves, fittings, equipment, pumps, tanks, controls, and structural components to form a piping system. Software by Trane and the University of Kentucky only provides components that are needed for piping sizing or hydraulic analysis. The University of Kentucky software provides annotation options for all hydraulic parameters. Softdesk does not offer all, and Trane provides only selections of annotation.

Trane and the University of Kentucky only offer schematic layout and symbols to represent the piping system and components and provide plan view engineering drawings as the standard option. Softdesk offers schematic, double-line, and 3D model with wire-frame 3D version of its components. Softdesk is the only computer program to translate automatically from single-line to 3D model. The translation is bidirectional. It also allows the translation to go from 3D wire-frame to double- or single-line. Softdesk offers a full range of engineering drawings including plan, section, elevation, isometric, and perspective views.

Softdesk's and University of Kentucky's software recognize most of the types of layouts, such as series, parallel, branching, and network. The check of continuity is the basis of the hydraulic calculation, which begins with the system continuity. The Softdesk software provides the most complete checking of graphical elements of the piping system including gaps, overlaps, pump location, and size compatibility of adjacent sections, followed by the University of Kentucky and Trane programs.

Computer-Aided Design Capabilities

System Size. The University of Kentucky software is for large piping systems and supports a system up to 1000 legs. The Trane and Softdesk software support 400 legs.

Pipe Sizing. Trane's and Softdesk's computer programs allow the designer many options based on maximum head loss, such as 2.5 ft/100 ft (2.5 m/100 m) or velocity for pipe sizing. The University of Kentucky computer program has extensive constraint capabilities that are linked to meet the pressure at given node(s). A node is the junction of pipes and is the place where the flow rate changes.

Pump and System Operations. Trane's and Softdesk's programs allow the designer to first set the system water flow rate and then calculate the pressure drop and the flow of the system components. The University of Kentucky software is able to determine system operating points for series, parallel, plant-building loop (primary and secondary), and variable-speed pumping applications by means of pump and system curves. The University of Kentucky software also has the capabilities to set control and pressure-regulating valves, or to locate check valves which affect the system hydraulic calculations.

Pressure Losses and Network Technique. All three programs use the Darcy-Weisbach equation with empirical fits to the Moody diagram to calculate pipe frictional losses. Unique pipe roughness is used in Trane's and Softdesk's calculations, but it can be varied for each pipe in the University of Kentucky software. Trane's and Softdesk's software uses equivalent length and C_v to calculate dynamic losses for pipe fittings and control valves, whereas the University of Kentucky software adopts the local loss coefficient k method. Trane's and Softdesk's software read the drawing and automatically places the node at points where there is a change in flow. The University of Kentucky software expects the designer to input the nodes as fittings. All automatically number the nodes and edit them if necessary.

Regarding piping network technique, Trane's and Softdesk's software programs adopt sequential method, a once-through stepwise approach, and use arithmetic sum of flow in parallel circuits to determine the combined flow and the component head loss. The University of Kentucky program uses the simultaneous method by solving simultaneous algebraic-equations through successive approximation.

Input Data and Reports. Trane's software relies on the AutoCAD attribute functions. Its data are stored with the drawing data. Softdesk's software uses extended entity capabilities, and data are stored in external AutoCAD's drawing exchange format (DXF) files. The University of Kentucky software has a dedicated CAD system. Trane's and Softdesk's software programs let the designer select a kind of fluid and temperature, and the computer program calculates the fluid properties. The University of Kentucky software requires the designer to input the data each time. Trane's and Softdesk's databases include size, cost, and hydraulic data, whereas the University of Kentucky only includes hydraulic data.

All three computer programs provide tabular reports of pipe diameter, length, flow, velocity, and head. Trane's software identifies the critical path, and Trane's and Softdesk's software programs offer quantity and cost bill of material part, whereas the University of Kentucky software offers multiple design condition reports, such as cavitation and metering reports.

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CHAPTER 8
HEATING SYSTEMS, FURNACES, AND BOILERS

8.1 HEATING SYSTEMS 8.1
Selection of a Heating System 8.2
8.2 WARM AIR FURNACES 8.3
Types of Warm Air Furnace 8.3
Upflow Gas-Fired Furnace 8.3
Horizontal Gas-Fired Furnace 8.6
Furnace Performance Factors 8.6
Saving Energy 8.7
Control and Operation 8.8
8.3 HOT WATER BOILERS 8.9
Selection of Fuel 8.9
Types of Hot Water Boiler 8.10
Fire-Tube Boiler 8.10
Scotch Marine Packaged Boiler 8.10
Cast-Iron Sectional Boiler 8.12
Gas and Oil Burners 8.13
Condensing and Noncondensing Boilers 8.13
Boiler Efficiency 8.13
Chimney, or Stack 8.14
Operation and Safety Controls 8.14
8.4 ELECTRIC FURNACES, HEATERS, AND BOILERS 8.15
Electric Heating Fundamentals 8.15
Electric Furnaces, Electric Heaters, and Duct Heaters 8.16
Electric Hot Water Boilers 8.17
8.5 LOW-PRESSURE DUCTED WARM AIR HEATING SYSTEMS 8.17
System Characteristics 8.17
Types of Low-Pressure Ducted Warm Air Heating System 8.18
Heat Supplied to Conditioned Space 8.18
Duct Efficiency and System Efficiency for Heating 8.20
Location of Furnace and Duct Insulation 8.20
Duct Leakage 8.20
Thermal Stratification 8.21
Part-Load Operation and Control 8.21
8.6 DUCTED WARM AIR HEATING SYSTEMS 8.22
8.7 HOT WATER HEATING SYSTEMS USING FINNED-TUBE HEATERS 8.23
Types of Hot Water Heating System 8.23
Two-Pipe Individual Loop System 8.23
Finned-Tube Heaters 8.24
Design Considerations 8.25
Part-Load Operation and Control 8.26
8.8 HYDRONIC RADIANT FLOOR-PANEL HEATING SYSTEMS 8.27
System Description 8.27
Radiant Floor Panel 8.27
Thermal Characteristics of Floor Panel 8.28
Design Nomograph 8.30
Design Considerations 8.30
Control and Operations of Multizone Hydronic Radiant Floor-Panel Heating System 8.30
System Characteristics and Applications 8.31
8.9 INFRARED HEATING 8.31
Basics 8.31
Gas Infrared Heaters 8.32
Electric Infrared Heaters 8.32
Design and Layout 8.33
REFERENCES 8.35

8.1 HEATING SYSTEMS

According to the data in DOE/EIA Commercial Buildings Consumption and Expenditures 1995 and Household Energy Consumption and Expenditures 1993 for the 54.3 billion ft^2 (5.05 billion m^2) of heated commercial buildings and the 96.6 million households in the United States, the percentage of use of various types of heating systems may be estimated as follows:

	Commercial, percent	Residential, percent
Heating systems using boilers	28	33
Warm air furnaces and packaged heating units	24	37
Heat pumps	10	30
Individual space heaters including electric, gas, and radiant heaters	28	
District heating	10	

In commercial buildings, the use of a boiler as a primary heat source had a share of about 28 percent to supply hot water or steam to the heating coils, radiators, baseboard heaters, and heating panels. Heating systems in new constructed commercial projects that use boilers are mainly hot water systems because these are energy-efficient, simpler to operate, and easier to maintain than steam heating systems. In new constructed residential buildings, the heat sources are mainly gas-fired warm air furnaces and heat pumps.

Heating systems that use finned coils to heat the air, including water heating coils, electric heating coils, steam coils, or condensing heating coils of heat pumps in air-handling units and packaged units, are convective or warm air heating systems. Heating systems that use high-temperature infrared heaters, radiant panels, or radiators in which emited radiant heat exceeds the released convective heat are radiant heating systems.

Selection of a Heating System

Several factors must be considered prior to selection of a suitable heating system. They include the following:

- Whether it is a separate heating system or part of the air conditioning system
- An open or enclosed space, or a space with high infiltration
- Size of heating system (small, medium, or large)
- Available existing heat source, such as hot water or steam
- Cost of gas, oil, or electricity
- Design criteria and local customs

In both commercial and residential buildings, a warm air furnace is generally linked with unitary packaged air conditioning systems and, therefore, packaged units. In buildings with rooftop packaged units, a warm air heating system using a warm air furnace is often the most direct, economical, and suitable choice. In locations where the outdoor climate is mild in winter, the heat pump is often a convenient and energy-efficient heatsource. For buildings with central hydronic air conditioning systems, a hot water heating system using a boiler is often a suitable choice because the central primary plant and boilers are usually far away from the air-handling units and conditioned space. According to DOE/EIA 0318 (1998), heat pumps consumed the least annual energy use, district heating and boilers the most, the "others in between. ASHRAE/IESNA Standard 90.1-1999 specifies that radiant heating shall be used when heating is required for unenclosed spaces except loading docks equipped with air curtains.

8.2 WARM AIR FURNACES

Types of Warm Air Furnace

A warm air furnace is a combustion and heating device in which gas or oil is directly fired to heat the air through a heat exchanger, or air is directly heated by electric resistance elements in order to supply warm air to the conditioned space. Warm air furnaces can be classified into various categories according to

- Fuel types: natural gas, liquefied petroleum gas (LPG), oil, electric energy, or wood
- Airflow directions: upflow, horizontal, or downflow
- Applications: residential, commercial, or industrial

Natural gas is the primary fuel used in warm air furnaces. Warm air furnaces used for residences usually have a heating capacity up to 175,000 Btu/h (51.3 kW). Upflow models are the most popular models in residences. For commercial applications, heating capacities are usually greater than 150,000 Btu/h (44 kW). Horizontal models mounted inside a rooftop packaged unit or packaged heat pump for supplementary heating are widely used.

Upflow Gas-Fired Furnace

An upflow natural gas-fired warm air furnace consists of: a one or more gas burners, a heat exchanger, a forced-draft circulating fan or blower, a venting system, a filter, and an outer casing, as shown in Figs. 8.1*a* and *b*.

Gas Burners. Gas burners in small furnaces in residences are often atmospheric burners or fan-assisted burners. An atmospheric burner consists of an air shutter, a gas orifice, and outlet ports and is usually die-formed and made of aluminum painted, heavy-gauge steel or aluminized steel (or sometimes stainless steel). Atmospheric burners are either in-shot or upshot and are installed in single or multiple ports. In-shot burners are installed horizontally and are also used for Scotch marine boilers. Upshot burners are placed vertically and are suitable for vertical fire-tube boilers. Both Scotch marine and fire-tube boilers are discussed in later sections. Atmospheric burners are simple, require only a minimal draft of air, and maintain sufficient gas pressure for normal functioning. A fan-assisted gas burner uses a small fan to induce the combustion air through the heat exchanger and forces it to the outdoors via an air vent.

A power burner that uses a fan to supply and control combustion air is often employed for large furnaces. Conversion burners are integrated with furnaces for safety and efficiency. Older gas furnaces often use conversion burners.

Ignition. The ignition device is often a standing pilot ignition. These pilots are small. The pilot's flame is monitored by a sensor that shuts off the gas supply if the flame is extinguished. Another type of ignition is called spark ignition. It ignites intermittently only when required. Spark ignition saves more gas fuel than a standing pilot if the furnace is not operating.

Heat Exchangers. They are in the shape of clamshell, bent tube, or crossflow plate type. The combustion air is burned and flows inside clamshells, or tubes. Air to be heated flows over the outer surface of the heat exchanger. A heat exchanger is usually made of aluminum painted heavy-gauge steel, aluminized steel, or stainless steel.

Circulating Fan or Blower. A circulating fan or blower is always installed in a warm air furnace to force the air to flow over the heat exchanger, to distribute it to the conditioned space (except

when the warm air furnace is a part of an air-handling unit or packaged unit in which a supply fan is always provided), and to extract warm space air for heating again. A centrifugal fan with forward-curved blades and a double inlet intake is usually used.

Filter. A disposable filter, which is discussed in Chap. 17, is used to remove dust from the recirculating air. It is often located upstream of the fan.

Venting Arrangements. In a natural-vent warm air furnace, a draft hood is employed to connect the flue gas exit at the top of the heat exchanger section to a vent pipe or chimney. A relief air opening is also used to guarantee that the pressure at the flue gas exit is always atmospheric. A draft hood diverts the backdraft from the chimney, bypassing the burner without affecting the combustion operation. A direct-vent warm air furnace does not have a draft hood. If the vent pipe or chimney is blocked, a control system shuts down the warm air furnace. A power vent including a fan-assisted combustion furnace uses a fan to force or induce the combustion products or flue gas flowing through the heat exchanger and air vent.

Casing. The outer casing is usually made of heavy-gauge steel with removable access panels.

Furnace Operation. Gas is generally brought from the main to the pressure regulator. The regulator reduces the gas pressure to about 3.5 in. WG (870 Pa). Gas then flows through a gas valve controlled by a room thermostat. When a solenoid gas valve opens, the gas flows to the burners and mixes with the necessary outside primary air for combustion. The primary air–gas mixture then

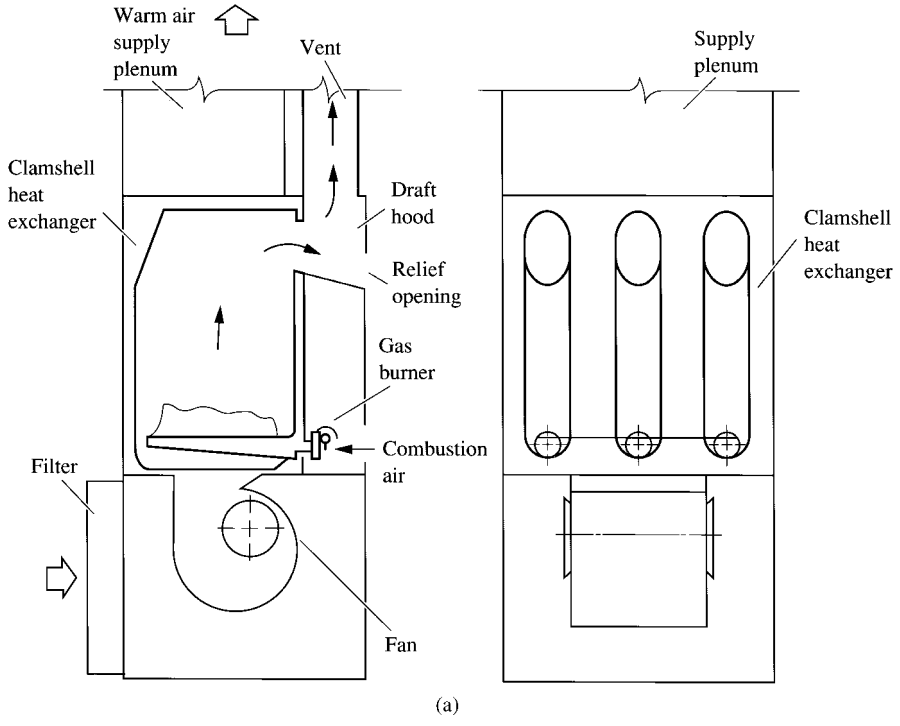


FIGURE 8.1 Upflow gas-fired furnaces. (*a*) Natural vent; (*b*) fan-assisted combustion.

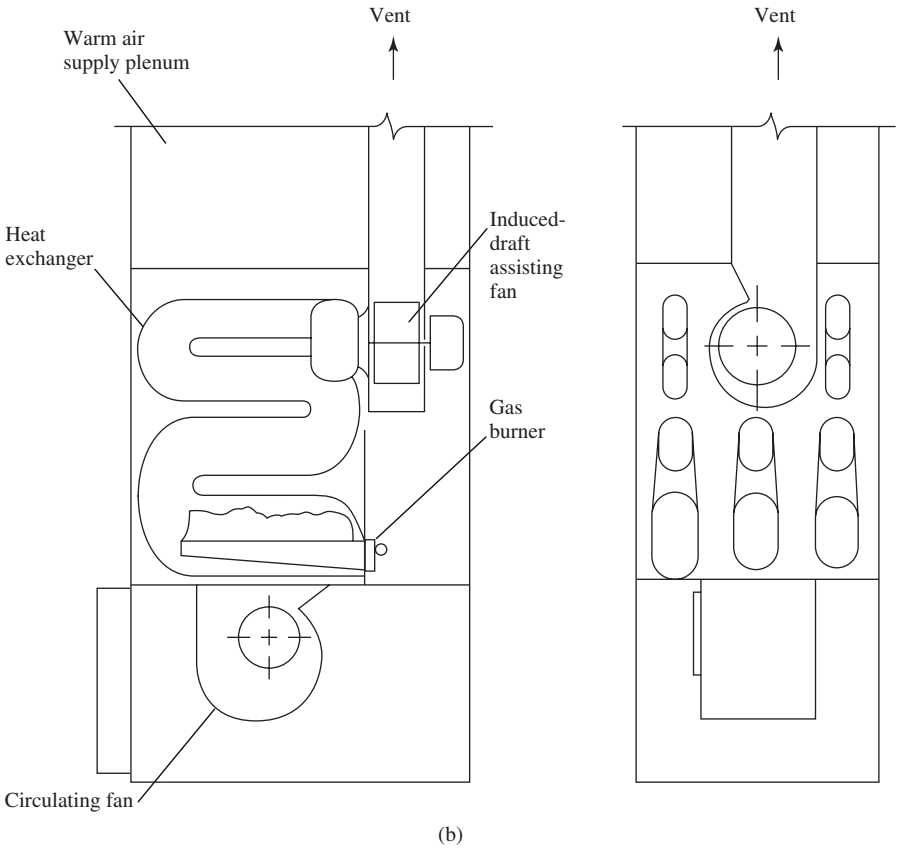


FIGURE 8.1 (*Continued*).

discharges from the port slots, mixes with ambient secondary air, and is burned. For a natural-vent furnace, the combustion products flow through the heat exchanger and the draft diverter and discharge outdoors through the vent pipe or chimney. For a fan-assisted combustion furnace, the induced-draft fan extracts the combustion products and the flue gas flowing through the heat exchanger, and forces them to discharge outdoors through the vent pipe or chimney.

A mixture of space recirculating air and outdoor air is pulled by the circulating fan from the return ducts and enters the bottom inlet of the warm air furnace. This air is then forced through the heat exchanger for heating again. For a heating-only furnace, the air temperature is raised by 50 to 80°F (28 to 45°C). Warm air discharges from the top outlet and is distributed to various conditioned spaces through the supply duct and outlets. An upflow gas-fired furnace is usually installed indoors and is often installed within a vented closet or vented basement in residential buildings.

Most natural gas furnaces can use liquefied petroleum gas. The main difference is their gas pressure. Natural gas usually needs a pressure of 3 to 4 in. WG (746 to 994 Pa) at the manifold, whereas LPG needs a higher pressure of about 10 in. WG (2486 Pa) and more primary air for gas burners.

Horizontal Gas-Fired Furnace

A typical horizontal gas-fired furnace in a rooftop packaged unit is shown in Fig. 8.2. It mainly consists of multiple gas burners, a heat exchanger, a combustion blower, and an ignition device. The supply fan used to force air to flow through the gas-fired furnace is the same supply fan used to force air through the filters and DX coil in the rooftop unit.

A power burner is often used in a horizontal gas-fired furnace. This type of burner provides better combustion and higher efficiency than atmospheric burners. A power draft centrifugal blower may be added to extract the combustion products and discharge them to a vent or chimney. The gas supply to the burner is controlled by a pressure regulator and a gas valve for the purpose of controlling the firing rate. In a premix power burner, gas and primary air are mixed first; then the mixture is forced to mix with secondary air in the combustion zone. A power burner usually has a higher gas pressure than the atmospheric burner used in residences.

The heat exchanger usually has a tubular two-pass arrangement, typically with 16-gauge (1.5-mm-thickness) stainless steel for primary surfaces and 18-gauge (1.2-mm-thickness) stainless steel for secondary surfaces, as shown in Fig. 8.2. The primary surface is the heat-transfer surface of the combustion chamber. The secondary surface is the surface of the tubes through which the flue gas flows after the combustion chamber. A cone-shaped flame is injected into a tubular or drum-shaped combustion chamber. A centrifugal blower is used to provide secondary air for forced combustion. Another small centrifugal blower may be used to induce the flue gas at the exit of the heat exchanger, the power vent, to maintain a negative pressure at the heat exchanger section so as to prevent the mixing of any leaked flue gas with the heated air. The mixture of outdoor and recirculating air is forced by the circulating blower and is heated when it flows over the primary and secondary surfaces.

Furnace Performance Factors

The performance of a gas-fired furnace is usually measured by the following parameters:

- *Thermal efficiency* E_t, in percent, is the ratio of the energy output of the fluid (air or water) to the fuel input energy. Input and output energy should be expressed in the same units. The value of E_t can be calculated as

$$E_t = \frac{100\,(\text{fluid energy output})}{\text{fuel energy input}} \tag{8.1}$$

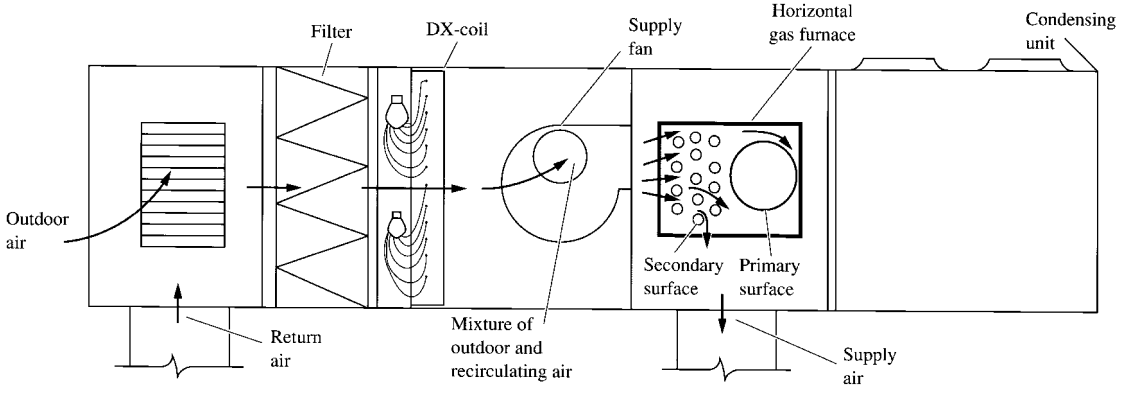


FIGURE 8.2 Horizontal gas-fired furnace in a rooftop packaged unit.

- *Annual fuel utilization efficiency* (AFUE) is the ratio of annual output energy from air or water to the annual input energy, expressed in the same units:

$$\text{AFUE} = \frac{100(\text{annual output energy})}{\text{annual input energy}} \tag{8.2}$$

AFUE also includes nonheating-season standing-pilot input energy loss. AFUE is similar to thermal efficiency E_t, except that AFUE is the ratio of annual energy output to energy input, whereas E_t is the ratio of energy output to energy input at specific test periods and conditions.

- The *steady-state efficiency* (SSE) is the efficiency of a given furnace according to an ANSI test procedure and is calculated as

$$\text{SSE} = \frac{100(\text{fuel input} - \text{fuel loss})}{\text{fuel input}} \tag{8.3}$$

The steady-state efficiency of gas-fired furnaces varies from 65 to 95 percent.

Test data in Jakob et al. (1986) for ASHRAE Special Project SP43, based on a nighttime setback period of 8 h and a setback temperature of 10°F (5.6°C), gave the following performance factors for gas-fired furnaces of two test houses with different construction characteristics:

Construction characteristics	AFUE, percent	SSE, percent
Natural vent		
Pilot ignition	64.5	77
Intermittent ignition	69	77
Intermittent ignition + venting damp	78	77
Power vent		
Noncondensing	81.5	82.5
Condensing	92.5	93

Saving Energy

Based on the above results, factors that affect the energy saving of the fuel and improve the furnace performance are as follows:

- *Condensing or noncondensing*. When the water vapor in the flue gas is condensed by indirect contact with recirculating air, part of the latent heat released during condensation is absorbed by the recirculating air, which increases furnace efficiency and saves fuel. The difference in the values of AFUE between condensing and noncondensing power vent furnaces may be around 10 percent. Corrosion-resistant materials such as stainless steel must be used for condensing heat exchangers due to the presence of chloride compounds in the condensate.
- *Power vent or natural vent*. Power vent guarantees the air supply for combustion and also forces the combustion product flows through heat exchangers including condensing heat exchangers with higher flow resistance.
- *Preheated outdoor combustion air*. If combustion air is taken from outdoors and preheated through an annular pipe that surrounds the flue pipe, part of the heat in the flue gas will be saved.
- *Automatic vent damper*. An automatic thermal or electric vent damper located in the vent pipe closes the vent when the furnace is not in operation. This decreases exfiltration from the house and restricts the amount of heat escaping from the heat exchanger.

- *Intermittent ignition*. Intermittent ignition saves more energy than standing-pilot ignition, especially during times when the furnace is not operating.

ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency of warm air furnaces, warm air duct furnaces, and warm air unit heaters for sizes as follows:

Equipment type	Size, Btu/h	Rating conditions	Minimum efficiency	Efficiency as of 10/29/2001
Gas-fired furnaces	$<$ 225,000		78% AFUE or 80% E_t	78% AFUE or 80% E_t
	$\geq$ 225,000	Maximum capacity	80% E_t	80% E_c
Oil-fired furnaces	$<$ 225,000		78% AFUE or 80% E_t	78% AFUE or 80% E_t
	$\geq$ 225,000	Maximum capacity	81% E_t	81% E_t
Gas-fired duct furnaces	All sizes	Maximum capacity	78% E_t	80% E_c
		Minimum capacity	75% E_t	
Gas-fired unit heaters	All sizes	Maximum capacity	78% E_t	80% E_c
		Minimum capacity	74% E_t	
Oil-fired unit heaters	All sizes	Maximum capacity	81% E_t	80% E_c
		Minimum capacity	81% E_t	

In 1987, the National Appliance Energy Conservation Act (NAECA) set minimum efficiency standards for gas furnaces at an AFUE of 78 percent, effective January 1, 1992. According to the venting systems and the steady-state efficiency, ANSI/AGA Standard Z21.47 classifies the central gas furnaces into the following four categories:

	Venting system	$T_{\text{flue}} - T_{f,\text{dew}}$,°F	Approximately SSE, percent
Category I	Nonpositive (natural vent)	$\geq$ 140 (noncondensing)	$<$ 83
Category II	Nonpositive (natural vent)	$<$ 140 (condensing)	$>$ 83
Category III	Positive (power vent)	$\geq$ 140 (noncondensing)	$<$ 83
Category IV	Positive (power vent)	$<$ 140 (condensing)	$>$ 83

Here T_{flue} represents the vent gas (flue gas) temperature, and $T_{f,\text{dew}}$ the dew-point temperature of the flue gas, both in °F.

Paul et al. (1993) reported that a category I fan-assisted gas furnace typically operated at an SSE of 80 to 83 percent, resulting in AFUEs greater than 78 percent (a 78 percent AFUE corresponds to an SSE of about 80.5 percent). Prior to 1987, the majority of gas furnaces installed in the United States for residences were draft-hood-equipped models. Because of the effect of the NAECA, for new installations and replacements, gas furnaces using natural vent draft hood systems will be gradually phased out in favor of the fan-assisted power vent furnaces.

Control and Operation

Capacity. The heating capacity of a gas-fired furnace is controlled by a gas valve and ignition device. For small furnaces the gas valve is usually controlled by a room thermostat that has on/off control. Either standing-pilot or intermittent ignition may be used. For large furnaces a two-stage gas valve, controlled by a two-stage thermostat, may be operated with a full gas supply, or at a reduced rate when the outdoor weather is mild. Energy or fuel savings may not be significant unless both the gas and the combustion air supply are controlled.

Nighttime Setback. ASHRAE research project SP43 found that a nighttime thermostat setback of 10°F (5.6°C) lower for 8 h improved E_t slightly (only a 0.4 percent increase). However, there is a 10 to 16 percent annual savings in energy input compared to those furnaces without nighttime setbacks.

Oversizing the furnace shortens the morning pickup time caused by nighttime setback. Use of an oversized furnace has a significant effect on the swing of space air temperature when an on/off control is used for the gas valve. Project SP43's results showed that when the furnace size corresponded to 1.4 times the design heating load (DHL), the furnace had a space temperature swing of 4.9°F (2.7°C) and a morning pickup time of about 1 h. If the furnace size was based on 1.7 times the DHL, the space temperature swing increased to 5.9°F (3.3°C), and the pickup time reduced to about 0.5 h. A furnace size based on 1.4 times the DHL is more suitable for a nighttime setback period of 8 h and a setback temperature of 10°F (5.6°C).

Fan Operation. In the past, continuous fan operation in small upflow gas-fired burners was said to offer the benefits of better air circulation, reduced noise (because the fan did not start and stop), and an even temperature distribution. Project SP43 showed that continuous fan operation resulted in a higher furnace efficiency. However, continuous operation consumes more electricity than intermittent operation in which the fan shuts off and the supply temperature drops below 90°F (32.2°C). In many locations with a high electricity-to-fossil-fuel cost ratio, an energy cost analysis based on AFUE may determine whether continuous or intermittent operation is more efficient.

The fan often starts about 1 min after the burner starts. Such a delay allows the heat exchanger to warm up and prevents a flow of cold air. The fan will shut down 2 to 3 min after the burner is shut off. The supply of residual heat from the heat exchanger also improves the performance of the furnace.

Safety. For safety, the vicinity of the furnace should be free of combustible gas, vapor, and material. Any passage to provide combustion air must be carefully planned. Gas and vent pipes should be installed according to local and federal codes.

8.3 HOT WATER BOILERS

A hot water boiler for space heating is an enclosed pressure vessel in which water is heated to a required temperature and pressure without evaporation. Hot water boilers are manufactured according to the American Society of Mechanical Engineers (ASME) boiler and pressure vessel codes. Boilers are usually rated according to their gross output heat capacity, i.e., the rate of heat delivered at the hot water outlet of the boiler, in MBtu/h, or thousands of Btu/h (kW). Hot water boilers are available in standard sizes up to 50,000 Mbtu/h (14,650 kW).

Selection of Fuel

Natural gas, oil, coal, and electricity are energy sources that can be used in hot water boilers. It is necessary to provide for an adequate supply during normal and emergency conditions and to take into account the limitations imposed by any building and boiler codes for certain types of equipment due to safety and environmental concerns. In addition, storage facilities and cost should be considered before a fuel is selected.

Whereas natural gas and electricity are supplied by a utility, LPG, oil, and coal all need space for storage within and outside the boiler plant.

Cost includes energy cost, initial cost, and maintenance cost. A gas-fired boiler plant requires the lowest initial costs and maintenance costs, oil-fired boiler plants are moderately higher, and coal-fired boiler plants are significantly higher (although their energy cost is lowest). An electric boiler is simple to operate and maintain. In addition, it does not require a combustion process,

chimney, or fuel storage. In locations where electricity costs are low, electric boilers become increasingly more attractive.

According to the data in the *Commercial Building Characteristics 1992* by the EIA, the percentages of floor area in all commercial buildings served by different kinds of primary energy source in hot water and steam boilers in 1992 in the United States are as follows:

Gas-fired boilers	71 percent
Oil-fired boilers	15 percent
Electric boilers	11 percent
Others	2 percent

Types of Hot Water Boiler

According to their working temperature and pressure, hot water boilers can be classified as follows:

1. *Low-pressure boilers*. These hot water boilers are limited to a working pressure of 160 psig (1103 kPa · g) and a working temperature of 250°F (121°C).
2. *Medium- and high-pressure boilers*. These boilers are designed to operate at a working pressure above 160 psig (1103 kPa · g) and a temperature above 250°F (121°C).

A low-pressure hot water boiler is generally used for a low-temperature water (LTW) heating system in a single building, regardless of the building's size. Medium- and high-pressure boilers are often used in medium-temperature water (MTW) and high-temperature water (HTW) heating systems for campuses or building complexes in which hot water temperature may range from 300 to 400°F (150 to 205°C).

Based on their construction and materials, hot water boilers can also be classified as fire-tube boilers, water-tube boilers, cast-iron sectional boilers, and electric boilers. Water-tube boilers, mainly used for steam at higher pressure and temperature, are not discussed here. Electric boilers are discussed in the next section.

Fire-Tube Boiler

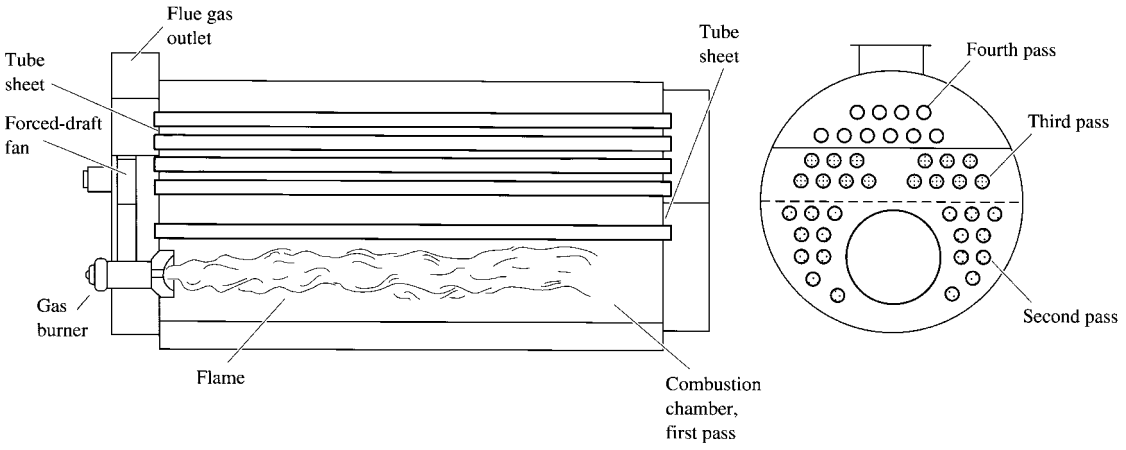
A fire-tube boiler's combustion chamber and flue gas passages are in tubes, which are all enclosed in a shell filled with water. Heat released from the combustion process and the flue gases is absorbed by the surrounding water, the temperature of which is increased to a required value.

Many kinds of fire-tube boilers have been developed. One of the more recently developed models is known as the modified Scotch marine boiler, which is a compact and efficient design originally used on ships. The Scotch marine boiler is probably the most popular hot water boiler manufactured today.

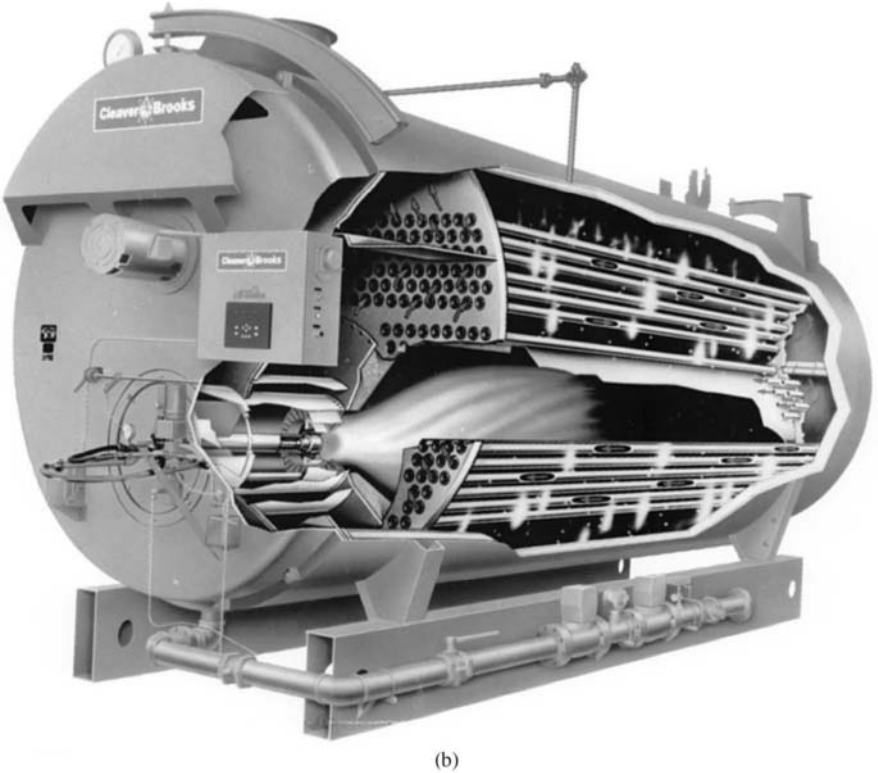
Scotch Marine Packaged Boiler

A Scotch marine packaged boiler consists mainly of a gas, oil, or gas/oil burner; a mechanical draft system; a combustion chamber; fire tubes; and a flue vent. A packaged boiler is a one-piece, integrated, factory-assembled boiler that includes a burner, outer steel shell, fire tubes, draft system, external insulation, controls, interconnecting piping, and wiring. A schematic diagram of a typical Scotch marine packaged boiler is shown in Fig. 8.3*a* and a cutaway illustration in Fig. 8.3*b*.

Flow Processes. Gas or oil and air are taken into the burner in measured quantities. The burner then injects a combustible air-fuel mixture containing the necessary combustion air into the



(a)



(b)

FIGURE 8.3 Scotch marine packaged boiler. (*a*) Schematic diagram; (*b*) a cutaway photograph of a typical product. *(Source: Cleaver-Brooks. Reprinted with permission.)*

combustion chamber, where it is initially ignited by an ignition device. The mixture burns, and the combustion process sustains itself once the heat it generates is greater than the heat it transfers to the water, i.e., once a high enough temperature in the combustion chamber is attained. The injected air-fuel mixture is then burned spontaneously.

The direct radiation from the flame and the high temperature in the combustion chamber both conduct heat through the wall of the chamber—the primary surface adjacent to the water that fills the shell. The combustion product from the combustion chamber—flue gas—is directed into fire tubes by headers. As the flue gas at a higher temperature flows through the fire tubes, it transfers heat to the surrounding water through the pipewall of the tubes—the secondary surface. The flue gas temperature drops, and its volume contracts. Because the number of fire tubes continuously decreases in the second, third, and fourth passes of fire tubes, which matches the volume contraction of flue gas, the gas velocity is maintained in a more uniform manner. Flue gas leaves after the fourth pass and is vented into the stack.

Return water enters the side of the boiler, sinks to the bottom, is heated, and rises again. Hot water is finally supplied at the top outlet. Such an arrangement prevents cold return water from surrounding the combustion chamber and producing thermal shock. It also promotes good water circulation.

Construction Characteristics. The constructional characteristics for Scotch marine packaged boilers are as follows:

- *Four flue gas passes*. Each pass means a horizontal run of the fluid flow passage. A four-pass flow arrangement for the flue gas, with a gradual reduction in gas flow area as the flue gas becomes colder, helps maintain a higher gas velocity and, therefore, a clean surface and a higher rate of heat transfer.
- *Sufficient heat-transfer surface area*. In a packaged Scotch marine boiler, the combustion chamber, fire tubes, and tube sheets at both ends are all heating surfaces. The amount of heating surface area directly affects a boiler's output. And 3 to 5 ft^2 (0.9 to 1.5 m^2) of heating surface area for each boiler horsepower, or 33,475 Btu/h (9.8 kW), of output, is the key value for providing sufficient capacity and a long-lasting boiler. A smaller heating surface area for each boiler horsepower of output often results in a higher heat flux, probably a higher surface temperature, and therefore a shorter life for the boiler.
- *Forced-draft arrangement*. In a forced-draft arrangement, the fan is located adjacent to the burner. The fan forces air into the combustion chamber. It also forces the flue gas to flow through the fire tubes and to discharge from the vent or chimney. A forced-draft fan supplies a controllable quantity of combustion air to the combustion chamber. The forced-draft fan is located upstream of the combustion chamber, so it handles dense, clean boiler room air at a comparatively lower temperature and lower volume flow rather than the hot, dirty, high-temperature, expanded flue gas downstream. The temperature fluctuation for the room air is smaller than that for the exhausted flue gas, so more accurate control of combustion air is possible than with an induced-draft design.

Cast-Iron Sectional Boiler

A cast-iron sectional boiler consists of many vertical cast-iron hollow sections in the shape of an inverted U filled with water. When the sections are linked together by bolts, the center part of the inverted U forms the furnace or the combustion chamber.

Because of its thick, heavy sections, a cast-iron sectional boiler has a large heat storage capacity and thus is slow to heat up. This reduces temperature swings when heat demand varies. Thick, heavy sections are also helpful in extending the boiler's service life by preventing corrosion.

The heating capacity of a cast-iron sectional boiler depends on the number of sections connected. Because of its relatively large combustion chamber and lower flow resistance, such a boiler is able to use atmospheric gas burners with a lower chimney height. Cast-iron sectional boilers are low-pressure boilers used in residences and in small and medium-size commercial buildings. They can be field-assembled and fitted in existing buildings.

Gas and Oil Burners

When natural gas is used as the fuel in boilers, as in gas-fired furnaces, atmospheric and power gas burners are usually used. Oil burners used in boilers differ from gas burners in that oil requires atomization and vaporization prior to combustion. Combustion air is supplied by either natural drafts or forced drafts. Ignition is usually provided by an electric spark or a standing pilot using gas or oil.

Oil burners used in residences are usually pressure atomizing-gun burners. This type of burner either directly atomizes oil at 100 to 300 psig (690 to 2069 kPa · g) high pressure or uses combustion air to atomize the oil. Combustion air is supplied by a blower, and electric spark ignition is generally adopted. Oil burners for boilers in commercial buildings inject oil and atomize it into fine sprays. Burners also force combustion air to mix with atomized oil and ignite the mixture, often with an electric spark. Complete combustion can be sustained with as little as 20 percent excess air.

Sometimes a boiler can be designed to use either gas or oil. A combination of a ring-type gas burner with an oil burner at the center is often used. Combination gas-oil burners are used more often than single-fuel burners in large boilers.

Condensing and Noncondensing Boilers

In condensing boilers, the water vapor in the flue gas is condensed and drained by means of a heat exchanger, as in gas furnaces. Thus, the latent heat of condensation can be recovered. In hot water boilers, the cooling medium is usually the return water from the conditioned space. The lower the temperature of the return water, the higher the amount of heat recovered. If the dew point of the flue gas is 130°F (54°C), return water or service water at a temperature below 125°F (51.7°C) may be used as the condensing cooling medium. Corrosion in the heat exchanger and flue gas passage caused by the condensate should be avoided. Noncondensing boilers have no way to condense the water vapor contained in the flue gas.

Boiler Efficiency

Boilers can be assessed by two efficiency values. The combustion efficiency E_c, in percent, is the ratio of heat output from the hot water or steam Q_{out} to the heat content rate of the fuel consumed Q_{fuel}, that is,

$$E_c = \frac{100Q_{\text{out}}}{Q_{\text{fuel}}} \tag{8.4}$$

Both Q_{out} and Q_{fuel} are expressed in Btu/h (kW). Annual fuel utilization efficiency (AFUE) is also an efficiency value used for hot water boilers. For noncondensing boilers, E_c varies from 80 to 85 percent. For condensing boilers, E_c varies from 85 to 90 percent.

ASHRAE/IESNA Standard 90.1-1999 mandates that the minimum efficiency requirements of gas- and oil-fired hot water boilers are as follows:

Equipment type	Size, Btu/h	Rating condition	Minimum efficiency	Efficiency as of 10/29/2001
Gas-fired boilers	$<$ 300,000	Hot water	80% AFUE	80% AFUE
	$<$ 300,000 and $\leq$ 2,500,000	Maximum capacity	80% E_c	75% E_t
	$>$ 2,500,000	Hot water	80% E_c	80% E_c
Oil-fired boilers	$<$ 300,000		80% AFUE	80% AFUE
	$<$ 300,000 and $\leq$ 2,500,000	Maximum capacity	83% E_c	78% E_t
	$>$ 2,500,000	Hot water	83% E_c	83% E_c

(Continued)

Equipment type	Size, Btu/h	Rating condition	Minimum efficiency	Efficiency as of 10/29/2001
Oil-fired (residual)	$<$ 300,000 and $\leq$ 2,500,000	Maximum capacity	83% E_c	78% E_c
	$>$ 2,500,000	Hot water	83% E_c	83% E_c

Chimney, or Stack

The chimney, or stack, is the vertical pipe or structure used to discharge flue gas. The breeching, or lateral, is the part of the horizontal duct that connects the flue gas outlet of the boiler and the vertical chimney. Breeching is commonly fabricated from 10-gauge steel (3.4-mm thickness) covered with a high-temperature insulation layer. Flue gas discharge from hot water boilers that burn gas or oil usually has a temperature rise of 300 to 400°F (167 to 222°C) over the ambient temperature. Chimneys for burning gas or oil should be extended to a certain height above adjacent buildings according to the local codes and topographical conditions. In high-rise buildings, locating the boiler room at the basement level may be very expensive because of the space occupied by the vertical stack and insulation inside the building.

Operation and Safety Controls

Heating Capacity Control. For hot water boilers, heating capacity control during periods of reducing heat demand is achieved by sensing the return water temperature and controlling the firing rate of the gas and oil burners. The firing rate of burners can be controlled in on/off, off/low/high, and modulating modes. An on/off control mode is usually used for small boilers in which hot water temperature control is not critical. A solenoid valve is used to open or close the fuel supply line to the burner. An off/low/high control mode offers better output control than an on/off mode. If the demand for heat is low, the boiler starts to fire at approximately one-half of its full capacity (the low fire setting). Upon further increase in demand, the burner fires at its maximum capacity (the high fire setting). A two-stage control valve and damper are used to control the supply of fuel and combustion air to the burners.

With modulating control, the boiler starts at a low fire of about one-third of the full-load capacity. As the demand for heat increases, an increasing amount of fuel and combustion air is supplied to the burner through the modulation of either a butterfly valve in a gas burner or an orifice of variable size in an oil burner. Both types of burner are linked to a damper to vary the supply of combustion air. For gas burners, two pressure sensors are often provided to maintain the gas pressure within a narrow range for proper operation. If one of the sensors detects an improper pressure, the gas supply is cut off and the boiler shuts down.

To maintain an optimum air-fuel ratio, the amount of O_2 in the stack flue gas can be maintained at 1 to 5 percent by modulating the air damper of combustion air to provide higher combustion efficiency. Usually, CO_2 is measured and monitored in small boilers, and O_2 is measured and monitored in large boilers.

Safety Control. One or more pressure or temperature relief valves should be equipped for each boiler. These are mechanical devices that open when the boiler pressure or temperature exceeds the rated value. An additional limit control is installed to open the switch and shut down the boiler as soon as the sensed water pressure or temperature exceeds the predetermined limit. When the pressure or temperature falls below the limit, the switch closes and the firing starts again.

A flame detector is used to monitor the flame. When the flame is extinguished, the controller closes the fuel valve and shuts down the boiler. An airflow sensor is also used to verify a continuous supply of combustion air. Once the combustion air is not present, the fuel valve closes before a dangerous fuel-air ratio can form.

Modern packaged boilers often include a totally enclosed and factory-assembled controlled cabinet that offers the latest microprocessor-based programmable direct digital controls, including flame safeguard and control systems.

8.4 ELECTRIC FURNACES, HEATERS, AND BOILERS

Electric Heating Fundamentals

Except for hot water electric boilers using electrodes, most electric furnaces, electric heaters, and electric heating coils used in HVAC&R installations are of the resistance type. When an electric current flows through a resistor under electric potential, heat is released to the ambient air or water. Electric resistance heaters of small wattage (e.g., an electric furnace in a residence with a heating capacity of less than 8 kW) usually use single-phase 120/240-V supply. Large-capacity electric resistance heaters for commercial and industrial applications usually use a three-phase 240/480-V supply.

For a single-phase electric heater, the electric power input P, in W, which is equal to the heat released from the resistor, can be calculated as

$$P = EI = I^2R \tag{8.5}$$

where E = electric voltage, V
I = electric current, A
R = electric resistance, Ω

For a three-phase electric heater, the electric power input is given as

$$P = 1.73E_l I_l = 1.73I_l^2R \tag{8.6}$$

where E_l = line voltage, V, and I_l = line current, A. When a three-phase electric heater is in a delta connection, as shown in Fig. 8.4*b*, then

$$E_p = E_l \qquad \text{and} \qquad I_p = \frac{I_l}{1.73} \tag{8.7}$$

where E_p = phase voltage, V, and I_p = phase current, A. When a three-phase electric heater is in a wye connection, then

$$E_p = \frac{E_l}{1.73} \qquad \text{and} \qquad I_p = I_l \tag{8.8}$$

Several kinds of overload protection devices are used for electric heaters to open the circuit if the electric current reaches a value that will cause a dangerous temperature:

- Fuses are alloys in the form of links to be inserted in the electric circuit. A fuse melts at a preset temperature, opening the circuit, when an overload current passes through.
- Circuit breakers are mechanical devices that open their contacts, by means of a warp action of a bimetallic strip or disk, when an excessive load passes through and heats them.

There are also thermal overload current-sensing devices and magnetic overload relays, both of which open a circuit when the electric circuit is overloaded.

National Electrical Codes (NEC) and local codes must be followed during the design, installation, and operation of electric heaters.

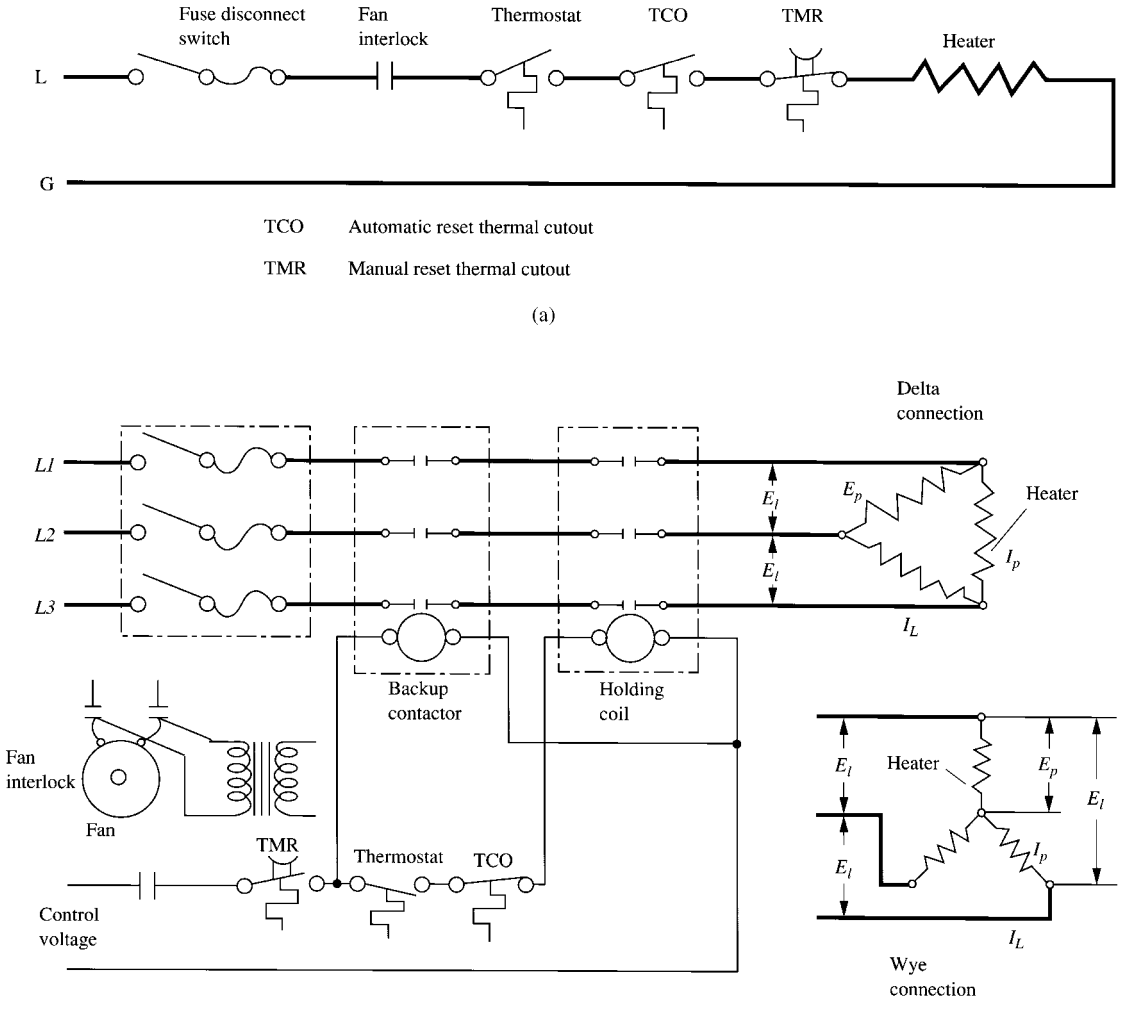


FIGURE 8.4 Typical electric heater wiring diagram. (*a*) Single-phase line voltage control with thermal cutout and fan interlock; (*b*) three-phase line voltage control with fan interlock and backup contactor.

Electric Furnaces, Electric Heaters, and Duct Heaters

Electric furnaces installed in rooftop packaged units, electric heaters equipped in air-handling units and packaged units, electric duct heaters mounted in air systems, and electric unit heaters installed directly above the conditioned space are all electric resistance warm air heaters. Air is forced through the electric heating element by the supply fan in a rooftop packaged unit or in an air-handling unit, or by the separately installed propeller fan in an electric unit heater.

Because the mixture of outdoor air and recirculating air or the outdoor air alone is clean and nonhazardous after passing through the air filter, most electric resistance heaters use heating

elements made of bare wire of various alloys, such as 80 percent nickel and 20 percent chromium wire. Bare wire is supported by bushes and brackets of insulating materials such as ceramics. Sheathed elements are also used for better protection.

The face velocity of air flowing through warm air electric heaters in rooftop packaged units is approximately the same as the face velocity of air at the cooling coil in AHUs. For a face velocity of 500 to 600 fpm (2.5 to 3 m/s), the temperature rise of warm air is often between 30 and 60°F (16.7 and 33.4°C).

The total pressure loss of the warm air flowing through an electric resistance heater in AHU or packaged units is usually less than 0.3 in. WC (75 Pa). For electric duct heaters, the air velocity and pressure loss may be higher. Figure 8.4*a* shows a typical wiring diagram for an electric heater using a single-phase electric power supply. Figure 8.4*b* shows a wiring diagram for a three-phase electric power supply. In the design and installation of an electric duct heater, the following requirements should be fulfilled:

- An electric duct heater should be installed at least 4 ft (1.2 m) downstream of the fan outlet, elbow, or other obstructions. Otherwise, devices should be installed to provide for an even distribution of airflow over the face of the heater.
- Do not install two electric duct heaters in series. Install electric duct heaters at least 4 ft (1.2 m) from the heat pumps and air conditioners. Following these suggestions prevents excessively high temperatures and the accumulation of moisture in the heater.
- A fan interlock circuit must be provided so that the electric heater does not operate unless the fan is on.
- Automatic-reset overload cutout safety control should be placed in series with the thermostat. A manual-reset overload cutout safety control is used in the backup safety control circuit. Limit controls may be used instead of a manual-reset cutout control. Disconnecting devices should be installed within sight of the electric heater.
- On each electric circuit, the maximum load of the electric heater is 48 A. The maximum fuse size is 60 A. Divide a larger electric heater into smaller heaters with smaller loads. The loading of an electric heater is often divided into stages for step control. Small heaters may be divided into two or three stages; large electric heaters may have as many as 16 stages.

Electric Hot Water Boilers

There are two types of electric hot water boiler: resistance and electrode. Resistance electric boilers use metal-sheathed heating elements, such as thin nickel chromium alloy wire, submerged in hot water. Such safety valves as a relief valve, thermal overload cutout, and high-limit switch should be provided. As with electric heaters, the electric load is divided into stages. The heating capacity is controlled in steps by sensing the return water temperature. Resistance boilers are more commonly used than electrode boilers. Resistance boilers are available in sizes with outputs up to about 3360 kW. Electrode boilers use water as a resistor. They are generally found in very large installations and require high-quality water and may operate at high voltages.

8.5 LOW-PRESSURE DUCTED WARM AIR HEATING SYSTEMS

System Characteristics

A low-pressure ducted warm air (forced-air) heating system has the following characteristics:

- It is often equipped with an upflow gas-fired furnace, oil-fired furnace, or electric heater as well as a centrifugal fan to force air through the furnace or electric heater.

- The external static pressure loss for the supply and return duct system is usually no greater than 0.5 in. WC (125 Pa).
- It uses a furnace heating capacity Q_f to airflow $\dot{V}_a$ ratio ranging from 50 to 70 Btu/h·cfm (31 to 43 W·s/L) and a temperature rise immediately after the furnace between 50 and 70°F (28 and 40°C).
- Because of the heat storage capacity of the supply duct system, the supply temperature differential $T_s - T_r$ is often 20 to 35°F (11.1 to 19.4°C). Here T_s represents the supply air temperature at the supply register, and T_r indicates the space temperature, both in °F (°C).
- The heating system is often integrated with the cooling system to form a heating/cooling air conditioning system.
- Most low-pressure warm air heating systems have a capacity no greater than 100,000 Btu/h (29.3 kW).

Low-pressure ducted warm air heating systems are usually used in residences and sometimes in small commercial buildings.

Types of Low-Pressure Ducted Warm Air Heating System

Low-pressure ducted warm air heating systems can be divided into two categories according to their duct systems:

- Supply and return duct system
- Supply duct and return plenum

Figure 8.5*a* shows a typical low-pressure ducted warm air heating system with a supply and return duct system for a single-family house. Recirculating air from the living room, dinning room, and bedrooms flows through return ducts and enters the lower part of the upflow gas furnace. It mixes with outdoor air. The mixture is forced through a gas furnace and heated to a required supply temperature at the supply plenum. Warm air is then supplied to various rooms through supply ducts and registers.

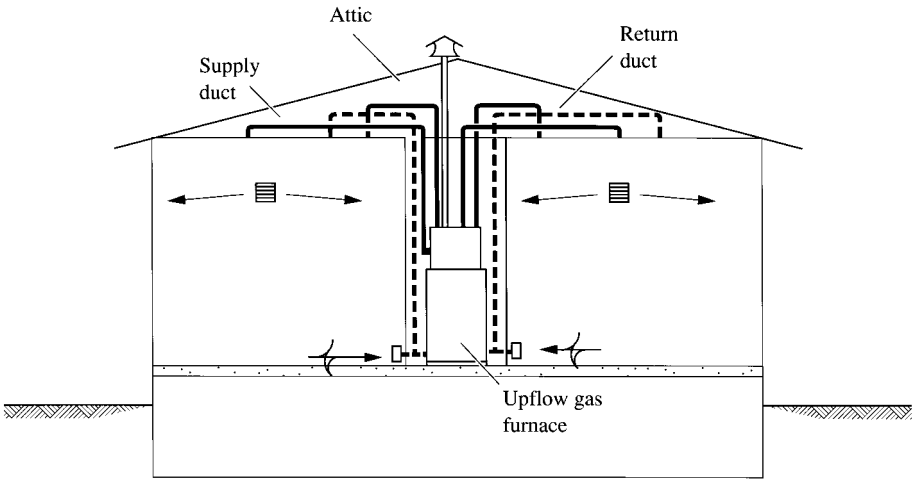
Figure 8.5*b* shows a typical low-pressure ducted warm air heating system with supply ducts and a return plenum. There are no return ducts. Warm air is supplied to various rooms through supply ducts. Recirculating air flows back to the return plenum by means of undercut or louvered doorways and corridors. Once an interior door between rooms is closed, air squeezes through the door undercuts and louvers to the return plenum as well as other cracks and gaps to the attic or outdoors. The room is therefore maintained at a positive pressure. This causes a reduction in the supply flow rate and an increase in the infiltration rate of the whole house.

Field tests performed by Cummings and Tooley (1989) on five Florida houses show that, for a low-pressure ducted warm air heating system with supply ducts and a return plenum, the average whole-house infiltration was 0.31 ach (air change per hour) when all the interior doors between rooms were open. If the interior doors were closed, the average whole house infiltration increased to 0.91 ach. The typical pressure difference between rooms separated by a closed door was 0.032 in. WC (8 Pa), for a door undercut of 0.5 in. (13 mm).

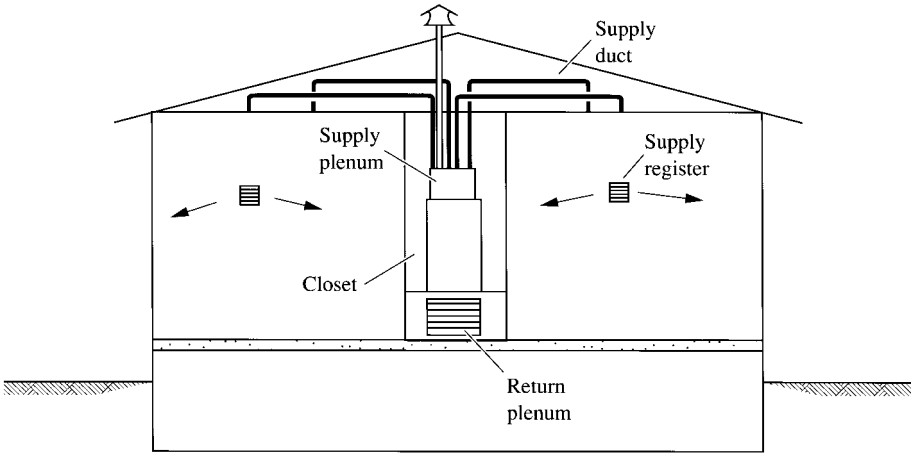
Heat Supplied to Conditioned Space

Heat supplied to the conditioned space by a low-pressure ducted warm air heating system, denoted by Q_{sh}, Btu/h (W), can be calculated as

$$Q_{\text{sh}} = 60\dot{V}_s\rho_s c_{pa}(T_s - T_r) \tag{8.9}$$



(a)



(b)

FIGURE 8.5 Typical low-pressure ducted warm air heating system. (*a*) Supply and return duct; (*b*) supply duct and return plenum.

where $\dot{V}_s$ = volume flow rate of supply air, cfm [$m^3/(60\ s)$]
T_s, T_r = temperature of supply and space air, °F (°C)

In Eq. (8.9), ρ_s represents the density of supply air. For a supply air temperature of 100°F (38°C) and a relative humidity around 20 percent, ρ_s can be taken as 0.07 lb/ft^3 (1.12 kg/m^3). The term c_{pa} indicates the specific heat of moist air. For simplicity, its value can still be taken as 0.243 Btu/lb·°F (1017 J/kg·°C).

Duct Efficiency and System Efficiency for Heating

Duct efficiency for heating $\eta_{\mathrm{du},h}$, in percent, can be calculated by dividing the heat energy output from the supply and return duct system by the heat energy input to it, i.e.,

$$\eta_{\mathrm{du},h} = 100\frac{q_{ho,s} + q_{ho,r}}{q_{hi,s} + q_{hi,r}} = 100\left(1 - \frac{q_{hl,s} + q_{hl,r}}{q_{hi,s} + q_{hi,r}}\right) \tag{8.10}$$

where $q_{ho,s}$, $q_{ho,r}$ = output heat energy from supply ducts and return ducts, Btu/h (W)
$q_{hi,s}$, $q_{hi,r}$ = input heat energy to supply ducts and return ducts, Btu/h (W)
$q_{hl,s}$, $q_{hl,r}$ = heat loss from supply ducts and return ducts, Btu/h (W)

and the supply duct efficiency for heating $\eta_{sd,h}$, in percent, is

$$\eta_{sd,h} = 100\frac{q_{ho,s}}{q_{hi,s}} = 100\frac{1 - q_{hl,s}}{q_{hi,s}} \tag{8.11}$$

The system efficiency for heating $\eta_{sy,h}$, in percent, can be calculated as

$$h_{sy,h} = 100\frac{q_{ho,s} + Q_{f,h}}{Q_{f,in}} = 100\left[E_t\eta_{\mathrm{du},h} + \frac{Q_{f,h}}{Q_{f,in}}\right] \tag{8.12}$$

where $Q_{f,h}$ = jacket loss and equipment losses from furnace and released to conditioned space, Btu/h (W)
$Q_{f,in}$ = total energy input to furnace, including auxiliary energy input, Btu/h (W)

Location of Furnace and Duct Insulation

The location of the furnace has a significant effect on the system efficiency of heating. In Jakob et al. (1986), if the gas furnace is installed in a closet, the supply duct is mounted inside the conditioned space, and the equipment losses of the furnace become the direct heat gains of the conditioned space, then $\eta_{sy,h}$ might be 20 percent higher than for those installations where the furnace and supply ducts are in the attic or basement.

Jakob et al. (1986) also showed that if ducts had an exterior or interior insulation of R5, that is, an insulation layer with an R value of 5 $\mathrm{h\cdot ft^2\cdot °F/Btu}$ (0.9 $\mathrm{m^2\cdot °C/W}$), the duct efficiency increased from 61 percent (without duct insulation) to about 78 percent. The system efficiency saw a smaller increase because a portion of the heat loss from the duct without insulation had been used to heat the attic or basement, which in turn reduced the heat loss from the conditioned space.

Duct Leakage

ASHRAE Standard 90.1-1999 mandates the minimum duct seal level as discussed in Sec. 17.8. Field tests in many houses have showed that actual duct leakage in many low-pressure ducted warm air heating systems is considerably higher.

Lambert and Robinson (1989) analyzed the duct leakage, whole-house leakage, and heat energy use of 800 electric-heated houses built since 1980 in the Pacific northwest. Tested houses were

divided into two groups: highly energy-efficient Model Conservation Standard (MCS) houses and a control group built to current (regional) practice (CP) or standards. Ducted heating systems in CP houses had 26 percent more air leakage than unducted systems and used 40 percent more heating energy. Ducted heating systems in MCS houses had 22 percent more air leakage and used 13 percent more heating energy than unducted MCS houses.

Gammage et al. (1986) studied the ducted warm air systems of 31 Tennessee houses. They found that the air infiltration rate was 0.44 ach when the forced-air systems were off and 0.78 ach when the forced-air systems were on.

Field tests have also shown that leakages are greater for the return duct than for the supply duct because of the greater importance of the supply air. Also, the return plenum is often not carefully sealed. In five Florida homes with low-pressure ducted warm air systems with supply and return plenums, Cummings and Tooley (1989) found that when repairs were made to seal the return plenums, the infiltration in these five homes dropped from an average of 1.42 ach to 0.31 ach.

When a ducted warm air heating system is operating, supply duct leakage in such nonconditioned space as the attic or basement raises the space pressure to a positive value and promotes exfiltration. Return duct leakage extracts the space air, lowers the space pressure to a negative value, and promotes infiltration. Both types of leakage increase the whole-house infiltration.

Suggested remedies to reduce duct leakage and energy use are as follows:

- Externally seal the duct with tape; seal the ducts internally if possible.
- Seal the return plenum and equipment if there is leakage. Seal the duct, pipes, and cable penetrations through the structures.
- Avoid locating ducts in unconditioned spaces. Provide insulation for ducts running in unconditioned spaces.
- For a low-pressure ducted warm air system without return ducts, an adequate door undercut or door louver should be provided.

Thermal Stratification

A low-pressure ducted warm air heating system with a gas-fired furnace always has a high supply temperature differential ΔT_s. If ΔT_s exceeds 30°F (16.7°C), or if there is a high ceiling, thermal stratification may form in the conditioned space. The vertical temperature difference may be greater than 5°F (2.8°C) and cause discomfort, as mentioned in Sec. 4.8. In addition, a higher temperature near the ceiling may increase heat transfer through the ceiling, attic, and roof.

A greater supply air volume flow rate, lower ΔT_s, higher downward air jet velocity, and suitable location for the supply grille are remedial measures that reduce thermal stratification and vertical temperature differences.

Part-Load Operation and Control

For low-pressure ducted warm air heating systems, a thermostat or a temperature sensor is usually installed in a representative space to control the gas valve of a furnace operating under an on/off or two-stage step control mode to maintain the required space temperature. The proportion of on and off time in each operating on/off cycle varies to meet the varying space heating load. Figure 8.6 shows a typical on/off control for a gas furnace, resulting in a temperature variation for a point 20 ft (6.1 m) downstream of the supply plenum and at the supply register.

When the space temperature increases above the upper limit as the space heating load falls, the controller shuts off the gas valve so that the heat supply to the space is cut off. When the space temperature falls below the lower limit, the gas valve will open again and raise the space temperature.

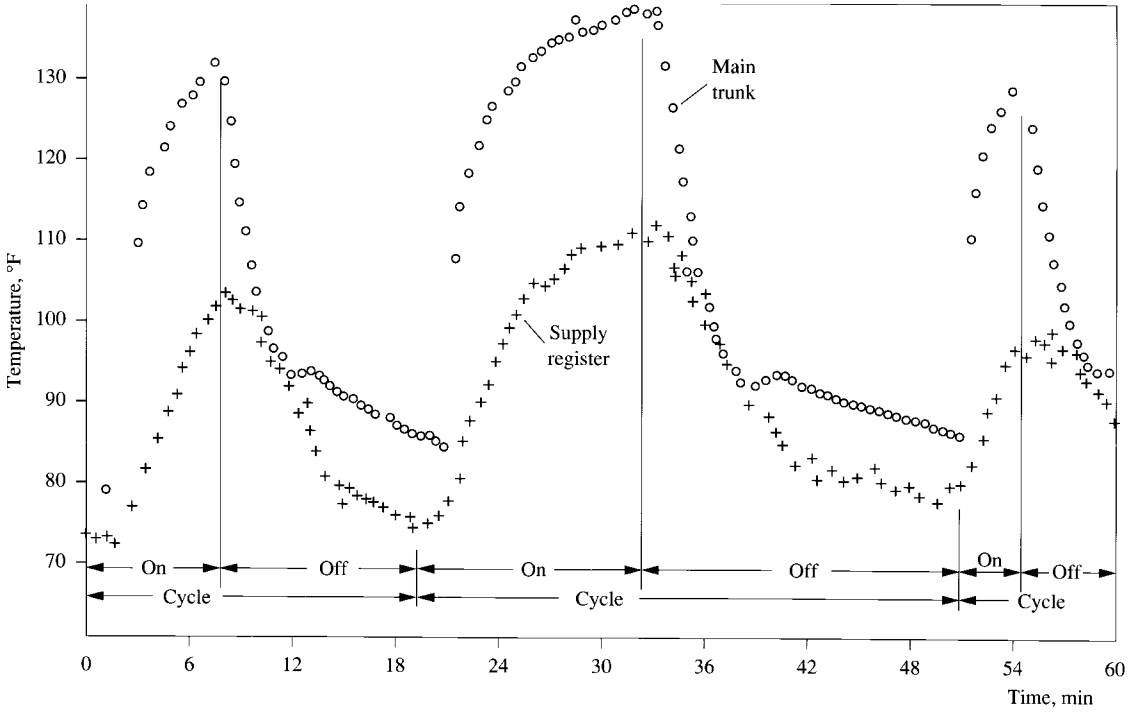


FIGURE 8.6 Typical on/off control of a gas furnace and the variation of temperature of supply plenum and supply register.

The time period for an on/off operating cycle is generally between 5 and 15 min. Too short a cycle may result in unstable operation—a condition known as hunting.

8.6 DUCTED WARM AIR HEATING SYSTEMS

Ducted warm air heating systems are heating systems that are part of the air-handling unit, packaged units, packaged heat pumps, or fan-coil units that use water heating coils, gas furnaces, electric heaters, or indoor coils in heat pumps to heat the air. Warm air is then supplied to the conditioned space through the duct system.

A ducted warm air system has the following characteristics:

- It is often integrated with a cooling system, forming a heating/cooling system.
- Its external pressure loss for a duct system is usually between 0.5 and 2 in. WC (125 and 500 Pa), except for fan-coil units.
- The heating capacity is controlled by modulating the water flow rate to the water heating coil, by modulating the gas valve, or by controlling the refrigeration flow in an on/off or step control mode in the heat pumps.
- Heating systems often have a capacity greater than 100,000 Btu/h (29 kW).

Because the perimeter zones of office buildings need strict control of the vertical temperature difference in their conditioned space, a smaller supply air temperature differential, such as $\Delta T_s \leq$

15°F (8.3°C) is usually used. The performance and design of these types of heating and cooling systems will be discussed in later chapters.

8.7 HOT WATER HEATING SYSTEMS USING FINNED-TUBE HEATERS

Types of Hot Water Heating System

Like hot water boilers, hot water heating systems can be classified according to their operating temperature into two groups:

- *Low-temperature water systems*. These operate at a temperature not exceeding 250°F (120°C), typically 190°F (88°C) supply and 150°F (65°C) return, and a maximum working pressure not exceeding 150 psig (1034 kPa · g), usually less than 30 psig (207 kPa · g).
- *Medium- and high-temperature water systems*. In medium-temperature water systems, the operating temperature is 350°F (177°C) or less, and the operating pressure is 150 psig (1034 kPa · g) or less. In high-temperature water systems, the maximum operating temperature can be from 400 to 450°F (205 to 232°C), and the maximum operating pressure is 300 psig (2070 kPa · g). In both medium- and high-temperature systems, the hot water supplied to and returned from the heating coils and space heaters is typically 190 and 150°F (88°C and 65°C), respectively.

Low-temperature hot water heating systems are widely used for space heating in residential and commercial buildings. Medium- and high-temperature hot water heating systems are often used in central heating plants for university campuses or groups of buildings, or in industrial applications for process heating. Only low-temperature hot water systems are discussed in this section.

In low-temperature hot water heating systems, the two-pipe individual loop and radiant floor-panel heating systems are currently used in residential, commercial, and industrial buildings.

Two-Pipe Individual Loop System

Most current low-temperature hot water heating systems that use finned-tube heaters are equipped with zone control facilities. Without such a control system, rooms that face south may be overheated and rooms that face north may be underheated in northern latitudes because of the effects of solar radiation. The typical piping arrangement for a low-temperature hot water heating system is the two-pipe individual-loop system, as shown in Fig. 8.7. On a cold winter day, water in a typical individual-loop system returns from various finned-tube baseboard heaters through different individual return loops at a temperature between 150 and 155°F (65 and 68°C). It is heated at the hot water boiler to a temperature of 190°F (88°C), extracted by the on-line circulating pump on each individual loop, and distributed to the finned-tube baseboard heaters by means of supply and branch pipes. Because of the higher average temperature of hot water in the baseboard heaters, heat is released to the space air to offset the heating load by means of radiation and convection in various control zones, such as north, south, east, and west zones. The temperature of hot water then drops to between 150 and 155°F (65 and 68°C) and returns to the hot water boiler again via a return main.

In Fig. 8.7, there are several individual loops, or control zones. In each individual loop, several finned-tube baseboard heaters in a large room can be connected in series. Finned-tube baseboard heaters in small rooms belonging to the same individual loop can be connected in a reverse-return arrangement. In such an arrangement, the length of pipeline that hot water travels is nearly the same for each baseboard heater for a better system balance.

If a hot water heating system using finned-tube heaters is part of an air conditoning system, then the outdoor ventilation air during winter heating will be provided for the occupants in the heating

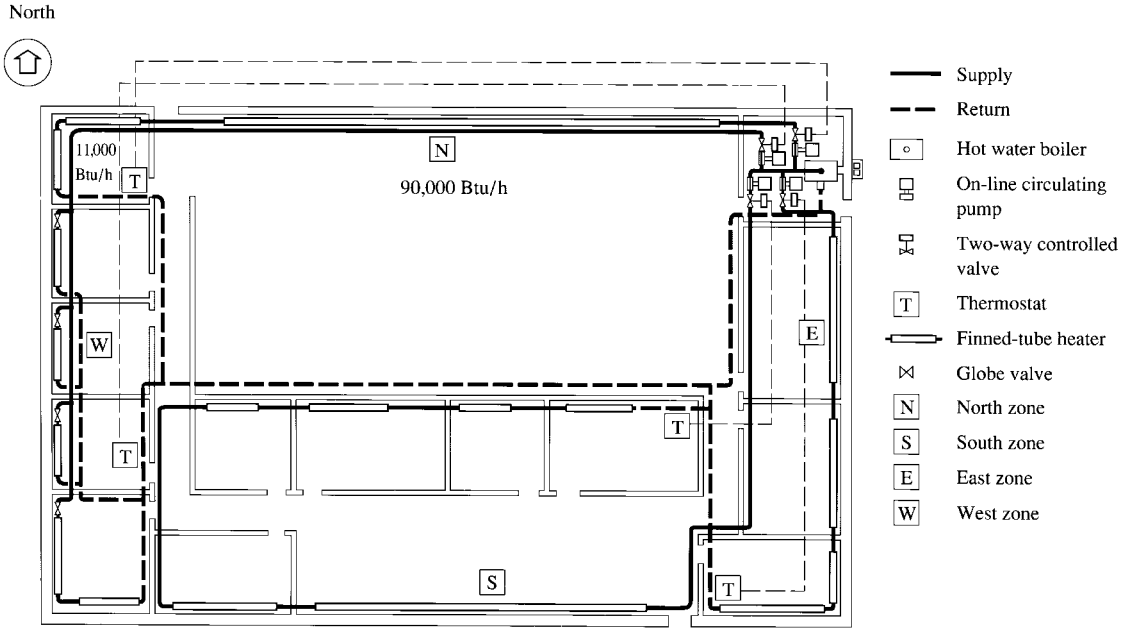


FIGURE 8.7 A two-pipe individual-loop low-temperature hot water heating system for a factory.

space by the air conditioning system that cools the conditioned space in summer. In locations of cold climate, if only winter heating is required, then a separate outdoor ventilation system sometimes is required instead of opening the windows manually. Outdoor ventilating systems are discussed in Chaps. 23 and 24.

The design procedure for a hot water heating system involves the following:

- Calculation of the space heating load, as discussed in Chap. 6
- Selection of suitable finned-tube heaters
- Division of the heating space into various control zones, or individual loops
- Planning of a piping layout, containing branches and necessary piping fittings
- Location of the boiler, circulating pumps, air separator, and expansion tank in the mechanical equipment room, and determination of their capacities
- Specification of the control functions of the capacity and safety control systems, including the sequence of operation

Finned-Tube Heaters

A finned-tube heater is a terminal unit installed directly inside the conditioned space to deliver heat energy to the space by means of radiation and convection. A finned-tube heater consists of a finned-tube element and an outer enclosure, as shown in Fig. 8.8. The tubes are often made of copper and steel. Copper tubes are usually of 0.75-, 1-, and 1.25-in. (19-, 25-, and 32-mm) diameters, and steel tubes of 1.25- and 2-in. (32- and 50-mm) diameters. For copper tubes, fins are often made of aluminum. For steel tubes, the fins are made of steel. Fin spacing varies from 24 to 60 fins per foot (79 to 197 fins per meter). A finned-tube heater may have a length up to 12 ft (3.6 m).

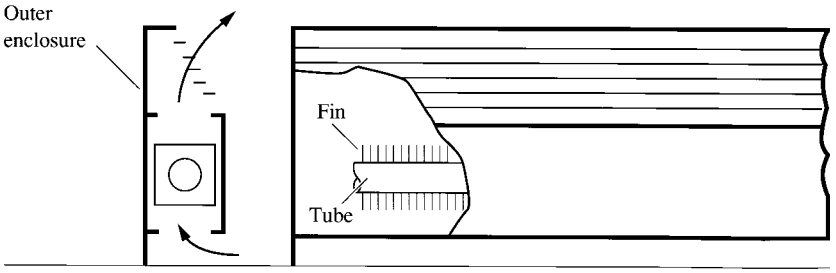


FIGURE 8.8 Baseboard finned-tube heater.

Although various configurations of the outer enclosure have been designed and manufactured, each enclosure must have a bottom inlet and a top outlet for better convection. The enclosure is usually made of 18-gauge (1.2-mm-thickness) steel with corrosion-resistant coating to provide protection and improve appearance. To allow a higher heating capacity, two finned-tube elements can be set in a two-tier (two-row) arrangement. The most widely used finned-tube heater is the baseboard heater, which is often installed at a level of 7 to 10 in. (175 to 250 mm) above the floor. It is usually 3 in. (75 mm) deep and has only one pier of finned-tube elements. Baseboard heaters are usually mounted on cold walls and release heat nearly at the floor level. They also interfere less with indoor decor than other heaters.

A wall finned-tube heater has a greater height and is available in various shapes to meet the architectural interior design. A convector has a cabinet-type enclosure with one or two tiers of finned-tube heating elements installed under the window sill. The rated heating capacity of a finned-tube heater depends mainly on its length, the fin spacing, the average water temperature flowing through the heating element, and the temperature of air entering the heater. Table 8.1 lists the rated heating capacity of typical finned-tube heaters for entering air at 65°F (18.3°C) and different average hot water temperatures.

Design Considerations

In the older low-temperature hot water heating system design, a water temperature drop of ΔT_w = 20°F (11.1°C) at the finned-tube element was usually used. The current trend is to use a greater temperature drop, such as 20 to 50°F (11.1 to 27.8°C). A detailed cost analysis should be performed to determine the optimum temperature drop in a finned-tube heater.

The sizing of low-temperature hot water pipes is usually based on a pressure drop of 0.5 to 1.5 psi (1 to 3 ft head loss per 100 ft or 1 to 3 m head loss per 100 m of pipe length). Friction charts for hot water steel pipes and the equivalent length of pipe fittings are discussed in Chap. 7.

TABLE 8.1 Heating Capacity of Finned-Tube Elements for Entering Air Temperature of 65°F, Btu/h · ft

		Average water temperature, °F					
Type of finned-tube heater	No. of rows	220	210	200	190	180	170
Steel tube: 1.25-in. dia., steel fin	1	1260	1140	1030	940	830	730
	2	2050	1850	1680	1520	1350	1190
Copper tube: 1-in. dia., aluminum fin	1	1000	900	820	740	660	580
	2	1480	1340	1210	1100	970	860

Source: Adapted with permission from *Handbook of HVAC Design 1990.*

For small low-temperature hot water heating systems, an open expansion tank is usually used. For medium-size and large systems, a diaphragm tank may be more suitable. Circulating water pumps are often on-line pumps (circulators) with low pump head.

Part-Load Operation and Control

In a low-temperature hot water heating system, one of the basic part-load controls is the variation of the hot water supply temperature from the boiler in response to a variation in outdoor temperature. For instance, in the midwest, a low-temperature hot water heating system has a winter design outdoor temperature $T_o = 0°F$ ($-17.8°C$). At the winter design outdoor temperature, the hot water supply temperature T_{ws} is 190°F (88°C). When the outdoor temperature drops, T_{ws} is reset as follows:

Outdoor temperature T_o, °F (°C)	Supply temperature T_{ws}, °F (°C)
0 (−17.8)	190 (88)
32 (0)	135 (57)
60 (16)	85 (29)

A hot water temperature sensor located at the hot water exit of the boiler, whose set point is reset by an outdoor temperature sensor, is used to control the firing rate of the boiler by means of a DDC unit. Zone control can be performed better by sensing the hot water temperature that returns from each individual loop and then modulating the control valve to vary the mass flow rate of hot water supplied to that zone through an on-line circulating pump (see Fig. 8.7).

ASHRAE/IESNA Standard 90.1-1999 specifies that a hot water system with a design heating capacity exceeding 300,000 Btu/h (88 kW) shall include controls to automatically reset to a lower hot water supply temperature according to building loads or outdoor temperature during part-load for energy saving except for systems that use variable flow to reduce pumping energy.

For a low-temperature hot water heating system installed with multiple boilers, the control strategy is to decide when to turn a boiler on or off. This optimum operation control can be accomplished by using a microprocessor-based controller that fires a standby boiler according to a preprogrammed software instruction. This strategy depends on not only the increase or reduction in heating demand but also how the operating cost of such a hot water heating system can be minimized. ASHRAE/IESNA Standard 90.1-1999 also specifies that for a boiler plant equipped with more than one boiler, provisions shall be made so that the hot water flow in the boiler plant can be automatically reduced when a boiler is shut down.

Example 8.1. A two-pipe individual-loop low-temperature hot water heating system is used to heat a factory that has a layout shown in Fig. 8.7. At winter design conditions, hot water is supplied to the heated space at a temperature of 190°F (88°C) and returns from the baseboard finned-tube heaters at a temperature of 150°F (65°C).

1. If the space heating load for the largest room (facing north) is 90,000 Btu/h (26.4 kW) and for the northwest corner room is 11,000 Btu/h (3.2 kW), and if steel tubes and fins are used, determine the number of feet of finned tubing required for each of these two rooms.
2. If a pressure drop of 1 ft/100 ft (1 m/100 m) of pipe is used and the hot water system is equipped with an open expansion tank, determine the diameter of the hot water supply main for these two rooms.
3. Divide this hot water system into appropriate control zones, or individual loops.

Solution

1. For the largest room, if a two-row finned-tube heater is used, then from Table 8.1, for an average hot water temperature of $(190 + 150)/2 = 170°F$ (77°C), the heat output of each foot of two-row

finned tube is 1190 Btu/h (350 W). The number of feet required is therefore

$$\frac{90{,}000}{1190} = 75.63 \text{ ft (76 ft, or 23 m)}$$

For the northwest corner room, a single-row finned-tube heater is used. The number of feet required is

$$\frac{11{,}000}{730} = 15.06 \text{ ft (15 ft, or 4.5 m)}$$

Along the north side's external wall, an 8-ft (2.5-m) finned tube is mounted. On the west side's external wall, a 7-ft (2-m) finned tube is mounted.

2. Because the density of water at 170°F is 60.8 lb/ft^3, from Eq. (7.1), the flow rate of hot water supplied to the largest room and the northwest corner room can be calculated as

$$V_{\text{gal}} = \frac{Q_w}{500(T_{we} - T_{wl})} = \frac{90{,}000 + 11{,}000}{500(190 - 150)} = 5.05 \text{ gpm}$$

Then from the water friction chart for water in steel pipes (Fig. 7.2 in Chap. 7), for a pressure drop of 1 ft/100 ft, the pipe diameter for the hot water supply main to the largest room and northwest corner room is 1.25 in. (32 mm).

3. This two-pipe individual-loop low-temperature hot water heating system should be divided into four zones: north, south, east, and west. Each zone, or individual loop, includes the rooms whose outer walls face that direction (except the south zone, which also includes interior rooms without external walls). Such a setup can offset the variation in solar radiation and the operating conditions for each zone to prevent overheating of rooms facing south and underheating of rooms facing north. For each individual loop, or control zone, a thermostat senses the return hot water temperature and modulates the control valve and the flow rate of hot water supplied to that loop by a DDC unit in order to meet the variations in zone heating load.

8.8 HYDRONIC RADIANT FLOOR-PANEL HEATING SYSTEMS

System Description

A hydronic (hot water) radiant floor-panel heating system provides heating and thermal comfort to the occupants in an enclosed indoor environment primarily by means of the radiant heat transfer from the floor panels. As with the hot water systems using finned-tube heaters, a hydronic radiant floor-panel heating system consists of a hot water boiler, several individual-loop pipings with radiant floor panels, an air separator, an expansion tank, circulating hot water pumps, control valves, and accessories, as shown in Fig. 8.9*a* and *b*. Instead of finned-tube heaters, radiant floor panels are used. For each individual loop that forms a control zone, there may be only one large radiant floor panel, or several small panels connected in series.

Radiant Floor Panel

Tubes made of plastic, rubber, steel, or copper of diameters from 0.3875 to 1.0 in. (10 to 25 mm) are often embedded in a floor slab with a tube spacing from 6 to 15 in. (150 to 375 mm) to form a heating floor panel, as shown in Fig. 8.9*c*. Tubes are laid in the shape of serpentine to ensure an even panel surface temperature. Floor slabs are often made of concrete with a thickness between

2 and 6 in. (50 and 150 mm). The thickness of the floor slab has a significant effect on its thermal storage and slow response to heating load changes. Floor coverings like carpet and wood affect the overall thermal resistance of the floor panel. The water temperature, therefore, should be increased accordingly in order to provide the required amount of heat that is released from the floor panel. An insulating layer and a moisture barrier under the concrete slab are often needed to reduce the heat losses, as shown in Fig. 8.9*c*.

Thermal Characteristics of Floor Panel

According to *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*, the heat flux of the radiant heat transfer irradiated from a heating floor panel q_r, Btu/h · ft^2 (W/m^2), can be calculated as

$$q_r = 0.15 \times 10^{-8}(T_{Rp}^4 - T_{R,\mathrm{AUST}}^4) \tag{8.13}$$

where T_{Rp} = absolute temperature of effective panel surface,°R (K)
$T_{R,\mathrm{AUST}}$ = absolute temperature of area-weighted average temperature of unheated surfaces exposed to panels (AUST),°R (K)

The heat flux due to natural convection flows upward from a heated floor panel q_c, Btu/h · ft^2 (W/m^2), is

$$q_c = 0.31(T_p - T_a)^{0.31}(T_p - T_a) \tag{8.14}$$

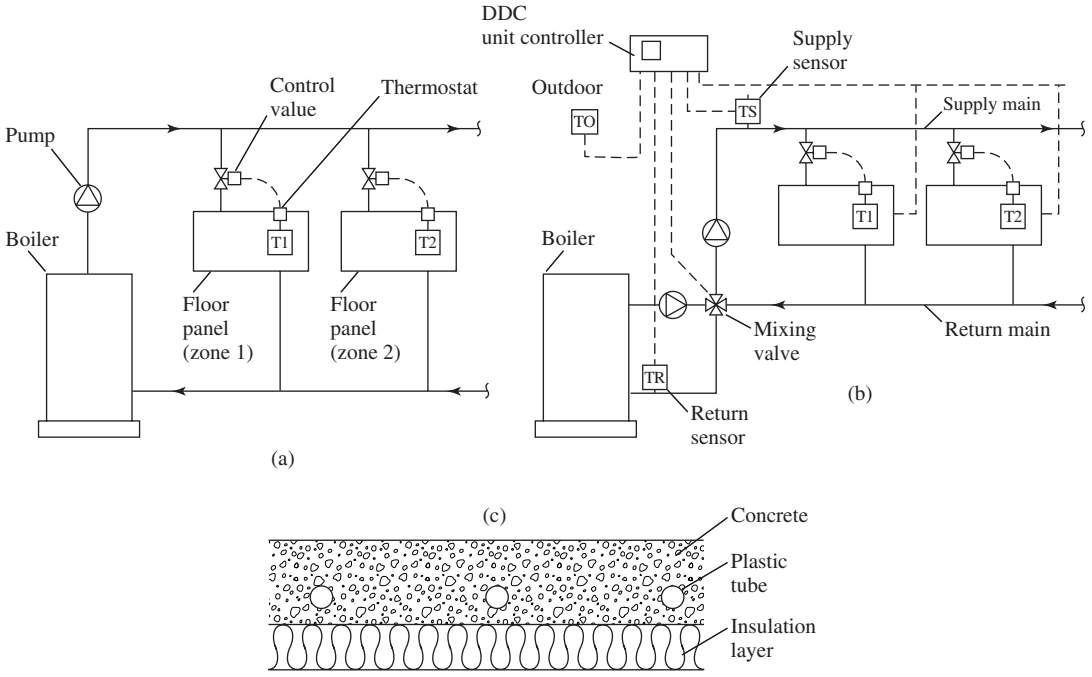
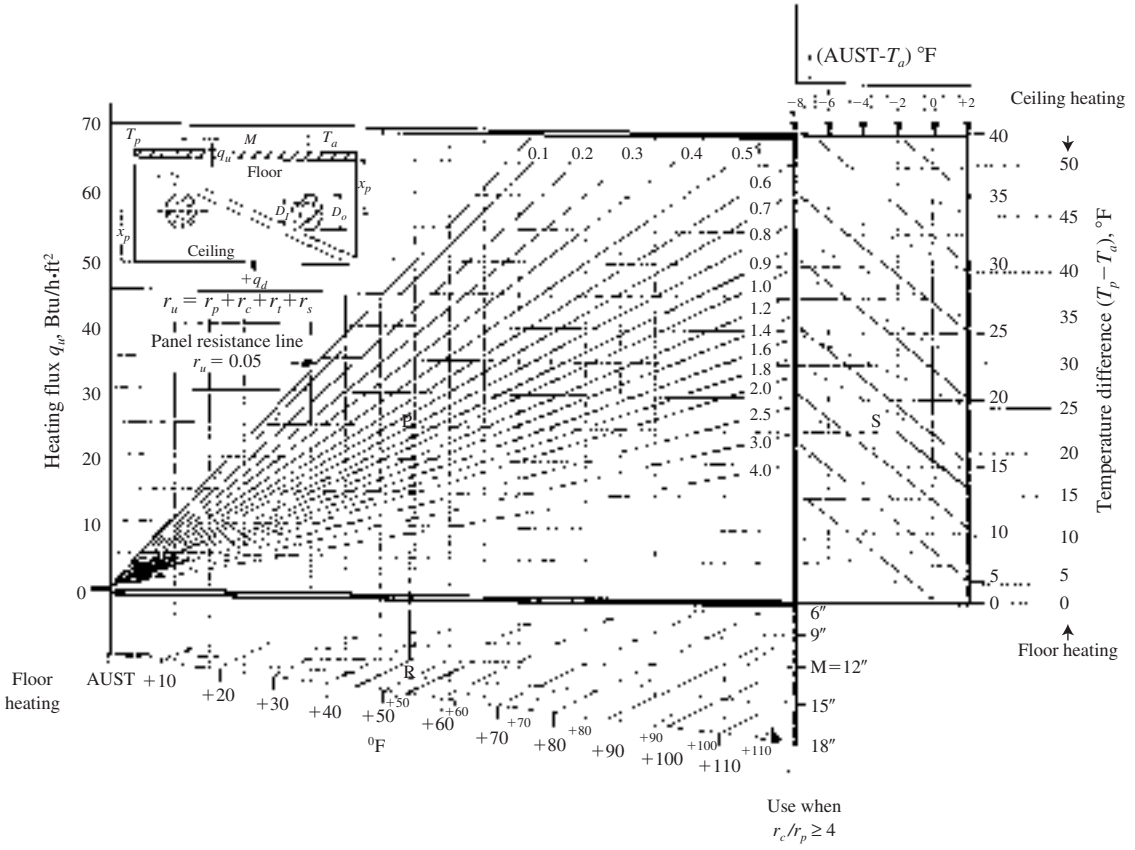


FIGURE 8.9 Hydronic radiant floor heating systems. (*a*) Zone control valves; (*b*) outdoor reset plus zone control valves; (*c*) floor panel; (*d*) design nomograph. *(Source: ASHRAE Transactions 1995 Part I, p. 1204, Kilkis and Coley. Reprinted by permission.)*



(d)

FIGURE 8.9 (*Continued*)

where T_p = effective panel surface temperature, °F (°C)
T_a = space air temperature that surrounds floor panel, °F (°C)

Then the heat flux from the radiant heating floor panel to the space air due to the combined effect of radiation and convection q_u, Btu / h · ft² (W/m²), can be calculated as

$$q_u = q_r + q_c \tag{8.15}$$

If the tubes are embedded in the concrete slab with a tube spacing of 1 ft (12 in. or 0.3 m), the characteristic panel R value r_u, h · ft² · °F/Btu (m² · °C/W), can be calculated as

$$r_u = r_t M' + r_p + r_c \tag{8.16}$$

where r_t = thermal resistance of tube wall per unit tube spacing, h · ft² · °F/Btu (m² · °C/W)
M' = tube spacing, ft (m)
r_p = R value of floor panel, h · ft² · °F/Btu (m² · °C/W)
r_c = R value of floor-panel covering, h · ft² · °F/Btu (m² · °C/W)

Design Nomograph

Kilkis and Coley (1995) noted that the mean hot water temperature needed to maintain a required heat flux from the heating panel depends primarily upon the heating panel's effective surface temperature, its tube spacing, and its thermal resistance. Kilkis and Coley also recommended a design nomograph for the heating floor panel to determine the required design parameters, as shown in Fig. 8.9*d*, which comprises the following parameters:

- Mean hot water temperature $T_{w,m}$, = (AUST + $T_{w,\text{add}}$), °F (°C); here, $T_{w,\text{add}}$ indicates the additional water temperature,°F (°C).
- Heat flux released from the floor panel $q_u = Q_{rh}/A_p$, Btu/h·ft^2/(W/m^2). Here Q_{rh} represents the space heating load, Btu/h (W), and A_p the area of the floor panel(s), ft^2 (m^2).
- Tube spacing M, in. (mm).
- Temperature differential AUST $- T_a$, °F (°C).
- Panel temperature differential $T_p - T_a$, °F (°C).

During the design of a hydronic radiant floor heating system, for a given indoor design space air temperature T_a, first AUST $- T_a$ must be predicted. For a heating space having small outdoor exposures like windows, skylights, external walls, and roof, AUST $\approx T_a$. For a room equipped with a normal size of external window and outdoor exposures, AUST $- T_a$ can be predicted to be -2°F (1.1°C).

For a given room, its winter indoor design temperature T_a is 68°F (20°C), the heat flux released from the floor panel q_u is 25 Btu/h·ft^2 (W/m^2), tube spacing $M = 12$ in. (300 mm), r_u is 0.8 h·ft^2·°F/Btu (0.14 m^2·°C/W), and $r_c/r_p < 4$. Then at the intersection of horizontal line $q_u = 25$ Btu/h·ft^2 (79 W/m^2) and incline line $r_u = 0.8$ h·ft^2·°F/Btu (0.14 m^2·°C/W), point P (Fig. 8.9*d*), draw a vertical line downward that intersects the tube spacing line $M = 12$ in. (300 mm) at point R. The reading of $T_{w,\text{add}}$ is then 45°F (7.2°C). The required mean hot water temperature is

$$T_{w,m} = \text{AUST} + T_{w,\text{add}} = 68 - 2 + 45 = 111°\text{F}\ (43.9°\text{C})$$

To determine the effective panel surface temperature T_p (floor temperature), extend the horizontal line of $q_u = 25$ Btu/h·ft^2 (79 W/m^2) until it intersects the vertical line AUST $- T_a = -2$°F at point S. From Fig. 8.9*d*, $T_p - T_a = 13$°F (7.2°C). Therefore, $T_p = T_a + 16 = 68 + 13 = 81$°F (27.2°C).

Design Considerations

- Because of a higher space mean radiant temperature when a hydronic radiant floor heating system is used, a lower winter indoor design temperature T_a, such as 68°F (20.0°C), is often adopted.
- For occupant comfort, the surface temperature of the floor heating panels T_p in occupied space should be less than 84°F (28.9°C).
- In an interior heating space if only a hydronic floor heating system is installed instead of a cooling/heating (air conditioning) system, an outdoor ventilating system is required to provide outdoor air for the occupants.
- Setback of the space temperature during nighttime unoccupied periods produces less satisfactory results for heavy floor heating panels. The surface temperature of the floor panels cannot drop quickly when the temperature is set back and cannot pick up quickly either during the warm-up period the next morning.

Control and Operations of Multizone Hydronic Radiant Floor-Panel Heating System

Gibbs (1994) simulated and compared two different control strategies for hydronic radiant floor heating systems:

- *Pulse-width-modulated zone control*. In this control, hot water is supplied from the boiler at a constant temperature, as shown in Fig. 8.9*a*. A room thermostat with a proportional control mode has a 3°F (1.7°C) proportional band and pulse-width-modulates the zone control valves 4 times per hour. If the zone air temperature is at a set point T_{set} of 68°F (20°C), the on time of the zone control valve is 7.5 min and the off time is also 7.5 min.

  During a cold day in winter, hot water at a constant temperature of 120°F (48.9°C) is supplied to various zones and returned at a temperature of 100°F (37.8°C). The surface of the floor panel is maintained at a mean temperature of 82°F (27.8°C), and the space air temperature of zone 1 is $T_{rl} = 68$°F (20°C). As soon as the space heating load reduces, T_{rl} increases to 68.3°F (20.2°C), the thermostat in zone 1 pulse-width-modulates the control valve so that it is now on 6 min and off 9 min.

  Pulse-width-modulated zone control is simple, effective, and lower in first cost. However, the boiler is not protected from cold return water.
- *Outdoor reset plus pulse-width-modulated zone control*. In this control, a four-way mixing valve is added to pulse-width-modulaed zone control, and a proportional-integral (PI) control is used instead of proportional control, as shown in Fig. 8.9*b*. The hot water supply temperature T_{ws} from the boiler is no longer constant; T_{ws} is reset according to outdoor temperature T_o. The zone requires that the hottest supply temperature provide the indoor temperature feedback to the outdoor reset control. When the zone air temperature is at the set point $T_{rl} = T_{set}$, the control valve is on (opened) all the time and cycles off only when the $T_{rl} > T_{set}$.

  To prevent excessive thermal stress in the boiler, when the return temperature is too cold to enter the boiler and is sensed by the return temperature sensor *TR* as shown in Fig. 8.9*b*, part of the return water is mixed with the recirculated hot water supply, until the return water temperature is raised above a safe limit.

  Outdoor reset plus pulse-width-modulated zone control is safer and more flexible and has more continuously distributed hot water. On the other hand, it is more expensive.

System Characteristics and Applications

Compared with a warm air heating system, a hydronic radiant floor-panel heating system has the following characteristics:

- It provides a better thermal comfort for occupants, because of less cold draft, and a higher mean radiant temperature and an even temperature distribution.
- There is a lower natural infiltration rate and a lower winter indoor design temperature, and therefore, it provides greater potential for saving energy.
- The peak load may be shaved (leveled) due to the thermal energy stored in the floor panels.
- Response to the load changes and heating capacity modulation are slower.
- It is higher in first cost if a separate outdoor ventilating system is installed.

Because of the use of plastic and rubber tubes for floor panels in recent years, radiant floor-panel heating systems were used by many residences in Europe since 1980s, as well as for buildings in locations where winter heating is dominant.

8.9 INFRARED HEATING

Basics

Infrared heating uses radiant heat transfer from a gas-fired or electrically heated high-temperature tube or panel to provide heating to a localized area for the comfort of the occupants or the maintenance of a

suitable environment for a manufacturing process. Heat radiates from an infrared heater in the form of electromagnetic waves in all directions. Most infrared heaters have reflectors to focus the radiation on a localized target; hence, they are often known as beam radiant heaters. Infrared heating is widely used in factories, warehouses, garages, gymnasiums, skating rinks, outdoor loading docks, and racetrack stands.

In environments with a low indoor ambient air temperature during the winter, the warmth that an occupant feels depends on the radiant energy absorbed by the occupant from all sources that have temperatures higher than that of the indoor ambient air. This warmth is closely related to the effective radiant flux I_{rad}, or ERF Btu/h·ft^2/(W/m^2). And ΔERF indicates the effective radiant flux that must be added to an unheated space, Btu/h·ft^2/(W/m^2). ERF and ΔERF can be calculated as

$$\begin{aligned} \text{ERF} &= h_r(T_{\text{rad}} - T_a) \\ &= h\,(T_o - T_a) \end{aligned} \tag{8.17a}$$

$$\Delta\text{ERF} = h(T_o - T_{uo}) \tag{8.17b}$$

where h_r = radiative heat-transfer coefficient, Btu/h·ft^2·°F (W/m^2·°C)
h = convective and radiative combined heat-transfer coefficient, Btu/h·ft^2·°F (W/m^2·°C)
T_{rad} = mean radiant temperature, °F (°C)
T_a = indoor ambient air temperature, °F (°C)
T_o = operative temperature as defined in Eq. (4.6), °F (°C)
T_{uo} = operative temperature of unheated space, °F (°C)

In Eq. (8.17), $T_{\text{rad}} - T_a$ indicates the amount T_{rad} is above the indoor ambient air T_a; it is important because it influences the infrared radiant heat transfer to the occupant. There are two types of infrared heaters: gas infrared heaters and electric infrared heaters.

Gas Infrared Heaters

Gas infrared heaters can be divided into indirect infrared heaters and porous matrix infrared heaters. An indirect infrared heater consists of a burner, a radiating tube, and a reflector. Combustion takes place within the radiating tube at a temperature up to 1200°F (650°C). In a porous matrix infrared radiation heater, a gas-air mixture is supplied to an enclosure and distributed evenly through a porous ceramic, stainless-steel, or metallic screen that is exposed at the other end. Combustion takes place at the exposed surface and has a maximum temperature of about 1600°F (870°C).

Gas infrared heaters are usually vented and have a small conversion efficiency. Only 10 to 20 percent of the energy output of an open combustion gas infrared heater is in the form of infrared radiant energy. Gas infrared heaters should not be operated under conditions in which the ambient air contains ignitable gas or materials that may decompose to hazardous or toxic gases or vapors.

Adequate combustion air must be provided. Venting is preferred in order to prevent a buildup of combustion products. Usually, 4 cfm (1.9 L/s) of makeup air is required for 1000 Btu/h (293 W) of gas input. If unvented infrared heaters are used, humidity and condensation control should be provided to account for the accumulation of water vapor that forms during combustion.

A thermostat usually controls the supply of gas by means of a gas valve in on/off mode. For standing-pilot ignition, a sensing element and controller are also used to cut off the gas supply when the pilot flame extinguishes. If the combustion air is blocked, the gas supply is also cut off for safety.

Electric Infrared Heaters

Electric infrared heaters are usually made of nickel-chromium wire or tungsten filament inside an electrically insulated metal tube or quartz tube with or without an inert gas. The heaters also contain

a reflector, which directs the radiant beam to the area that needs heating. Nickel-chromium wires often operate at a temperature between 1200 and 1800°F (650 and 980°C). The tungsten filament can stand a temperature as high as 4000°F (2200°C) while the outer envelope is at a temperature of about 1000°F (540°C).

Electric infrared heaters also use a thermostat to switch on or cut off the electric current to control the input to the nickel-chromium electric heater. Some input controllers can preset the on/off time period. For a quartz tube, the output can be controlled by varying its voltage supply. As with gas infrared heaters, electric infrared heaters should not be used where there is a danger of igniting flammable dust or vapors or decomposing contaminated matter into toxic gases.

Electric infrared heaters have a far higher infrared radiant energy conversion efficiency than gas infrared heaters. They are clean and more easily managed. Although the cost of electric energy is several times higher than that of natural gas, a comprehensive analysis should be conducted before selecting an electric or gas infrared heater.

Design and Layout

Determination of Watt Density. According to tests, an acceptable temperature increase $T_{\text{rad}} - T_a$ for normal clothed occupants in an indoor environment using infrared heating is often between 20 and 25°F (11.1 and 13.9°C). The effective radiant flux, or required watt density, provided by the infrared heaters has been determined from field experiments, with the data listed in Table 8.2 for different operating conditions.

Single or Multiple Heaters. Based on the required coverage area and the width and length of the floor area that can be covered with the specified watt density listed in Table 8.3, it is possible to determine whether a single heater or multiple heaters should be installed. It is also necessary to select the coverage pattern: three-fourths overlap or full overlap. If there are many occupants within the covering area or if the occupant will stay for a long period, a full-overlap pattern is preferable.

Mounting Height and Clearances. The mounting height of infrared heaters should not be below 10 ft (3 m); otherwise, the occupant may feel discomfort from the radiant beam overhead. Because of the spread of the radiant beam, a higher mounting height results in a smaller watt density and a greater coverage, and allows a greater spacing between heaters. An optimum mounting height should be selected based on the ceiling height of the building or the outdoor structures.

Adequate clearance (recommended by the manufacturer) between the infrared heater and any combustible material, especially between the heater and the roof, must be maintained to prevent fire.

Other Considerations. When infrared heaters are used for total indoor space heating, the following must be taken into account:

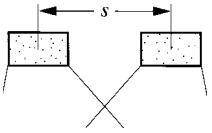
TABLE 8.2 Required Watt Density, W/ft^2

Temperature increase $T_{\text{rad}}-T_a$, q_c	Tight uninsulated building	Drafty indoors or large glass area	Loading area, one end open	Outdoor shielded, less than 5 mph wind speed	Outdoor unshielded, less than 10 mph wind speed
20	30	40	50	55	60
25	37	50	62	70	75
30	45	60	75	85	90
40	60	80	100	115	120
50	75	100	125	145	150

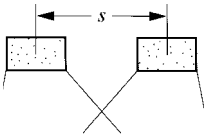
***Source*:** Abridged with permission from *Handbook of HVAC Design 1990*, Chapter 7 (Lehr Associates).

TABLE 8.3 Watt Density and Coverage

Two asymmetric heaters



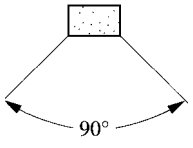
Three-fourths overlap

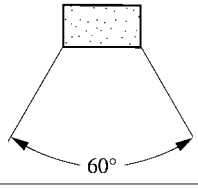


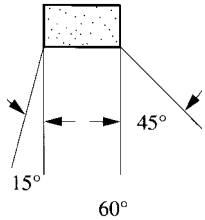
Full overlap

Mounting height, ft	Watt density, W/ft^2	$W \times L$, ft	Spacing s, ft	Wattt density, W/ft^2	$W \times L$, ft	Spacing s, ft
10	33	13 × 12	8.5	38	11 × 12	6
11	27	14 × 13	9	33	12 × 13	6.5
12	24	15 × 14	10	29	12 × 14	7
13	21	16 × 15	10.5	25	13 × 15	7.5
14	18	17 × 16	11	22	14 × 16	8
15	16	18 × 17	12	20	15 × 17	8.5
16	14	20 × 18	12.5	18	16 × 18	9
18	11	22 × 20	14	14	18 × 20	10
20	9.6	24 × 22	15.5	11	20 × 22	11
22	8.0	26 × 24	17	9.6	22 × 24	12
25	6.4	30 × 27	19	8	24 × 27	13.5

Single heater







Asymmetric

Mounting height, ft	Watt density, W/ft^2	$W \times L$, ft	Wat density, W/ft^2	$W \times L$, ft	Wattt density, W/ft^2	$W \times L$, ft
10	18	12 × 12	25	8.5 × 12	19	11 × 12
11	15	13 × 13	22	9 × 13	16	12 × 13
12	13	14 × 14	18	10 × 14	14	12 × 14
13	11	15 × 15	16	10 × 15	13	13 × 15
14	10	16 × 16	14	11 × 16	11	14 × 16
15	9	17 × 17	13	12 × 17	9.6	15 × 17
16	8	18 × 18	11	12 × 18	8.8	16 × 18
18	6.4	20 × 20	8.8	14 × 20	7.2	18 × 20
20	5.2	22 × 22	7.6	15 × 22	5.8	20 × 22
22	4.4	24 × 24	6.2	17 × 24	4.8	24 × 24
25	3.5	27 × 27	5.0	19 × 27	3.9	24 × 27

W = width, L = length, both in ft.

***Source*:** Abridged with permission from *Handbook of HVAC Design 1990*, Chapter 7 (Lehr Associates).

- The space heating load can be lower because of a lower space temperature.
- The area near the external walls needs a greater intensity of infrared heating.
- Heaters are often mounted around the perimeter at a suitable height and are usually arranged 15 to 20 ft (4.5 to 6 m) from the corners of the building.
- The key to successful design is to supply a proper amount of heat evenly to the occupying zone.

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CHAPTER 9
REFRIGERANTS, REFRIGERATION CYCLES, AND REFRIGERATION SYSTEMS

9.1 REFRIGERATION AND REFRIGERATION SYSTEMS 9.2
9.2 REFRIGERANTS 9.3
Refrigerants, Cooling Media, and Liquid Absorbents 9.3
Azeotropic, Near Azeotropic, and Zeotropic 9.3
Numbering of Refrigerants 9.4
9.3 PROPERTIES AND CHARACTERISTICS OF REFRIGERANTS 9.5
Safety Requirements 9.5
Effectiveness of Refrigeration Cycle 9.5
Evaporating and Condensing Pressures 9.6
Oil Miscibility 9.6
Inertness 9.6
Thermal Conductivity 9.6
Refrigeration Capacity 9.6
Physical Properties 9.6
Operating Characteristics 9.6
9.4 PHASEOUT OF OZONE DEPLETION REFRIGERANTS 9.7
Refrigerant Use 9.7
Ozone Depletion and Global Warming Potentials 9.7
Phaseout of CFCs, Halons, and HCFCs 9.10
Montreal Protocol and Clean Air Act 9.10
Action and Measures 9.11
Status of CFC Replacements 9.13
9.5 CLASSIFICATION OF REFRIGERANTS 9.13
Hydrofluorocarbons 9.13
Azeotropic HFC 9.14
Near-Azeotropic HFC 9.14
Zeotropic HFC 9.15
HCFCs and Their Zeotropes 9.15
Inorganic Compounds 9.16
CFCs, Halons, and Their Zeotropes 9.16
9.6 REFRIGERATION PROCESSES AND REFRIGERATION CYCLES 9.16
Refrigeration Processes 9.16
Refrigeration Cycles 9.17
Unit of Refrigeration 9.17
9.7 GRAPHICAL AND ANALYTICAL EVALUATION OF REFRIGERATION 9.17
Pressure-Enthalpy Diagram 9.17
Temperature-Entropy Diagram 9.18
Analytical Evaluation of Cycle Performance 9.19
9.8 CARNOT REFRIGERATION CYCLE 9.19
Performance of Carnot Refrigeration Cycle 9.19
9.9 COEFFICIENT OF PERFORMANCE OF REFRIGERATION CYCLE 9.21
9.10 SINGLE-STAGE IDEAL VAPOR COMPRESSION CYCLE 9.22
Flow Processes 9.22
Cycle Performance 9.22
Determination of Enthalpy by Polynomials 9.24
Refrigeration Effect, Refrigerating Load, and Refrigerating Capacity 9.25
9.11 SUBCOOLING AND SUPERHEATING 9.26
Subcooling 9.26
Superheating 9.26
9.12 MULTISTAGE VAPOR COMPRESSION SYSTEMS 9.29
Compound Systems 9.29
Interstage Pressure 9.30
Flash Cooler and Intercooler 9.31
9.13 TWO-STAGE COMPOUND SYSTEM WITH A FLASH COOLER 9.31
Flow Processes 9.31
Fraction of Evaporated Refrigerant in Flash Cooler 9.31
Enthalpy of Vapor Mixture Entering Second-Stage Impeller 9.32
Coefficient of Performance 9.33
Characteristics of Two-Stage Compound System with Flash Cooler 9.33
9.14 THREE-STAGE COMPOUND SYSTEM WITH A TWO-STAGE FLASH COOLER 9.35
Flow Processes 9.35
Fraction of Refrigerant Vaporized in Flash Cooler 9.35
Coefficient of Performance of Three-Stage System 9.38
9.15 TWO-STAGE COMPOUND SYSTEM WITH A VERTICAL INTERCOOLER 9.38
Comparison between Flash Cooler and Vertical Coil Intercooler 9.40

9.16 CASCADE SYSTEMS 9.40
Advantages and Disadvantages 9.40
Performance of Cascade System 9.42
9.17 AIR EXPANSION REFRIGERATION CYCLES 9.45
Thermodynamic Principle 9.45
Flow Processes of a Basic Air Expansion System for Aircraft 9.47
Air Expansion Cycle 9.48
9.18 REFRIGERATION SYSTEMS—CLASSIFICATIONS AND DEVELOPMENTS 9.49
Classifications 9.49
Recent Developments 9.51
9.19 REFRIGERATION COMPRESSORS 9.51
Positive Displacement and Non–Positive Displacement Compressors 9.51
Hermetic, Semihermetic, and Open Compressors 9.53
Direct Drive, Belt Drive, and Gear Drive 9.53
9.20 PERFORMANCE OF COMPRESSORS 9.53
Volumetric Efficiency 9.53
Motor, Mechanical, and Compression Efficiency 9.54
Isentropic and Polytropic Analysis 9.54
Energy Use Index 9.55
9.21 SAFETY REQUIREMENTS AND MACHINERY ROOM 9.56
Refrigerant Safety 9.56
Application Rules for High-Probability Systems 9.56
Application Rules for Low-Probability Systems 9.57
Refrigerating Systems of 100 hp (74.6 kW) or less 9.58
Refrigerating Machinery Room 9.58
Storage of Refrigerants 9.59
REFERENCES 9.59

9.1 REFRIGERATION AND REFRIGERATION SYSTEMS

Refrigeration is defined as the process of extracting heat from a lower-temperature heat source, substance, or cooling medium and transferring it to a higher-temperature heat sink. Refrigeration maintains the temperature of the heat source below that of its surroundings while transferring the extracted heat, and any required energy input, to a heat sink, atmospheric air, or surface water.

A refrigeration system is a combination of components and equipment connected in a sequential order to produce the refrigeration effect. The refrigeration systems commonly used for air conditioning can be classified by the type of input energy and the refrigeration process as follows:

1. *Vapor compression systems.* In vapor compression systems, compressors activate the refrigerant by compressing it to a higher pressure and higher temperature level after it has produced its refrigeration effect. The compressed refrigerant transfers its heat to the sink and is condensed to liquid form. This liquid refrigerant is then throttled to a low-pressure, low-temperature vapor to produce refrigerating effect during evaporation. Vapor compression systems are the most widely adopted refrigeration systems in both comfort and process air conditioning.
2. *Absorption systems.* In an absorption system, the refrigeration effect is produced by thermal energy input. After absorbing heat from the cooling medium during evaporation, the vapor refrigerant is absorbed by an absorbent medium. This solution is then heated by direct-fired furnace, waste heat, hot water, or steam. The refrigerant is again vaporized and then condensed to liquid to begin the refrigeration cycle again.
3. *Air or gas expansion systems.* In an air or gas expansion system, air or gas is compressed to a high pressure by mechanical energy. It is then cooled and expanded to a low pressure. Because the temperature of air or gas drops during expansion, a refrigeration effect is produced.

9.2 REFRIGERANTS

Refrigerants, Cooling Media, and Liquid Absorbents

Refrigerants. A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure. Most refrigerants undergo phase changes during heat absorption—evaporation—and heat releasing—condensation.

Cooling Media. A cooling medium is the working fluid cooled by the refrigerant to transport the cooling effect between a central plant and remote cooling units and terminals. In a large, centralized system, it is often more economical to use a coolant medium that can be pumped to remote locations where cooling is required. Chilled water, brine, and glycol are used as cooling media in many refrigeration systems. The cooling medium is often called a secondary refrigerant, because it obviates extensive circulation of the primary refrigerant.

Liquid Absorbents. A solution known as liquid absorbent is often used to absorb the vaporized refrigerant (water vapor) after its evaporation in an absorption refrigeration system. This solution, containing the absorbed vapor, is then heated at high pressure. The refrigerant vaporizes, and the solution is restored to its original concentration for reuse.

Lithium bromide and ammonia, both in a water solution, are the liquid absorbents used most often in absorption refrigerating systems.

Azeotropic, Near Azeotropic, and Zeotropic

A refrigerant can either be a single chemical compound or a mixture (blend) of multiple compounds.

Azeotropic. These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do not change their volumetric composition or saturation temperature when they evaporate or condense at a constant pressure. Components in a mixture of azeotropes cannot be separated from their constituents by distillation. Properties of azeotropic refrigerants are entirely different from those of their components and may be conveniently treated as a single chemical compound.

Near Azeotropic. Near-azeotropic refrigerants are blends whose characteristics are near to azeotropic. Although properties of near-azeotropic refrigerants are nearer to azeotropic than to nonazeotropic (zeotropic), near-azeotropic refrigerants are defined as zeotropic or nonazeotropic.

Zeotropic. These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do change volumetric composition or saturation temperature when they evaporate or condense at a constant pressure.

Blends. Mixtures of refrigerants of two or more chemical compounds are blends. The advantage of a blend of multiple chemical compounds compared to a single compound is that the required properties of the blend can possibly be achieved by varying the fractional composition of the components.

Glide. Zeotropic mixtures, including near-azeotropic blends, show changes in composition because of the leaks, the difference between liquid and vapor phases, or the difference between the charge and circulation, or their combined effect. The shift in composition causes the change in evaporating and condensing temperature and pressure. The difference in dew point and bubble point

in the temperature-concentration diagram of a zeotropic refrigerant during evaporation and condensation is called *glide*, expressed in °F (°C). A near-azeotropic refrigerant has a smaller glide than a zeotropic one. The midpoint between the dew point and bubble point is usually taken as the evaporating or condensing temperature for a nonazeotropic and near-azeotropic refrigerant. Hwang et al. (1997) showed that temperature drops during condensation and temperature increases during evaporation. Ideal or perfect azeotropic refrigerants are uncommon, whereas near-azeotropic ones are fairly common.

Numbering of Refrigerants

Before the invention of chlorofluorocarbons (CFCs), refrigerants were called by their chemical names. Because of the complexity of these names, especially the CFCs, the fully halogenated CFCs, and hydrochlorofluorocarbons (HCFCs), the not fully halogenated HCFCs (see Fig. 9.1), a numbering system was developed for hydrocarbons and halocarbons, and is used widely in the refrigeration industry. According to ANSI/ASHRAE Standard 34-1997, the first digit from the right is the number of fluorine atoms in the compound. The second digit from the right is one more than the number of hydrogen atoms in the compound. The third digit from the right is one less than the number of the carbon atoms in the compound. If the digit is zero, it is omitted from the number. The fourth digit from the right is the number of unsaturated carbon-carbon bonds in the compound. If the digit is zero it is also omitted from the number. For example, the chemical formula of HCFC-123 is $CHCl_2CF_3$:

There are 3 fluorine atoms, first digit from the right is	3
There is 1 hydrogen atom, second digit from the right is	$1 + 1 = 2$
There are 2 carbon atoms, third digit from the right is	$2 - 1 = 1$
No unsaturated C—C bonds, the fourth digit from the right is	0

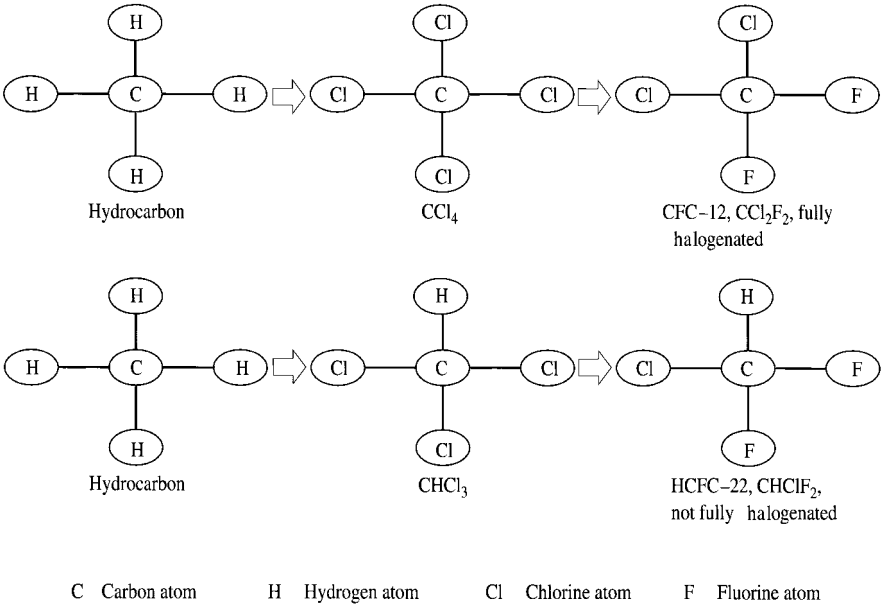


FIGURE 9.1 Fully halogenated CFCs and not fully halogenated HCFCs.

9.3 PROPERTIES AND CHARACTERISTICS OF REFRIGERANTS

Today, the preservation of the ozone layer is the first priority of refrigerant selection. In addition, the global warming effect and the following factors should be considered.

Safety Requirements

Refrigerant may leak from pipe joints, seals, or component parts during installation, operation, or accident. Therefore, refrigerants must be acceptably safe for humans and manufacturing processes, with little or no toxicity or flammability.

In ANSI/ASHRAE Standard 34-1997, the toxicity of refrigerants is classified as class A or B. Class A refrigerants are of lower toxicity. A class A refrigerant is one whose toxicity has not been identified when its concentration is less than or equal to 400 ppm, based on threshold limit value–time-weighted average (TLV-TWA) or equivalent indices. The TLV-TWA concentration is a concentration to which workers can be exposed over an 8-h workday and a 40-h workweek without suffering adverse effect. Concentration ppm means parts per million by mass.

Class B refrigerants are of high toxicity. A class B refrigerant produces evidence of toxicity when workers are exposed to a concentration below 400 ppm based on a TLV-TWA concentration. Flammable refrigerants explode when ignited. If a flammable refrigerant is leaked in the area of a fire, the result is an immediate explosion. Soldering and welding for installation or repair cannot be performed near such gases.

ANSI/ASHRAE Standard 34-1997 classifies the flammability of refrigerants into classes 1, 2, and 3. Class 1 refrigerants show no flame propagation when tested in air at a pressure of 14.7 psia (101 kPa) at 65°F (18.3°C). Class 2 refrigerants have a lower flammability limit (LFL) of more than 0.00625 lb/ft^3 (0.1 kg/m^3) at 70°F (21.1°C) and 14.7 psia (101 kPa abs.), and a heat of combustion less than 8174 Btu/lb (19,000 kJ/kg). Class 3 refrigerants are highly flammable, with an LFL less than or equal to 0.00625 lb/ft^3 (0.1 kg/m^3) at 70°F (21.1°C) and 14.7 psia (101 kPa abs.) or a heat of combustion greater than or equal to 8174 Btu/lb (19,000 kJ/kg).

A refrigerant's safety classification is its combination of toxicity and flammability. According to ANSI/ASHRAE Standard 34-1997, safety groups are classified as follows:

- A1 lower toxicity and no flame propagation
- A2 lower toxicity and lower flammability
- A3 lower toxicity and higher flammability
- B1 higher toxicity and no flame propagation
- B2 higher toxicity and lower flammability
- B3 higher toxicity and higher flammability

For zeotropic blends whose flammability and toxicity may change as their composition changes, a dual safety classification should be determined. The first classification denotes the classification of the formulated composition of the blend. The second classification lists the classification of the blend composition at the worst case of fractionation.

Effectiveness of Refrigeration Cycle

The effectiveness of refrigeration cycles, or *coefficient of performance* (COP), is one parameter that affects the efficiency and energy consumption of the refrigeration system. It will be clearly defined in a later section. The COP of a refrigeration cycle using a specific refrigerant depends mainly upon the isentropic work input to the compressor at a given condensing and evaporating pressure differential, as well as the refrigeration effect produced.

Evaporating and Condensing Pressures

It is best to use a refrigerant whose evaporating pressure is higher than that of the atmosphere so that air and other noncondensable gases will not leak into the system and increase the condensing pressure. The condensing pressure should be low because high condensing pressure necessitates heavier construction of the compressor, piping, condenser, and other components. In addition, a high-speed centrifugal compressor may be required to produce a high condensing pressure.

Oil Miscibility

When a small amount of oil is mixed with refrigerant, the mixture helps to lubricate the moving parts of a compressor. Oil should be returned to the compressor from the condenser, evaporator, accessories, and piping, in order to provide continuous lubrication. On the other hand, refrigerant can dilute oil, weakening its lubricating effect; and when the oil adheres to the tubes in the evaporator or condenser, it forms film that reduces the rate of heat transfer.

Inertness

An inert refrigerant does not react chemically with other materials, thus avoiding corrosion, erosion, or damage to the components in the refrigerant circuit.

Thermal Conductivity

The thermal conductivity of a refrigerant is closely related to the efficiency of heat transfer in the evaporator and condenser of a refrigeration system. Refrigerant always has a lower thermal conductivity in its vapor state than in its liquid state. High thermal conductivity results in higher heat transfer in heat exchangers.

Refrigeration Capacity

The cubic feet per minute (cfm) suction vapor of refrigerant required to produce 1 ton of refrigeration (liters per second to produce 1 kW of refrigeration) depends mainly on the latent heat of vaporization of the refrigerant and the specific volume at the suction pressure. It directly affects the size and compactness of the compressor and is one of the criteria for refrigerant selection.

Physical Properties

Discharge Temperature. A discharge temperature lower than 212°F (100°C) is preferable because temperatures higher than 300°F (150°C) may carbonize lubricating oil or damage some of the components.

Dielectric Properties. Dielectric properties are important for those refrigerants that will be in direct contact with the windings of the motor (such as refrigerants used to cool the motor windings in a hermetically sealed compressor and motor assembly).

Operating Characteristics

Leakage Detection. Refrigerant leakage should be easily detected. If it is not, gradual capacity reduction and eventual failure to provide the required cooling will result. Most of the currently used refrigerants are colorless and odorless. Leakage of refrigerant from the refrigeration system is often detected by the following methods:

- *Halide torch*. This method is simple and fast. When air flows over a copper element heated by a methyl alcohol flame, the vapor of halogenated refrigerant decomposes and changes the color of the flame (green for a small leak, bluish with a reddish top for a large leak).
- *Electronic detector*. This type of detector reveals a variation of electric current due to ionization of decomposed refrigerant between two oppositely charged electrodes. It is sensitive, but cannot be used where the ambient air contains explosive or flammable vapors.
- *Bubble method*. A solution of soup or detergent is brushed over the seals and joints where leakage is suspected, producing bubbles that can be easily detected.

9.4 PHASEOUT OF OZONE DEPLETION REFRIGERANTS

Refrigerant Use

The use of CFCs and HCFCs is a global concern. Approximately two-thirds of all fully halogenated CFCs were used outside the United States in the mid-1980s. In 1985, the total use of halocarbons in the United States was 611 million lb (0.28 million ton). These halocarbons were used in foam insulation, automotive air conditioners, new systems of Air Conditioning and Refrigeration Institute (ARI) members, and other products. Foam insulation blown by CFCs was the largest user. Automotive air conditioners made up 19 percent of the total and CFCs purchased by ARI members for new systems made up 5 percent of the total use. Of the CFCs and HCFCs purchased by ARI members, HCFC-22 made up 77 percent, while CFC-11 and CFC-12 each made up about 10 percent.

Ozone Depletion and Global Warming Potentials

To compare the relative ozone depletion caused by various refrigerants, an index called the *ozone depletion potential* (ODP) has been proposed. ODP is the ratio of the rate of ozone depletion of 1 lb of any halocarbon to that of 1 pound of CFC-11. The ODP of CFC-11 is assigned a value of 1. The following are the ODP values for various refrigerants:

Refrigerant	Chemical formula	ODP value
CFC-11	CCl_3F	1.0
CFC-12	CCl_2F_2	1.0
CFC-13B1	$CBrF_3$	10
CFC-113	CCl_2FCClF_2	0.8
CFC-114	$CClF_2CClF_2$	1.0
CFC-115	$CClF_2CF_3$	0.6
CFC/HFC-500	CFC-12 (73.8%)/HFC-152a (26.2%)	0.74
CFC/HCFC-502	HCFC-22 (48.8%)/CFC-115 (51.2%)	0.33
HCFC-22	$CHClF_2$	0.05
HCFC-123	$CHCl_2CF_3$	0.02
HCFC-124	$CHClFCF_3$	0.02
HCFC-142b	CH_3CClF_2	0.06
HFC-125	CHF_2CF_3	0
HFC-134a	CF_3CH_2F	0
HFC-152a	CH_3CHF_2	0

Similar to the ODP, the *halocarbon global warming potential* (HGWP) is the ratio of calculated warming for each unit mass of gas emitted to the calculated warming for a unit mass of reference gas CFC-11. The HGWPs of various refrigerants are listed in Table 9.1. In addition to the HGWP, another global warming index uses CO_2 as a reference gas. For example, 1 lb of HCFC-22 has the

TABLE 9.1 Properties of Commonly Used Refrigerants at 40°F Evaporating and 100°F Condensing

		Chemical formula	Molecular mass	Ozone depletion potential (ODP)	Halocarbon global warming potential (HGWP)	Evaporating pressure, psia	Condensing pressure, psia	Compression ratio	Refrigeration effect, Btu/lb
Hydrofluorocarbons (HFCs)									
HFC-32	Difluoromethane	CH_2F_2	52.02	0	0.11	135.6	340.2	2.51	
HFC-125	Pentafluoroethane	CHF_2CF_3	120.02	0	0.84	112.4	276.8	2.46	36.4
HFC-134a	Tetrafluoroethane	CF_3CH_2F	102.3	0	0.28	49.8	138.9	2.79	
HFC-143a	Trifluoroethane	CH_3CF_3	84.0	0	1.1				
HFC-152a	Difluoroethane	CHF_2CH_3	66.05	0	0.03	44.8	124.3	2.77	
HFC-245ca	Pentafluoropropane	$CF_3CF_2CH_3$	134.1	0	0.09				
Azeotropic HFC									
HFC-507	HFC-125/HFC-143a(45/55)			0	0.98				
HFC-507A	HFC-125/HFC-143a(50/50)			0		104.6	257.6	2.46	
Near-azeotropic HFC									
HFC-404A	HFC-125/HFC-143a/ HFC-134a (44/52/4)			0	0.95	109.16	251.18	2.30	
HFC-410A	HFC-32/HFC-125(50/50)			0	0.48	132.90	332.33	2.50	
HFC Zeotropic									
HFC-407A	HFC-32/HFC-125/ HFC-134a (20/40/40)			0	0.47				
HFC 407C	HFC-32/HFC-125/ HFC-134a (23/25/52)			0	0.38	86.13	225.22	2.61	
Hydrochlorofluorocarbons (HCFCs) and their zeotropes									
HCFC-22	Chlorodifluoromethane	$CHClF_2$	86.48	0.05	0.34	82.09	201.5	2.46	69.0
HCFC-123	Dichlorotrifluoroethane	$CHCl_2CF_3$	153.0	0.016	0.02	5.8	20.8	3.59	62.9
HCFC-124	Chlorotetrafluoroethane	$CHFClCF_3$	136.47	0.02	0.10	27.9	80.92	2.90	52.1
Near-azeotropic HCFC									
HCFC-402A	HCFC-22/HFC-125/ PRO-290 (38/60/2)			0.02					
Zeotropic HCFC									
HCFC-401A	HCFC-22/HCFC-124/ HFC-152a (53/34/13)			0.037	0.22				
HCFC-401B	HCFC-22/HCFC-124/ HFC-152a (61/28/11)			0.04	0.24				
Inorganic compounds									
R-717	Ammonia	NH_3	17.03	0	0	71.95	206.81	2.87	467.4
R-718	Water	H_2O	18.02	0					
R-729	Air		28.97	0					
Chlorofluorocarbons (CFCs), halons, and their azeotropes									
CFC-11	Trichlorofluoromethane	CCl_3F	137.38	1.00	1.00	6.92	23.06	3.33	68.5
CFC-12	Dichlorodifluoromethane	CCl_2F_2	120.93	1.00	3.1	50.98	129.19	2.53	50.5
BFC-13B1	Bromotrifluoromethane	$CBrF_3$	148.93	10					
CFC-113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.39	0.80	1.4	2.64	10.21	3.87	54.1
CFC-114	Dichlorotetrafluoroethane	CCl_2FCF_3	170.94	1.00	3.9	14.88	45.11	3.03	42.5
CFC-500	CFC-12/HFC-152a (73.8/26.2)		99.31	0.74		59.87	152.77	2.55	60.5
CFC-502	HCFC-22/CFC-115 (48.8/51.2)		111.63	0.22	3.7				

***Source*:** Adapted with permission from *ASHRAE Handbook 1997, Fundamentals*, and ANSI/ASHRAE Standard 34-1992 and Addenda 1997.

Alternatives		Trade name	Specific volume of suction vapor, ft^3/lb	Compressor, cfm/ton	Power consumption, hp/ton	Critical temperature, °F	Discharge temperature, °F	Flammability	Safety
Hydrofluorocarbons (HFCs)									
HFC-32			0.63			173.1		Lower flammability	A2
HFC-125			0.33			150.9	103	Nonflammable	A1
HFC-134a	CFC-12, HCFC-22		0.95			213.9		Nonflammable	A1
HFC-143a								Lower flammability	A2
HFC-152a			1.64			236.3		Lower flammability	A2
HFC-245ca	CFC-11, HCFC-123					353.1			
Azeotropic HFC									
HFC-507	CFC-502, CFC-12	Genetron AZ-50						Nonflammable	A1
HFC-507A			0.43					Nonflammable	A1
Near-azeotropic HFC									
HFC-404A	CFC-502, CFC-12	SUVA HP-62	0.44						A1/A1
HFC-410A	HCFC-22, CFC-502	AZ-20	0.45						A1/A1
Zeotropic HFC									A1/A1
HFC-407A	CFC-502, CFC-12	KLEA60, AC-9000							A1/A1
HFC-407C	HCFC-22, CFC-502	KLEA66	0.63						
Hydrochlorofluorocarbons (HFCs) and their zeotropes									
HCFC-22			0.66	1.91	0.696	204.8	127	Nonflammable	A1
HCFC-123	CFC-11		5.88	18.87	0.663	362.6		Nonflammable	B1
HCFC-124			1.30	5.06	0.698	252.5		Nonflammable	A1
Near-azeotropic HCFC									
HCFC-402A	CFC-502	SUVAHP-80							A1/A1
Zeotropic HCFC									
HCFC-401A	CFC-12	MP-39							A1/A1
HCFC-401B	CFC-12	MP-66							A1/A1
Inorganic compounds									
R-717			3.98	1.70	0.653	271.4	207	Lower flammability	B2
R-718								Nonflammable	A1
R-729								Nonflammable	
Chlorofluorocarbons (CFCs), halons, and their azeotropes									
CFC-11			5.43	15.86	0.636	388.4	104	Nonflammable	A1
CFC-12			0.79	3.08	0.689	233.6	100	Nonflammable	A1
BFC-13B1			0.21			152.6	103	Nonflammable	A1
CFC-113			10.71	39.55	0.71	417.4	86	Nonflammable	A1
CFC-114			2.03	9.57	0.738	294.3	86	Nonflammable	A1
CFC-500	CFC-12/HFC152a (73.8/26.2)		0.79	3.62	0.692	221.9	105	Nonflammable	A1
CFC-502	HCFC-22/CFC115 (48.8/51.2)						98	Nonflammable	A1

same effect on global warming as 4100 lb (1860 kg) of CO_2 in the first 20 years after it is released into the atmosphere. Its impact drops to 1500 lb (680 kg) at 100 years.

Phaseout of CFCs, Halons, and HCFCs

The theory of depletion of the ozone layer was proposed in 1974 by Rowland and Molina. (The 1995 Nobel Prize was awarded to F. Sherwood Rowland, Mario Molina, and Paul Crutzen for their work in atmospheric chemistry and theory of ozone depletion.) Network station in Halley Bay, Antarctica, established a baseline trend of ozone levels that helped scientists to discover the ozone hole in 1985. National Aeronautics and Space Administration (NASA) flights into the stratosphere over the arctic and antarctic circles found CFC residue where the ozone layer was damaged. Approximately the same ozone depletion over the antarctic circle was found in 1987, 1989, 1990, and 1991. By 1988, antarctic ozone levels were 30 percent below those of the mid-1970s. The most severe ozone loss over the antarctic was observed in 1992. Ground monitoring at various locations worldwide in the 1980s has showed a 5 to 10 percent increase in ultraviolet radiation. Although there is controversy about the theory of ozone layer depletion among scientists, as discussed in Rowland (1992), action must be taken immediately before it is too late.

Montreal Protocol and Clean Air Act

In 1978, the Environmental Protection Agency (EPA) and the Food and Drug Administration (FDA) of the United States issued regulations to phase out the use of fully halogenated CFCs in nonessential aerosol propellants, one of the major uses at that time. On September 16, 1987, the European Economic Community and 24 nations, including the United States, signed the Montreal Protocol. This document is an agreement to phase out the production of CFCs and halons by the year 2000. The Montreal Protocol had been ratified by 157 parties.

The Clean Air Act Amendments, signed into law in the United States on November 15, 1990, governed two important issues: the phaseout of CFCs and a ban (effective July 1, 1992) on the deliberate venting of CFCs and HCFCs. Deliberate venting of CFCs and HCFCs must follow the regulations and guidelines of the EPA. In February 1992, then-President Bush called for an accelerated phaseout of the CFCs in the United States. Production of CFCs must cease from January 1, 1996, with limited exceptions for service to certain existing equipment.

In late November 1992, representatives of 93 nations meeting in Copenhagen also agreed to the complete cessation of CFC production beginning January 1, 1996, and of halons by January 1, 1994, except continued use from existing (reclaimed or recycled) stock in developed nations. In addition, the 1992 Copenhagen amendments and later a 1995 Vienna meeting revision agreed to restrict the production of HCFCs relative to a 1989 level beginning from 2004 in developed nations according to the following schedule:

Date	Production limit
January 1, 1996	100 percent cap Cap = 2.8 percent of ODP of 1989 CFC consumption plus total ODP of 1989 HCFC consumption
January 1, 2004	65 percent cap
January 1, 2010	35 percent cap
January 1, 2015	10 percent cap
January 1, 2020	0.5 percent cap
January 1, 2030	Complete cessation of production

Consumption indicates the production plus imports minus exports and feedstocks. The value of 2.8 percent cap is the revised value of the Vienna meeting in 1995 to replace the original value of

3.1 percent in the Copenhagen amendments. The Copenhagen amendments had been ratified by 58 parties.

Action and Measures

The impact of CFCs on the ozone layer poses a serious threat to human survival. The following measures are essential:

Conversions and Replacements. Use alternative refrigerants (substitutes) to replace the CFCs in existing chillers and direct-expansion (DX) systems. During the conversion of the CFC to non–ozone depletion alternative refrigerants, careful analysis should be conducted of capacity, efficiency, oil miscibility, and compatibility with existing materials after conversion. For many refrigeration systems that already have a service life of more than 15 years, it may be cost-effective to buy a new one using non-CFC refrigerant to replace the existing refrigeration package.

- HFC-134a and HCFC-22 are alternative refrigerants to replace CFC-12.
- HCFC-123, and HFC-245ca are alternative refrigerants to replace CFC-11 in large chillers.

It is important to realize that HCFC-123 and HCFC-22 themselves are interim refrigerants and will be restricted in consumption beginning in 2004. HCFC-123 has a very low global warming potential and is widely used in centrifugal chillers. HCFC-22 is widely used in small and medium-size DX systems.

- HFC-134a, HFC-407C, and HFC-410A are alternative refrigerants to replace HCFC-22. HFC-407C is a near-azeotropic refrigerant of HFC-32/HFC-125/HFC-134a (23/25/52) [means (23%/25%/525)], and HFC-410A also a near-azeotropic refrigerant of HFC-32/HFC-125 (50/50).
- HFC-245ca or another new HFC possibly developed before 2004 will be the hopeful alternative to replace HCFC-123.

In supermarkets, CFC-502 is a blend of HCFC-22/CFC-115 (48.8/51.2).

- HFC-404A, HFC-507, and HFC-410A are alternative refrigerants to replace CFC-502. HFC-404A is a near-azeotropic refrigerant of HFC-125/HFC-143a/HFC-134a(44/52/4); and HFC-507 is an azeotropic refrigerant of HFC-125/HFC-143a (45/55).

Reducing Leakage and Preventing Deliberate Venting. To reduce the leakage of refrigerant from joints and rupture of the refrigeration system, one must detect the possible leakage, tighten the chillers, improve the quality of sealing material, and implement preventive maintenance.

Prevent the deliberate venting of CFCs and HCFCs and other refrigerants during manufacturing, installation, operation, service, and disposal of the products using refrigerants.

Avoid CFC and HCFC emissions through recovery, recycle, and reclaiming. According to ASHRAE Guideline 3-1990, recovery is the removal of refrigerant from a system and storage in an external container. Recycle involves cleaning the refrigerant for reuse by means of an oil separator and filter dryer. In reclamation, refrigerant is reprocessed for new product specifications.

To avoid the venting of CFCs and HCFCs and other refrigerants, an important step is to use an ARI-certified, portable refrigerant recovering/recycling unit to recover all the liquid and remaining vapor from a chiller or other refrigeration system. An outside recovery/reclaiming service firm may also be employed. A typical refrigerant recovery unit is shown in Fig. 9.2. It includes a recovery cylinder, a vacuum pump or compressor, a water-cooled condenser, a sight glass, a shutoff float switch, necessary accessories, pipes, and hoses.

To recover refrigerant from a chiller that has been shut down involves two phases: liquid recovery and vapor recovery. Liquid recovery is shown in Fig. 9.2*a*. The vacuum pump or compressor in the recovery unit creates a low pressure in the recovery cylinder. Liquid refrigerant is then extracted from the bottom of the chiller into the recovery cylinder. If the recovery cylinder is not large enough, the shutoff float switch ceases to operate the vacuum pump or compressor when the

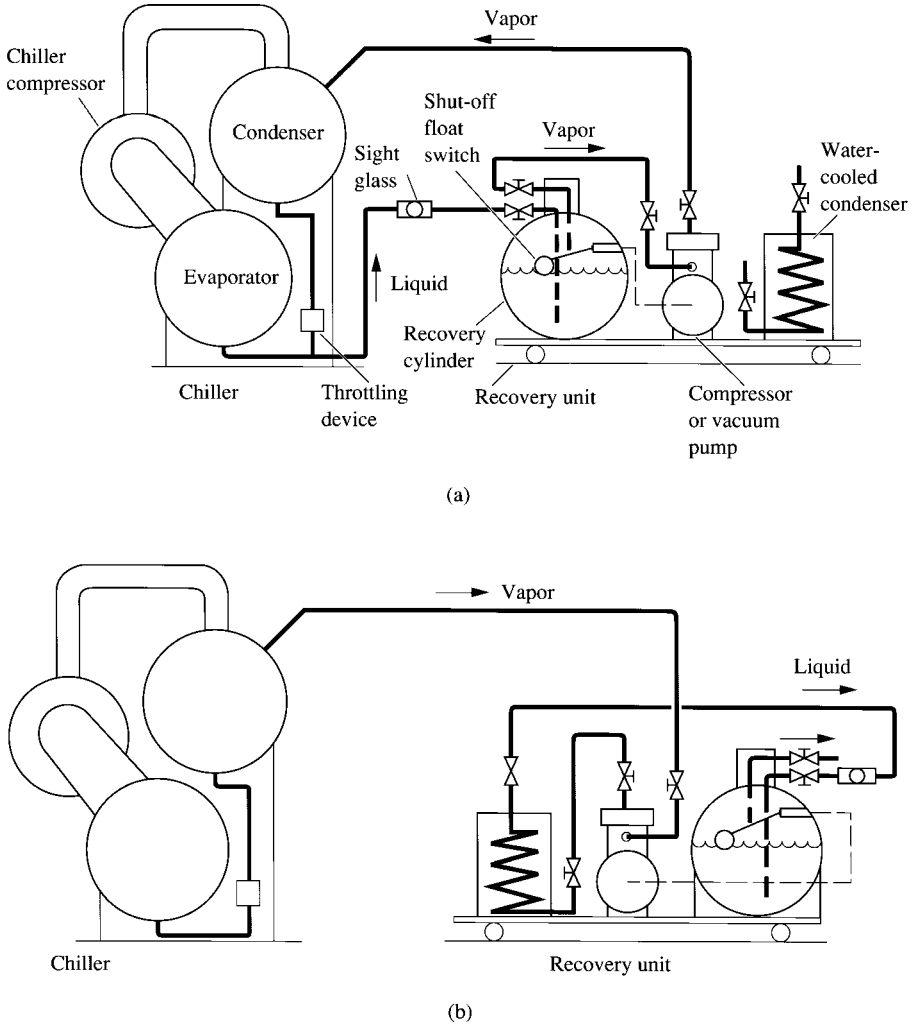


FIGURE 9.2 A typical refrigerant recovery unit: (*a*) liquid recovery; (*b*) vapor recovery.

recovery cylinder is 80 percent full. Another empty recovery cylinder is used to replace the filled cylinder. If the vapor enters the sight glass, which means that the liquid refrigerant is all extracted, then the vacuum pump or compressor is stopped and the vapor recovery phase begins.

Vapor recovery is shown in Fig. 9.2*b*. The vacuum pump or compressor extracts the refrigerant vapor from the top of the chiller. Extracted refrigerant vapor is then condensed to liquid form that flows through the water-cooled condenser and is stored in the recovery cylinder. Noncondensable gases are purged into the atmosphere from the recovery unit. Water at a temperature between 40 and 85°F (4.4 and 29.4°C) is often used as the condensing cooling medium. The recovered refrigerant can be recycled or reclaimed as required.

In addition to the recovery of refrigerants from the chiller or other refrigeration system, refrigerant vapor detectors should be installed at locations where refrigerant from a leak is likely to concentrate. These detectors can set off an alarm to notify the operator to seal the leak.

Because of the worldwide effort to phase out CFCs, the latest result of a study conducted by the National Oceanic and Atmospheric Administration (NOAA) and published in the journal *Science* was reported by UPI science writer Susan Milius ". . . scientists are cheering a 1 percent reduction during 1995 in the chemicals that slowly carry chlorine and bromine aloft to the stratosphere." (*Air Conditioning, Heating and Refrigeration News*, June 10, 1996, p. 2).

Status of CFC Replacements

Dooley (1997) showed that according to the ARI survey, in the United States there were about 80,000 CFC large-tonnage chillers in 1992, and most used CFC-11 as refrigerant. At the beginning of 1997, some 18,981 chillers, or 24 percent of the total 80,000, had been phased-out CFC refrigerants. Of these, 4813 chillers were converted to non-CFC refrigerant, and 14,168 chillers were replaced by new chillers which used non-CFC refrigerant. The ratio of new replacements to conversions was about 3 to 1. It is often cost-effective to replace an old chiller instead of to convert the CFC refrigerant to alternatives in an existing chiller.

The ARI estimates that 53 percent of the 80,000 chillers in 1992 will remain in service on January 1, 2000. This shows that the actual phaseout process was slower than expected.

For CFC-12, automotive cooling and supermarkets acount for more than 90 percent of their servicing requirements. Due to the vibrations and unsteady operating conditions, automotive cooling required greater servicing losses and a faster phaseout schedule than large-tonnage chillers. The annual amount of CFCs required to compensate for servicing losses came from stockpiles of virgin CFCs and reclaimed CFCs. Because of the slower phaseout of CFC chillers, servicing demands were greater than could be supplied from the reclaimed CFCs. The using up of the stockpiles of virgin CFCs and thus a shortage of CFC supply may occur at the begining of the twenty-first century in the United States. The slower phaseout of CFCs indicated that there is a possibility of a considerably longer period of servicing of HCFC equipment than called for in the production phaseout schedule in the twenty-first century.

9.5 CLASSIFICATION OF REFRIGERANTS

Before the introduction of chlorofluorocarbons in the 1930s, the most commmonly used refrigerants were air, ammonia, sulfur dioxide, carbon dioxide, and methyl chloride. Until 1986, nontoxic and nonflammable halogenated hydrocarbons with various ozone depletion potentials were used almost exclusively in vapor compression refrigeration systems for air conditioning. The impact of ozone depletion of CFCs, halons, and HCFCs since the 1980s caused a worldwide decision to phase out these refrigerants. A new classification of refrigerants into six groups based mainly on ozone depletion will be helpful for the selection of non–ozone depletion refrigerants as well as replacement of CFCs by alternative refrigerants (Table 9.1).

Hydrofluorocarbons

HFCs contain only hydrogen, fluorine, and carbon atoms. They contain no chlorine atoms, therefore are environmentally safe, and cause no ozone depletion. They are designated by the prefix *HFC*.

HFC-134a is an attractive, long-term alternative to replace CFC-12 in reciprocating, scroll, screw, and centrifugal compressors; and a long-term alternative for HCFC-22. It has a low 0.28 HGWP. HFC-134a is nonflammable, has an extremely low toxicity, and is classified as AI in ANSI/ASHRAE Standard 34-1997 safety rating.

HFC-134a has a molecular mass of 102.3 instead of CFC-12's molecular mass of 120.93. At a condensing temperature of 100°F (37.8°C), HFC-134a's condensing pressure is 138.83 psia (957 kPa abs.), whereas CFC-12's is 131.65 psia (908 kPa abs.). A larger impeller of higher speed is

needed for a centrifugal chiller to provide the same cooling capacity. Parsnow (1993) reported a capacity loss of direct conversion from CFC-12 to HFC-134a of 8 to 10 percent, and an efficiency loss of 1 to 2 percent. In Lowe and Ares (1995), in the conversion from CFC-12 to HFC-134a in the Sears Tower centrifugal chillers, the compressor's speed increased about 8.5 percent, there was a cooling capacity loss of 12 to 24 percent, and efficiency was 12 to 16 percent worse.

HFC-134a has a poor mutual solubility with mineral oil because of a higher interfacial tension between them. Polyolester-based synthetic lubricants should be used. Polyolester-based synthetic oils are hygroscopic, so monitoring of the moisture content of the refrigerant is important. Halocarbons, including HFC-134a, are compatible with containment materials. Concerning nonmetallic or elastomer (such as gaskets) compatibility, Corr et al. (1993) reported that HFC-134a, an ester-based synthetic oil mixture, has a smaller volume change of elastomer than CFC-12 and mineral oil. HFC-134a may become one of the most widely used single-chemical-compound refrigerants during the first half of the twenty-first century.

HFC-245ca also does not contain chlorine and bromine atoms, and its ODP is 0. Compared to CFC-11, its efficiency will be 3 to 4 percent lower. Synthetic polyolester lubricant oil will be used. Except for neoprene at high moisture content, common materials used in the refrigeration system were shown to be compatible with HFC-245ca in tests. Smith et al. (1993) reported that mixtures of HFC-245ca in air with a relative humidity of 43 percent and a HFC-245ca concentration range of 7 to 14.4 percent were observed to be flammable in tests.

Because of the higher isentropic work required by HFC-245ca compared to CFC-11 and HCFC-123, for direct-drive centrifugal chillers, a large impeller is required during the conversion from CFC-11 to HFC-245ca or from HCFC-123 to HFC-245ca.

HFC-245ca is a possible long-term alternative to CFC-11 and HCFC-123 in large centrifugal chillers in the future. In the HFC group, HFC-32, HFC-125, HFC-143a, and HFC-152a all are seldom used as a refrigerant of single compound only.

Azeotropic HFC

HFC-507 is an azeotrope of refrigerant blends of HFC-125/HFC-143a (45/55) of zero ozone depletion and an HGWP of 0.96. It is a long-term alternative refrigerant to replace CFC-502 and CFC-12 in low-temperature refrigeration systems whose evaporating temperatures are below −10°F (−12.2°C). HFC-507 needs synthetic lubricant oil. According to ANSI/ASHRAE Standard 34-1997, HFC-507 is allowed alternative designations for HFC-507A, a refrigeration blend of HFC-125/HFC-143a (50/50).

The Linton et al. (1995) test results showed that compared to CFC-502, the cooling capacity of HFC-507 was between 0.95 and 1.05. HFC-507 had an energy efficiency of 0.87 to 0.97 compared to CFC-502.

Near-Azeotropic HFC

Near-azeotropic HFC is a refrigerant blend of zero ozone depletion and having rather small changes in volumetric composition or saturation temperature, a small glide, when it evaporates or condenses at a constant pressure. Near-azeotropic HFC-404A and HFC-410A require synthetic lubricant oil instead of mineral oil and are nontoxic and nonflammable with a safety classification of A1/A1.

HFC-404A is a blend of HFC-125/HFC-143a/HFC-134a (44/52/4) of zero ozone depletion and an HGWP of 0.94. It is a long-term alternative refrigerant for CFC-502 and CFC-12 both in low-temperature refrigeration systems. HFC-404A has a temperature glide of 0.9°F (0.5°C) during evaporation and a temperature glide of 0.6°F (0.33°C) during condensation. Snelson et al. (1995) compared HFC-404A with CFC-502 from their test results. HFC-404A had the same, slightly higher, or lower evaporating capacity at various condensing and evaporating temperatures. The energy efficiency ratio of 0.89 to 0.99 was found at different evaporating temperatures T_{ev}. The lower the T_{ev}, the lower the energy efficiency ratio, because of the higher compressor pressure ratio.

HFC-410A is a blend of HFC-32/HFC-125 (50/50) of zero ozone depletion and an HGWP of 0.43. It is a long-term alternative refrigerant to replace HCFC-22 and CFC-502. HFC-410A has a temperature glide of 0.2°F (0.11°C) during evaporation and condensation. Hickman (1994) showed that the compressor displacement, cfm/ton (L/s·kW), for HFC-410A is about 50 percent smaller than that for HCFC-22; and the discharge pressure for 130°F (54.4°C) condensing is about 490 psia (3379 kPa abs.), which is much higher than that for HCFC-22. It is often necessary to change the original reciprocating compressor to a scroll compressor. A higher energy efficiency was reported by a refrigerant manufacturer.

Zeotropic HFC

Zeotropic (nonazeotropic) HFCs are refrigerant blends of zero ozone depletion that have greater temperature glide during evaporation and condensation. Zeotropic HFC-407A and HFC-407C also require synthetic lubrication oil, instead of mineral oil; and both are nontoxic and nonflammable with a safety classification of A1/A1.

HFC-407A is a blend of HFC-32/HFC-125/HFC-134a (20/40/40) of zero ozone depletion with an HGWP of 0.49. It is a long-term alternative refrigerant for CFC-502 and CFC-12 in low-temperature refrigeration sysems. HFC-407A showed a reduction in heat transfer in the evaporator of a low-temperature system during tests. The system performance of HFC-407A was the lowest compared to HFC-404A and HFC-507.

HFC-407C is a blend of HFC-32/HFC-125/134a (23/25/52) of zero ozone depletion with an HGWP of 0.38. It is a long-term alternative refrigerant to replace HCFC-22 and CFC-502. Bivens et al. (1994) compared HFC-407C to HCFC-22 during tests. For cooling and heating, the capacity ratio ranged from 0.93 to 1.06, and the energy ratio ranged from 0.94 to 0.97. In-tube heat-transfer coefficients during evaporation and condensation were 85 to 95 percent of HCFC-22 values.

HCFCs and Their Zeotropes

HCFCs contain hydrogen, chlorine, fluorine, and carbon atoms and are not fully halogenated. HCFCs have a much shorter atmospheric life than CFCs and cause far less ozone depletion (0.02 to 0.1 ODP). They are designated by the prefix *HCFC*. Their consumption is scheduled to be reduced gradually starting from 2004 and will be completely phased out in 2030 in developed nations, except for a limited amount for service, as discussed previously.

HCFC-22 has an ODP of 0.05 and an HGWP of 0.40. It is nonflammable with a safety classification of A1. HCFC-22 is partially miscible with mineral oil. At 40°F (4.4°C), its evaporating pressure is 82.09 psia (566 kPa abs.), and at 100°F (37.8°C) its condensing pressure is 201.5 psia (1389 kPa abs.), the highest of currently used HCFC and CFC refrigerants. HCFC-22 has a smaller compressor displacement among the HCFCs and CFCs. All these factors make it an interim alternative to replace CFC-12. HCFC-22 was the most widely used refrigerant in reciprocating and scroll compressors in small and medium-size packaged units in the 1990s in the United States.

HCFC-123 is an interim alternative to replace CFC-11 in low-pressure centrifugal chillers. It has an ODP of 0.02 and a very low HGWP of 0.02. HCFC-123 is nonflammable and of lower toxicity with a safety classification of B1. In 1997, DuPont raised the *allowable exposure limit* (AEL) of HCFC-123 to 50 ppm. Smithhart and Crawford (1993) reported that for chillers with direct conversion from CFC-11 to HCFC-123, there was about a decrease of 0 to 5 percent in capacity and a 2 to 4 percent decrease in efficiency. A conversion of refrigerant from CFC-11 to HCFC-123 in an existing chiller may require changing its lubricants, seals, and motor windings of hermetic compressors.

Because HCFC-123 has a low ODP, a very low HGWP, and a 15 percent higher energy efficiency in centrifugal chillers than HFC-134a does, if no acceptable alternative can be found, the use of HCFC-123 in centrifugal chillers may be considered longer than the cap specified in the Vienna meeting in 1995, as listed in Sec. 9.4, in the twenty-first century.

HCFC-124 has an ODP of 0.02. It is nonflammable and has a safety classification of A1. HCFC-124 is an interim alternative refrigerant to replace CFC-114.

Near-azeotropic HCFC-402A is a blend of HCFC-22/HFC-125/PRO-290 (38/60/2) with an ODP of 0.02 and an HGWP of 0.63. Here PRO-290 represents propane, which is a more highly flammable refrigerant with a safety classification of A3. HCFC-402A is nonflammable and has a safety classification of A1/A1. It needs polyolester or alkyl-benzene-based lubricant oil. HCFC-402A is an interim alternative refrigerant to replace CFC-502.

Zeotropic HCFC-401A is a blend of HCFC-22/HCFC-124/HFC-152a (53/34/13) with an ODP of 0.037 and an HGWP of 0.22; and HCFC-401B is a blend of HCFC-22/HCFC-124/HFC-152a (61/28/11) with an ODP of 0.04 and an HGWP of 0.24. Both HCFC-401A and HCFC-401B are nonflammable and have a safety classification of A1/A1. They both need alkyl-benzene-based lubricant oil. HCFC-401A is an interim alternative refrigerant to replace CFC-12, and HCFC-401B is an interim alternative refrigerant to replace CFC-12 in low-temperature refrigeration systems.

Inorganic Compounds

These compounds include ammonia (NH_3), water (H_2O), and gases used in the gas expansion systems. As refrigerants, they were used far earlier than the halocarbons. Air is a mixture of nitrogen, oxygen, argon, rare gases, and water vapor. Air has zero ozone depletion and is a zeotropic blend that has a temperaure glide of 5.5°F (3°C) at atmospheric pressure. Ammonia also has zero ozone depletion. It has a high operating pressure at 40°F (4.4°C) evaporating and 100°F (37.8°C) condensing. Ammonia compressors show a smaller cfm/ton displacement and higher energy efficiency than HCFC-22 compressors. Leakage of ammonia is easily detected due to its objectionable odor.

Ammonia attacks copper even in the presence of a small amount of moisture. It is of higher toxicity. An ammonia-air mixture is flammable if the concentration of NH_3 by volume is within 16 to 25 percent. The mixture may explode if the ignition source is above 1200°F (650°C). Because the safety classification of ammonia is B2—lower flammability and higher toxicity—it is not allowed to be used in comfort air conditioning in the United States.

Water has a zero ODP and is readily available. At 40°F (4.4°C) evporating and 100°F (37.8°C) condensing, water's evaporating and condensing pressures are both below atmospheric. Air and other noncondensable gases must be purged out of the refrigeration system periodically.

CFCs, Halons, and Their Zeotropes

CFCs including CFC-11, CFC-12, CFC-113, and CFC-114, have an ODP from 0.8 to 1.0. Halons including BFC-13B1 have an ODP of 10. Their azeotropics CFC-500 and CFC-502 have ODPs of 0.74 and 0.22, respectively. Production of all these CFCs, halons, and their azeotropes ceased in developed nations since January 1, 1996. However, a limited amount of these refrigerants, used to service the refrigeration systems that have not been converted or replaced by non–ozone depletion refrigerants, may be extended to the beginning of the twenty-first century.

9.6 REFRIGERATION PROCESSES AND REFRIGERATION CYCLES

Refrigeration Processes

A refrigeration process indicates the change of thermodynamic properties of the refrigerant and the energy transfer between the refrigerant and the surroundings. The following refrigeration processes occur during the operation of a vapor compression refrigerating system:

- *Evaporation.* In this process, the refrigerant evaporates at a lower temperature than that of its surroundings, absorbing its latent heat of vaporization.
- *Superheating*. Saturated refrigerant vapor is usually superheated to ensure that liquid refrigerant does not flow into the compressor.

- *Compression*. Refrigerant is compressed to a higher pressure and temperature for condensation.
- *Condensation*. Gaseous refrigerant is condensed to liquid form by being desuperheated, then condensed, and finally subcooled, transferring its latent heat of condensation to a coolant.
- *Throttling and expansion*. The higher-pressure liquid refrigerant is throttled to the lower evaporating pressure and is ready for evaporation.

The following refrigeration processes occur during the operation of an air or gas expansion refrigeration system:

- *Compression*. Air or gas is compressed to a higher pressure and temperature.
- *Heat release*. Heat is released to the surroundings at constant pressure in order to reduce the temperature of the air or gas.
- *Throttling and expansion*. Air or gas is throttled and expanded so that its temperature is lowered.
- *Heat absorption*. Heat is absorbed from the surroundings because of the lower air or gas temperature.

Refrigeration Cycles

Most refrigerants undergo a series of evaporation, compression, condensation, throttling, and expansion processes, absorbing heat from a lower-temperature reservoir and releasing it to a higher-temperature reservoir in such a way that the final state is equal in all respects to the initial state. It is said to have undergone a closed refrigeration cycle. When air or gas undergoes a series of compression, heat release, throttling, expansion, and heat absorption processes, and its final state is not equal to its initial state, it is said to have undergone an open refrigeration cycle.

Both vapor compression and air or gas expansion refrigeration cycles are discussed in this chapter. Absorption refrigeration cycles are covered in Chap. 14.

Unit of Refrigeration

In inch-pound (I-P) units, refrigeration is expressed in British thermal units per hour, or simply Btu/h. A *British thermal unit* is defined as the amount of heat energy required to raise the temperature of one pound of water one degree Fahrenheit from 59°F to 60°F; and 1 Btu/h = 0.293 watt (W).

Another unit of refrigeration widely used in the HVAC&R industry is ton of refrigeration, or simply ton. As mentioned before, 1 ton = 12,000 Btu/h of heat removed. This equals the heat absorbed by 1 ton (2000 lb) of ice melting at a temperature of 32°F over 24 h.

Because the heat of fusion of ice at 32°F is 144 Btu/lb,

$$1 \text{ ton} = \frac{1 \times 2000 \times 144}{24} = 12{,}000 \text{ Btu/h}$$

also 1 ton = 3.516 kW

9.7 GRAPHICAL AND ANALYTICAL EVALUATION OF REFRIGERATION

Pressure-Enthalpy Diagram

The pressure-enthalpy *p-h* diagram is the most common graphical tool for analysis and calculation of the heat and work transfer and performance of a refrigeration cycle. A single-stage refrigeration

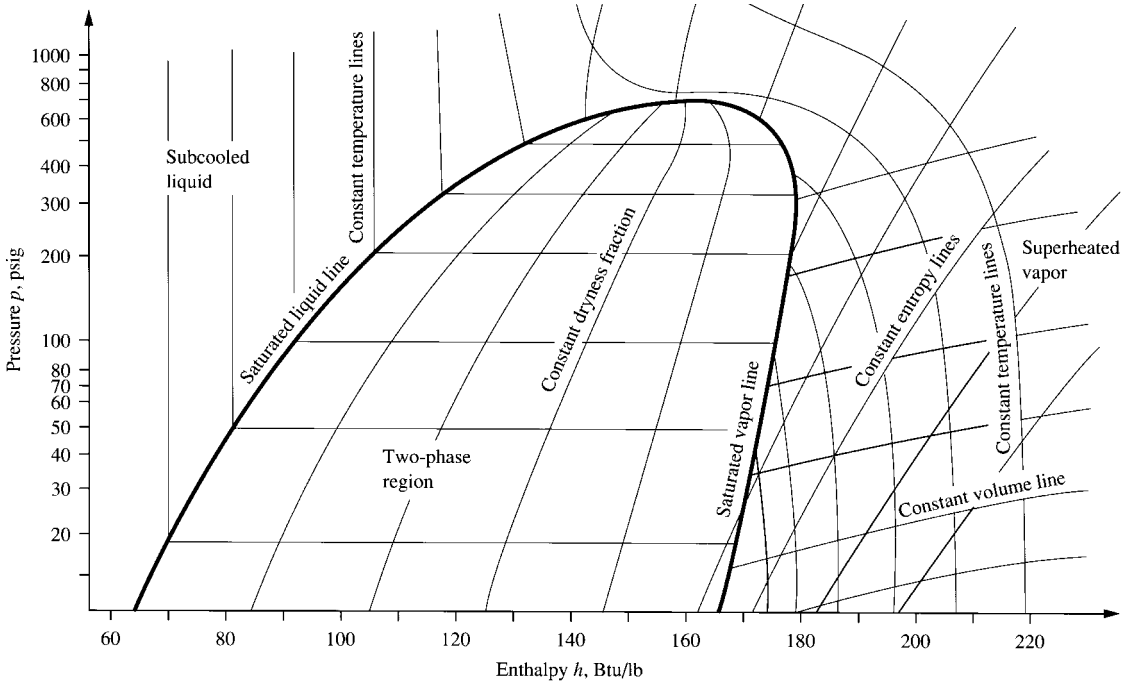


FIGURE 9.3 Skeleton of pressure-enthalpy *p-h* diagram for HCFC-22.

cycle consists of two regions: the high-pressure region, or high side, and the low-pressure region, or low side. The change in pressure can be clearly illustrated on the *p-h* diagram. Also, both heat and work transfer of various processes can be calculated as the change of enthalpy and are easily shown on the *p-h* diagram.

Figure 9.3 is a skeleton *p-h* diagram for refrigerant HCFC-22. Enthalpy h (in Btu/lb) is the abscissa, and absolute pressure (psia) or gauge pressure (psig), both expressed in logarithmic scale, is the ordinate. The saturated liquid line separates the subcooled liquid from the two-phase region in which vapor and liquid refrigerants coexist. The saturated vapor line separates this two-phase region from the superheated vapor. In the two-phase region, the mixture of vapor and liquid is subdivided by the constant-dryness-fraction quality line.

The constant-temperature lines are nearly vertical in the subcooled liquid region. At higher temperatures, they are curves near the saturated liquid line. In the two-phase region, the constant-temperature lines are horizontal. In the superheated region, the constant-temperature lines curve down sharply. Because the constant-temperature lines and constant-pressure lines in the two-phase region are horizontal, they are closely related. The specific pressure of a refrigerant in the two-phase region determines its temperature, and vice versa.

Also in the superheated region, the constant-entropy lines incline sharply upward, and constant-volume lines are flatter. Both are slightly curved.

Temperature-Entropy Diagram

The temperature-entropy *T-s* diagram is often used to analyze the irreversibilities in a refrigeration cycle, as well as in the system, in order to select optimum operating parameters and improve performance of the system. In a temperature-entropy *T-s* diagram, entropy s, Btu/lb · °R, is the abscissa of

the diagram and temperature T, °R, is the ordinate. A T-s diagram is more suitable for evaluating the effectiveness of an air expansion refrigeration cycle.

Analytical Evaluation of Cycle Performance

Swers et al. (1972) proposed a thermodynamic analysis of degradation of available energy and irreversibility in a refrigerating system, and Tan and Yin (1986) recommended a method of exergy analysis. The *exergy* of a working substance e, Btu/lb (kJ/kg), is defined as

$$e = h - h_a - T_{Ra}(s - s_a) \tag{9.1}$$

where h, h_a = enthalpy of working substance and ambient state, Btu/lb (kJ/kg)
T_{Ra} = absolute temperature of ambient state, °R (K)
s, s_a = entropy of working substance and ambient state, Btu/lb · °R (kJ/kg · K)

Both analyses are effective tools in the selection of optimum design and operating parameters by means of complicated analysis. They require extensive supporting data and information.

For most analyses of refrigeration cycle performance and design and operation of refrigeration systems in actual applications, satisfactory results can be obtained by using the steady flow energy equation, heat and work transfer, and energy balance principle. If a more precise and elaborate analysis is needed in research or for detailed improvements of refrigeration systems, the references at the end of this chapter can be consulted.

9.8 CARNOT REFRIGERATION CYCLE

The Carnot refrigeration cycle is a reverse engine cycle. All processes in a Carnot refrigeration cycle are reversible, so it is the most efficient refrigeration cycle.

Figure 9.4*a* is a schematic diagram of a Carnot cycle refrigerating system, and Fig. 9.4*b* shows the Carnot refrigeration cycle using gas as the working substance. This Carnot cycle is composed of four reversible processes:

1. An isothermal process 4-1 in which heat $q_{\#1}$ is extracted at constant temperature T_{R1} per lb (kg) of working substance
2. An isentropic compression process 1-2
3. An isothermal process 2-3 in which $q_{\#2}$ is rejected at constant temperature T_{R2} per lb (kg) of working substance
4. An isentropic expansion process 3-4

Figure 9.4*c* shows the Carnot refrigeration cycle using vapor as the working substance. Wet vapor is the only working substance where heat supply and heat rejection processes can occur easily at constant temperature. This is because the temperatures of wet vapor remain constant when latent heat is supplied or rejected.

As in the gas cycle, there are two isothermal processes 4-1 and 2-3 absorbing heat at temperature T_{R1} and rejecting heat at T_{R2}, respectively, and two isentropic processes, one for compression 1-2 and another for expansion 3-4.

Performance of Carnot Refrigeration Cycle

According to the first law of thermodynamics, often called the law of conservation of energy, when a system undergoes a thermodynamic cycle, the net heat supplied to the system is equal to the net

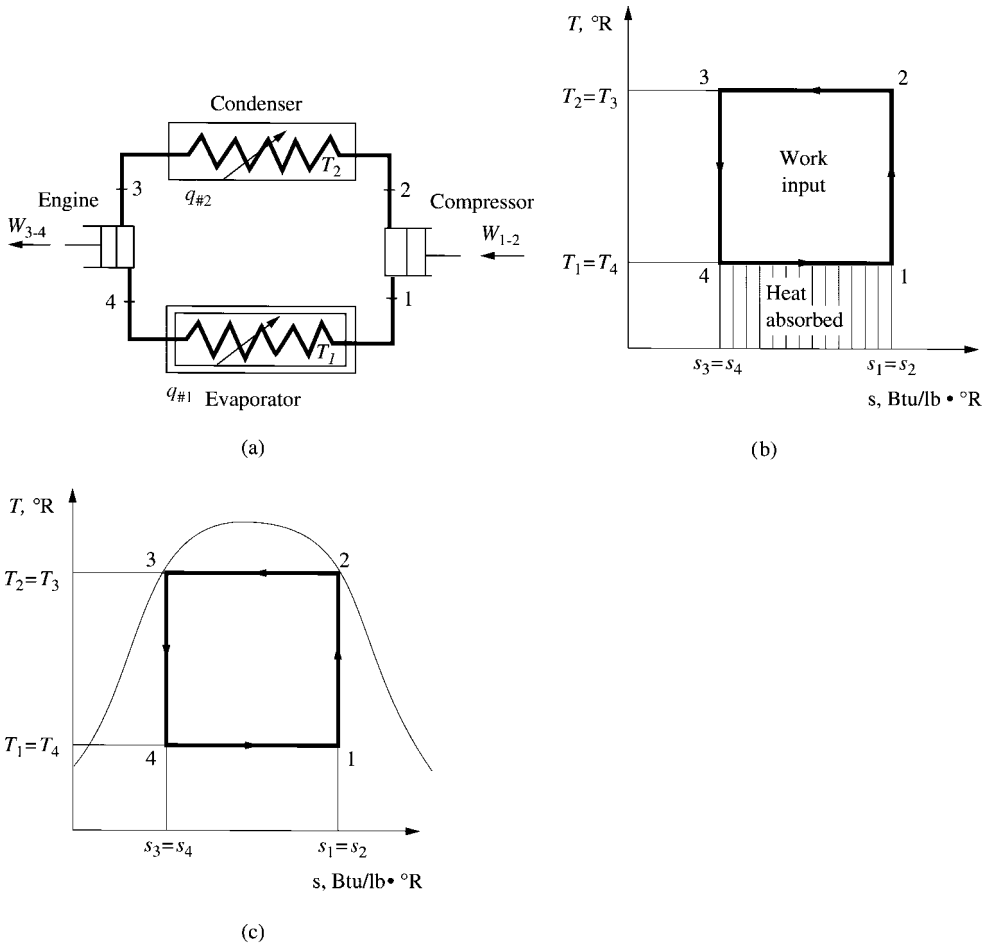


FIGURE 9.4 Carnot refrigeration cycle: (*a*) schematic diagram; (*b*) gas cycle; (*c*) vapor cycle.

work done, or

$$\text{Heat supply} + \text{heat rejected} = \text{net work done} \tag{9.2}$$

Referring to Fig. 9.4*a*, in a Carnot refrigeration cycle,

$$q_{\#1} - q_{\#2} = -W$$

or

$$q_{\#1} = q_{\#2} - W \tag{9.3}$$

$$q_{\#2} = q_{\#1} + W$$

where $q_{\#1}$ = heat supplied from surroundings per lb (kg) of working substance at temperature T_1; sign of $q_{\#1}$ is positive

$q_{\#2}$ = heat rejected to sink per lb (kg) of working substance at temperature T_2; sign of $q_{\#2}$ is negative

W = net work done by system; sign is positive, or if it is a work input to system, sign is negative

The heat extracted from the source at temperature T_{R1} by the working substance, i.e., the refrigerating effect per lb (kg) of working substance, is

$$q_{\#1} = T_{R1}(s_1 - s_4) \tag{9.4}$$

where s_1, s_4 = entropy at state points 1 and 4, respectively, Btu/lb · °R (kJ/kg · K). Heat rejected to the heat sink at temperature T_{R2} can be calculated as

$$q_{\#2} = -T_{R2}(s_3 - s_2) = T_{R2}(s_2 - s_3) \tag{9.5}$$

where s_2, s_3 = entropy at state points 2 and 3, respectively, Btu/lb · °R (kJ/kg · K). Because in the isentropic process 1-2, $s_1 = s_2$, and in isentropic process 3-4, $s_3 = s_4$,

$$q_{\#2} = T_{R2}(s_1 - s_4) \tag{9.6}$$

9.9 COEFFICIENT OF PERFORMANCE OF REFRIGERATION CYCLE

The coefficient of performance is an index of performance of a thermodynamic cycle or a thermal system. Because the COP can be greater than 1, COP is used instead of thermal efficiency. The coefficient of performance can be used for the analysis of the following:

- A refrigerator that is used to produce a refrigeration effect only, that is, COP_{ref}
- A heat pump in which the heating effect is produced by rejected heat COP_{hp}
- A heat recovery system in which both the refrigeration effect and the heating effect are used at the same time, COP_{hr}

For a refrigerator, COP is defined as the ratio of the refrigeration effect $q_{\#1}$ to the work input W_{in}, both in Btu/lb (kJ/kg), that is,

$$\begin{aligned}\text{COP}_{\text{ref}} &= \text{Refrigeration effect/Work input} \\ &= q_{\#1}/W_{\text{in}}\end{aligned} \tag{9.7}$$

For the Carnot refrigeration cycle, from Eq. (9.3),

$$\begin{aligned}\text{COP}_{\text{ref}} &= \frac{q_{\#1}}{q_{\#2} - q_{\#1}} \\ &= \frac{T_{R1}(s_1 - s_4)}{(T_{R2} - T_{R1})(s_1 - s_4)} = \frac{T_{R1}}{T_{R2} - T_{R1}}\end{aligned} \tag{9.8}$$

With a heat pump, the useful effect is the heating effect because of the rejected heat $q_{\#2}$, so COP_{hp} is the ratio of heat rejection to the work input, or

$$\text{COP}_{\text{hp}} = \frac{q_{\#2}}{W_{\text{in}}} \tag{9.9}$$

For a heat recovery system, the useful effect is $q_{\#1}$ and $q_{\#2}$. In such a condition, COP_{hr} is defined as the ratio of the sum of the absolute values of refrigerating effect and heat rejection to the absolute value of the work input, i.e.,

$$\text{COP}_{\text{hr}} = \frac{|q_{\#1}| + |q_{\#2}|}{W_{\text{in}}} \tag{9.10}$$

9.10 SINGLE-STAGE IDEAL VAPOR COMPRESSION CYCLE

The Carnot cycle cannot be achieved for the vapor cycle in actual practice because liquid slugging would occur during compression of the two-phase refrigerant. In addition, the mixture, mostly liquid, does very little work when it expands after condensation in the heat engine. Therefore, a single-stage ideal vapor compression cycle is used instead of the Carnot cycle.

Figure 9.5 shows an ideal single-stage vapor compression cycle in which compression occurs in the superheated region. A throttling device, such as an expansion valve, is used instead of the heat engine. Single-stage means that there is only one stage of compression. An ideal cycle is one in which the compression process is isentropic and the pressure losses in the pipeline, valves, and other components are negligible. All the refrigeration cycles covered in this chapter are ideal cycles except the air expansion refrigeration cycle.

Vapor compression means that the vapor refrigerant is compressed to a higher pressure, and then the condensed liquid is throttled to a lower pressure to produce the refrigerating effect by evaporation. It is different from the absorption or air expansion refrigeration cycle.

Flow Processes

Figure 9.5*b* and *c* shows the refrigeration cycle on *p-h* and *T-s* diagrams. The refrigerant evaporates entirely in the evaporator and produces the refrigerating effect. It is then extracted by the compressor at state point 1, compressor suction, and is compressed isentropically from state point 1 to 2. It is next condensed to liquid in the condenser, and the latent heat of condensation is rejected to the heat sink. The liquid refrigerant, at state point 3, flows through an expansion valve, which reduces it to the evaporating pressure. In the ideal vapor compression cycle, the throttling process at the expansion valve is the only irreversible process, usually indicated by a dotted line. Some of the liquid flashes into vapor and enters the evaporator at state point 4. The remaining liquid portion evaporates at the evaporating temperature, thus completing the cycle.

Cycle Performance

For the evaporating process between points 4 and 1, according to the steady flow energy equation,

$$\frac{h_4 + v_4^2}{2g_c \times 778} + q_\# = \frac{h_1 + v_1^2}{2g_c \times 778} + W \tag{9.11}$$

where h_1, h_4 = enthalpy of refrigerant at points 1 and 4, respectively, Btu/lb (J/kg)
v_1, v_4 = velocity of refrigerant at points 1 and 4, respectively, ft/s (m/s)
$q_\#$ = heat supplied per lb (kg) of working substance during evaporation process, Btu/lb (J/kg)
g_c = dimensional conversion factor, 32 $\text{lb}_\text{m} \cdot \text{ft}/\text{lb}_\text{f} \cdot \text{s}^2$

Because no work is done during evaporation, the change of kinetic energy between points 4 and 1 is small compared with other terms in Eq. (9.11), and it is usually ignored. Then

$$h_4 + q_\# = h_1 + 0$$

The refrigerating effect q_{rf}, Btu/lb (J/kg), is

$$q_{rf} = q_\# = h_1 - h_4 \tag{9.12}$$

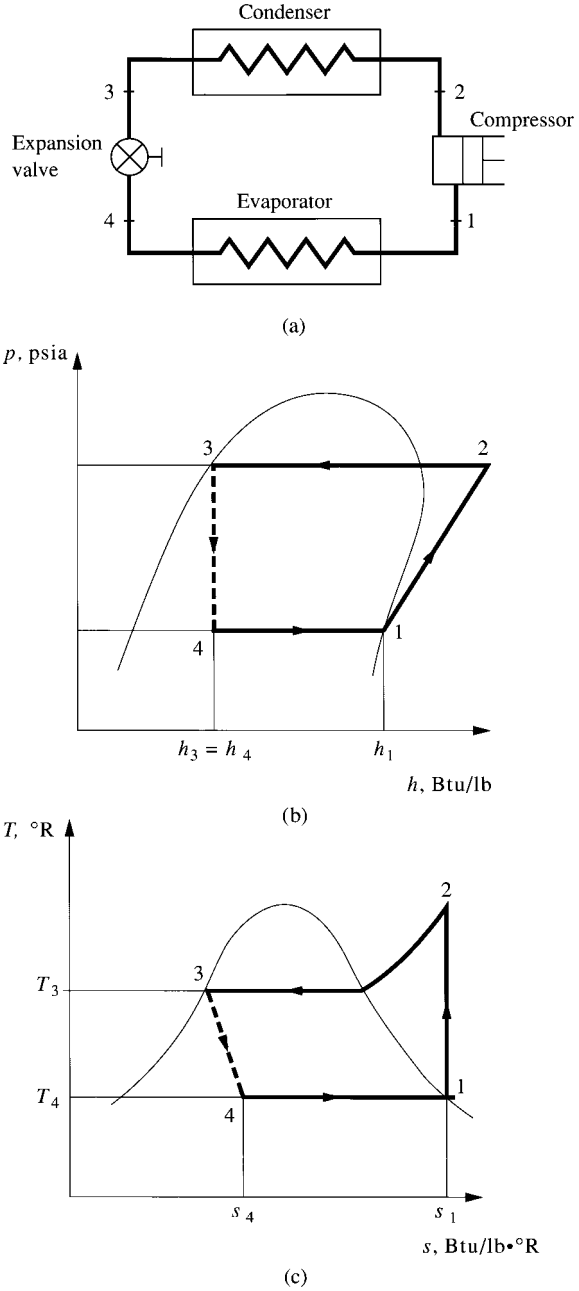


FIGURE 9.5 Single-stage ideal vapor compression cycle: (*a*) schematic diagram; (*b*) *p-h* diagram; (*c*) *T-s* diagram.

For isentropic compression between points 1 and 2, applying the steady flow energy equation and ignoring the change of kinetic energy, we have

$$h_1 + 0 = h_2 + W$$
$$-W = h_2 - h_1$$

Work input to the compressor W_{in}, Btu/lb (kJ/kg), is given as

$$W_{in} = h_2 - h_1 \tag{9.13}$$

Similarly, for condensation between points 2 and 3,

$$h_2 + q_{\#} = h_3 + 0$$

The heat released by the refrigerant in the condenser $-q_{\#}$, Btu/lb (kJ/kg), is

$$-q_{\#} = h_2 - h_3 \tag{9.14}$$

For the throttling process between points 3 and 4, assuming that the heat loss is negligible,

$$h_3 + 0 = h_4 + 0$$
$$\text{or } h_3 = h_4 \tag{9.15}$$

The COP_{ref} of the single-stage ideal vapor compression cycle is

$$\text{COP}_{ref} = \frac{\text{refrigerating effect}}{\text{work input}} = \frac{q_{rf}}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{9.16}$$

The mass flow rate of refrigerant $\dot{m}_r$, lb/h (kg/s), flowing through the evaporator is

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} \tag{9.17}$$

where Q_{rc} = refrigerating capacity, Btu/h (W). From Eq. (9.16), the smaller the difference between the condensing and evaporating pressures, or between condensing and evaporating temperatures, the lower the W_{in} input to the compressor at a specific Q_{rc} and, therefore, the higher the COP. A higher evaporating pressure p_{ev} and evaporating temperature T_{ev} or a lower condensing pressure p_{con} and condensing temperature T_{con} will always conserve energy.

Determination of Enthalpy by Polynomials

During the performance analysis of a refrigeration cycle, the enthalpies h of the refrigerant at various points must be determined in order to calculate the refrigeration effect, work input, and COP. The enthalpy of refrigerant at saturated liquid and saturated vapor state is a function of saturated temperature or pressure. In other words, saturated temperature T_s and saturated pressure p_{ss} of the refrigerant are dependent upon each other. Therefore, it is more convenient to evaluate the enthalpy of refrigerant in terms of saturated temperature T_s within a certain temperature range

$$h = f(T_s) \tag{9.18}$$

The enthalpy differential along the constant-entropy line within a narrower temperature range can be calculated as

$$h_2 - h_1 = F(T_{s2} - T_{s1}) \tag{9.19}$$

where h_1, h_2 = enthalpy of refrigerant on constant-entropy line at points 1 and 2, Btu/lb (kJ/kg)
T_{s1}, T_{s2} = temperature of saturated refrigerant at points 1and 2,°F (°C)

From the refrigerant tables published by ASHRAE, the following polynomial can be used to calculate the enthalpy of saturated liquid refrigerant h_{lr}, Btu/lb (kJ/kg), from its temperature T_{s1} at a saturated temperature from 20 to 120°F (− 7 to 50°C) with acceptable accuracy:

$$h_{lr} = a_1 + a_2 T_{s1} + a_3 T_{s1}^2 + a_4 T_{s1}^3 \tag{9.20}$$

where a_1, a_2, a_3, a_4 = coefficients. For HCFC-22,

$$a_1 = 10.409 \qquad a_2 = 0.26851 \qquad a_3 = 0.00014794 \qquad a_4 = 5.3429 \times 10^{-7}$$

Similarly, the polynomial that determines the enthalpy of saturated vapor refrigerant h_{vr}, Btu/lb (kJ/kg), from its temperature T_{sv} in the same temperature range is

$$h_{vr} = b_1 + b_2 T_{sv} + b_3 T_{sv}^2 + b_4 T_{sv}^3 \tag{9.21}$$

where b_1, b_2, b_3, b_4 = coefficients. For HCFC-22,

$$b_1 = 104.465 \qquad b_2 = 0.098445 \qquad b_3 = -0.0001226 \qquad b_4 = -9.861 \times 10^{-7}$$

The polynomial that determines the enthalpy changes of refrigerant along the constant-entropy line for an isentropic compression process between initial state 1 and final state 2 is

$$h_2 - h_1 = c_1 + c_2(T_{s2} - T_{s1}) + c_3(T_{s2} - T_{s1})^2 + c_4(T_{s2} - T_{s1})^3 \tag{9.22}$$

where c_1, c_2, c_3, c_4 = coefficients
T_{s1}, T_{s2} = saturated temperature of vapor refrigerant corresponding to its pressure at initial state 1 and final state 2,°F (°C)

For HCFC-22 within a saturated temperature range of 20 to 100°F:

$$c_1 = -0.18165 \qquad c_2 = +0.21502 \qquad c_3 = -0.0012405 \qquad c_4 = +8.198 \times 10^{-6}$$

Computer programs are available that calculate the coefficients based on ASHRAE's refrigerant tables and charts.

Refrigeration Effect, Refrigerating Load, and Refrigerating Capacity

The refrigeration effect q_{rf}, Btu/lb (J/kg or kJ/kg), is the heat extracted by a unit mass of refrigerant during the evaporating process in the evaporator. It can be calculated as

$$q_{rf} = h_{\text{lv}} - h_{\text{en}} \tag{9.23}$$

where h_{en}, h_{lv} = enthalpy of refrigerant entering and leaving evaporator, Btu/lb (J/kg). Refrigerating load Q_{rl}, Btu/h (W), is the required rate of heat extraction by the refrigerant in the evaporator. It

can be calculated as

$$Q_{rl} = \dot{m}_r(h_{\text{lv}} - h_{\text{en}}) \tag{9.24}$$

where $\dot{m}_r$ = mass flow rate of refrigerant flowing through evaporator, lb/h (kg/s).

Refrigerating capacity, or cooling capacity, Q_{rc}, Btu/h (W), is the actual rate of heat extracted by the refrigerant in the evaporator. In practice, the refrigeration capacity of the equipment selected is often slightly higher than the refrigerating load. This is because the manufacturer's specifications are a series of fixed capacities. Occasionally, equipment can be selected so that its capacity is just equal to the refrigeration load required. Refrigeration capacity Q_{rc} can be calculated as

$$Q_{rc} = \dot{m}_r(h_{r\text{lv}} - h_{r\text{en}}) \tag{9.25}$$

where $h_{r\text{en}}$, $h_{r\text{lv}}$ = enthalpy of refrigerant actually entering and leaving evaporator, Btu/lb (J/kg)

9.11 SUBCOOLING AND SUPERHEATING

Subcooling

Condensed liquid refrigerant is usually subcooled to a temperature lower than the saturated temperature corresponding to the condensing pressure of the refrigerant, shown in Fig. 9.6*a* as point 3′. This is done to increase the refrigerating effect. The degree of subcooling depends mainly on the temperature of the coolant (e.g., atmospheric air, surface water, or well water) during condensation, and the construction and capacity of the condenser.

The enthalpy of subcooled liquid refrigerant h_{sc}, Btu/lb (J/kg), can be calculated as

$$h_{sc} = h_{s,\text{con}} - c_{\text{pr}}(T_{s,\text{con}} - T_{sc}) \tag{9.26}$$

where $h_{s,\text{con}}$ = enthalpy of saturated liquid refrigerant at condensing temperature, Btu/lb (J/kg)
c_{pr} = specific heat of liquid refrigerant at constant pressure, Btu/lb · °F (J/kg · °C)
$T_{s,\text{con}}$ = saturated temperature of liquid refrigerant at condensing pressure,°F (°C)
T_{sc} = temperature of subcooled liquid refrigerant,°F (°C)

Enthalpy h_{sc} is also approximately equal to the enthalpy of the saturated liquid refrigerant at subcooled temperature.

Superheating

As mentioned before, the purpose of superheating is to avoid compressor slugging damage. Superheating is shown in Fig. 9.6*b*. The degree of superheat depends mainly on the type of refrigerant feed and compressor as well as the construction of the evaporator. These are covered in detail in Chap. 11.

Example 9.1. A 500-ton (1760-kW) single-stage centrifugal vapor compression system uses HCFC-22 as refrigerant. The vapor refrigerant enters the compressor at dry saturated state. The compression process is assumed to be isentropic. Hot gas is discharged to the condenser and condensed at a temperature of 95°F (35°C). The saturated liquid refrigerant then flows through a throttling device and evaporates at a temperature of 35°F (1.7°C). Calculate:

1. The refrigeration effect
2. The work input to the compressor

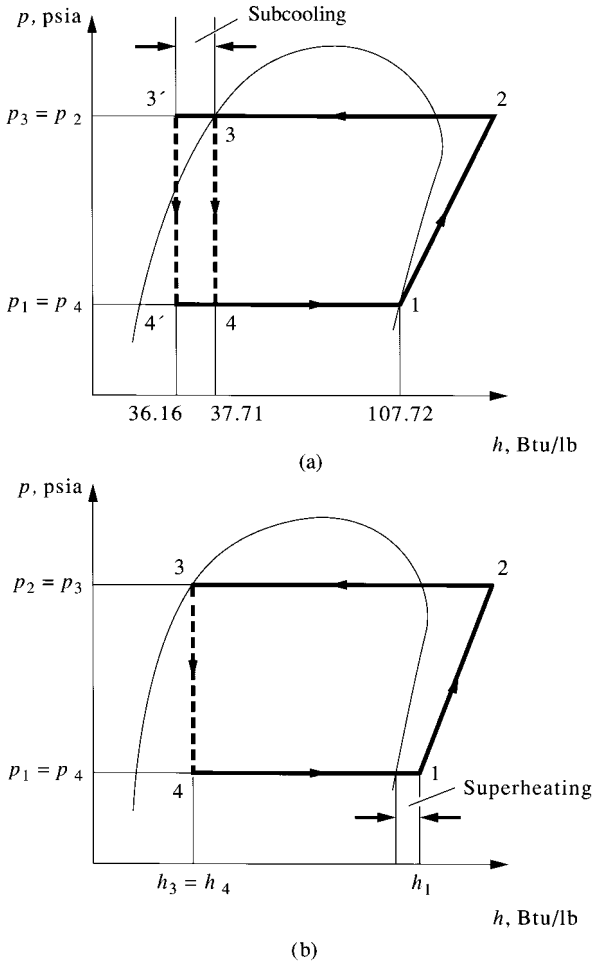


FIGURE 9.6 (*a*) Subcooling and (*b*) superheating.

3. The coefficient of performance of this refrigeration cycle

4. The mass flow rate of the refrigerant

Recalculate the COP and the energy saved in work input if the refrigerant is subcooled to a temperature of 90°F (32.2°C).

Solution

1. From Eq. (9.20), the enthalpy of the saturated liquid refrigerant at a temperature of 95°F (35°C), point 3 as shown in Fig. 9.5*a*, is

$$h_3 = h_4 = 10.409 + 0.26851T_s + 0.0001479T_s^2 + 5.3429 \times 10^{-7}T_s^3$$
$$= 10.409 + 0.26851(95) + 0.0001479(95)^2 + 5.3429 \times 10^{-7}(95)^3$$
$$= 10.409 + 25.508 + 1.335 + 0.458 = 37.71 \text{ Btu/lb}$$

From Eq. (9.21), the enthalpy of saturated vapor refrigerant at a temperature of 35°F (1.7°C), point 1, is

$$\begin{aligned} h_1 &= 104.465 + 0.098445T_s - 0.0001226T_s^2 - 9.861 \times 10^{-7}T_s^3 \\ &= 104.47 + 0.098445(35) - 0.0001226(35)^2 - 9.861 \times 10^{-7}(35)^3 \\ &= 104.47 + 3.445 - 0.150 - 0.042 = 107.72 \text{ Btu/lb} \end{aligned}$$

Then the refrigeration effect is calculated as

$$q_{rf} = h_1 - h_4 = 107.72 - 37.71 = 70.01 \text{ Btu/lb (162.8 kJ/kg)}$$

2. From Eq. (9.22), the enthalpy differential $h_2 - h_1$ on the constant-entropy line corresponding to a saturated temperature differential $T_{s2} - T_{s1} = 95 - 35 = 60$°F in the two-phase region is

$$\begin{aligned} h_2 - h_1 &= -0.18165 + 0.21502(T_{s2} - T_{s1}) - 0.0012405(T_{s2} - T_{s1})^2 \\ &\quad + 8.1982 \times 10^{-6}(T_{s2} - T_{s1})^3 \\ &= -0.18165 + 0.21502(60) - 0.0012405(60)^2 + 8.1982 \times 10^{-6}(60)^3 \\ &= -0.182 + 12.901 - 4.466 + 1.771 = 10.02 \text{ Btu/lb} \end{aligned}$$

That is, the work input $W_{\text{in}} = h_2 - h_1 = 10.024$ Btu/lb (23.73 kJ/kg).

3. According to Eq. (9.16), COP_{ref} of the refrigerating system is calculated as

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{\text{in}}} = \frac{70.01}{10.024} = 6.98$$

4. From Eq. (9.17), the mass flow rate of the refrigerant can be calculated as

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{500 \times 12{,}000}{70.01} = 85{,}702 \text{ lb/h (38,874 kg/h)}$$

If the liquid refrigerant is subcooled to a temperature of 90°F, (32.2°C), then

$$\begin{aligned} h_3 = h_4 &= 10.409 + 0.26851(90) + 0.0001479(90)^2 + 5.3429 \times 10^{-7}(90)^3 \\ &= 10.409 + 24.166 + 1.198 + 0.389 = 36.16 \text{ Btu/lb} \end{aligned}$$

Refrigeration effect is then increased to

$$q_{rf} = 107.72 - 36.16 = 71.56 \text{ Btu/lb (166 kJ/kg)}$$

Also COP_{ref} is increased to

$$\text{COP}_{\text{ref}} = \frac{71.56}{10.024} = 7.14$$

Since 1 ton = 200 Btu/min and 1 hp = 42.41 Btu/min, electric power input to the compressor P_{in} without subcooling is

$$P_{\text{in}} = \frac{500 \times 200}{42.41 \times 6.98} = 337.8 \text{ hp (252 kW)}$$

With subcooling,

$$P_{\text{in,s}} = \frac{500 \times 200}{42.41 \times 7.14} = 330.2 \text{ hp (246 kW)}$$

Savings in electric energy are calculated as

$$\frac{337.8 - 330.2}{337.8} = 0.022, \text{ or } 2.2\%$$

9.12 MULTISTAGE VAPOR COMPRESSION SYSTEMS

When a refrigeration system uses more than single-stage compression process, it is called a multistage system (as shown in Fig. 9.7), and may include the following:

1. A high-stage compressor and a low-stage compressor
2. Several compressors connected in series
3. Two or more impellers connected internally in series and driven by the same motor or prime mover, as shown in Fig. 9.7
4. A combination of two separate refrigeration systems

The discharge pressure of the low-stage compressor, which is equal to the suction pressure of the high-stage compressor, is called the interstage pressure.

The reasons for using a multistage vapor compression system instead of a single-stage system are as follows:

1. The compression ratio R_{com} of each stage in a multistage system is smaller than that in a single-stage unit, so compressor efficiency is increased. Compression ratio R_{com} is defined as the ratio of the compressor's discharge pressure p_{dis}, psia (kPa abs.), to the suction pressure at the compressor's inlet p_{suc}, psia (kPa abs.), or

$$R_{\text{com}} = \frac{p_{\text{dis}}}{p_{\text{suc}}} \tag{9.27}$$

2. Liquid refrigerant enters the evaporator at a lower enthalpy and increases the refrigeration effect.
3. Discharge gas from the low-stage compressor can be desuperheated at the interstage pressure. This results in a lower discharge temperature from the high-stage compressor than would be produced by a single-stage system at the same pressure differential between condensing and evaporating pressures.
4. Two or three compressors in a multistage system provide much greater flexibility to accommodate the variation of refrigeration loads at various evaporating temperatures during part-load operation.

The drawbacks of the multistage system are higher initial cost and a more complicated system than that for a single-stage system.

Compound Systems

Multistage vapor compression systems are classified as compound systems or cascade systems. Cascade systems are discussed in a later section.

A compound system consists of two or more compression stages connected in series. For reciprocating, scroll, or screw compressors, each compression stage usually requires a separate

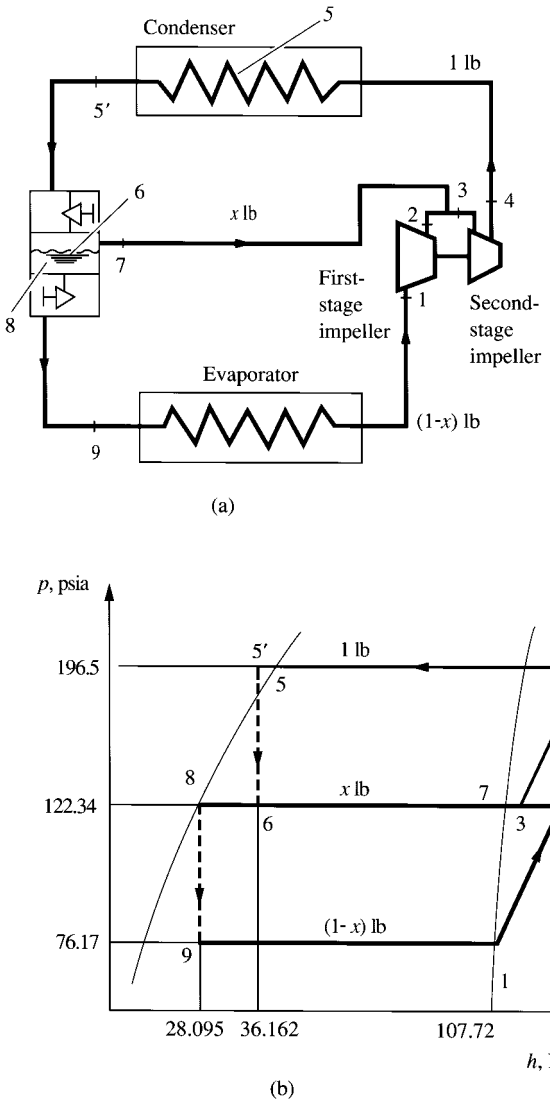


FIGURE 9.7 Two-stage compound system with a flash cooler: (*a*) schematic diagram; (*b*) refrigeration cycle.

compressor. In multistage centrifugal compressors, two or more stages may be internally compounded by means of several impellers connected in series.

Interstage Pressure

Interstage pressure is usually set so that the compression ratio at each stage is nearly the same for higher COPs. For a two-stage compound system, interstage pressure p_i, psia (kPa abs.), can be

calculated as

$$p_i = \sqrt{p_{\text{con}} p_{\text{ev}}} \tag{9.28}$$

where p_{con} = condensing pressure, psia (kPa abs.)
p_{ev} = evaporating pressure, psia (kPa abs.)

For a multistage vapor compression system with z stages, the compression ratio R_{com} for each stage can be calculated as

$$R_{\text{com}} = \left(\frac{p_{\text{con}}}{p_{\text{ev}}} \right)^{1/z} \tag{9.29}$$

Flash Cooler and Intercooler

In compound systems, flash coolers are used to subcool liquid refrigerant to the saturated temperature corresponding to the interstage pressure by vaporizing part of the liquid refrigerant. Intercoolers are used to desuperheat the discharge gas from the low-stage compressor and, more often, to subcool also the liquid refrigerant before it enters the evaporator.

9.13 TWO-STAGE COMPOUND SYSTEM WITH A FLASH COOLER

Flow Processes

Figure 9.7*a* is a schematic diagram of a two-stage compound system with a flash cooler, and Fig. 9.7*b* shows the refrigeration cycle of this system. Vapor refrigerant at point 1 enters the first-stage impeller of the centrifugal compressor at the dry saturated state. It is compressed to the interstage pressure p_i at point 2 and mixes with evaporated vapor refrigerant from the flash cooler, often called an economizer. The mixture then enters the second-stage impeller at point 3. Hot gas, compressed to condensing pressure p_{con}, leaves the second-stage impeller at point 4. It is then discharged to the condenser, in which the hot gas is desuperheated, condensed, and often subcooled to liquid state at point 5′. After the condensing process, the subcooled liquid refrigerant flows through a throttling device, such as a float valve, at the high-pressure side. A small portion of the liquid refrigerant flashes into vapor in the flash cooler at point 7, and this latent heat of vaporization cools the remaining liquid refrigerant to the saturation temperature corresponding to the interstage pressure at point 8. Inside the flash cooler, the mixture of vapor and liquid refrigerant is at point 6. Liquid refrigerant then flows through another throttling device, a small portion is flashed at point 9, and the liquid-vapor mixture enters the evaporator. The remaining liquid refrigerant is vaporized at point 1 in the evaporator. The vapor then flows to the inlet of the first-stage impeller of the centrifugal compressor and completes the cycle.

Fraction of Evaporated Refrigerant in Flash Cooler

In the flash cooler, out of 1 lb of refrigerant flowing through the condenser, x lb of it cools down the remaining portion of liquid refrigerant $(1 - x)$ lb to saturated temperature T_8 at interstage pressure p_i. Because h_5' is the enthalpy of the subcooled liquid refrigerant entering the flash cooler, h_6 is the enthalpy of the mixture of vapor and liquid refrigerant after the throttling device, for a throttling process, $h_5' = h_6$. Enthalpies h_7 and h_8 are the enthalpies of the saturated vapor and saturated liquid, respectively, at the interstage pressure, and h_9 is the enthalpy of the mixture of vapor and liquid refrigerant leaving the flash cooler after the low-pressure side throttling device. Again, for a throttling process, $h_8 = h_9$.

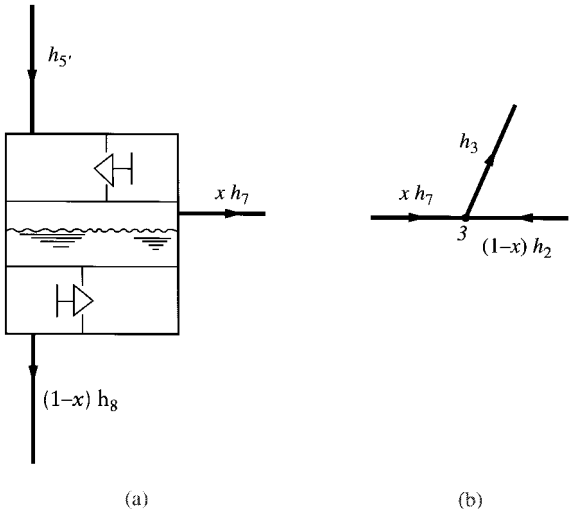


FIGURE 9.8 Heat balance of entering and leaving refrigerants in a flash cooler and at the mixing point: (*a*) in the flash cooler; (*b*) at the mixing point 3 before entering the second-stage impeller.

If the heat loss from the insulated flash cooler to the ambient air is small, it can be ignored. Heat balance of the refrigerants entering and leaving the flash cooler, as shown in Fig. 9.8*a*, gives

Sum of heat energy of refrigerant entering flash cooler

= Sum of heat energy of refrigerant leaving flash cooler

that is,

$$h_{5'} = xh_7 + (1 - x)h_8$$

The fraction of liquid refrigerant evaporated in the flash cooler x is given as

$$x = \frac{h_{5'} - h_8}{h_7 - h_8} \tag{9.30}$$

The fraction x also indicates the quality, or dryness fraction, of the vapor and liquid mixture in the flash cooler at the interstage pressure.

Enthalpy of Vapor Mixture Entering Second-Stage Impeller

Ignoring the heat loss from mixing point 3 to the surroundings, we see that the mixing of the gaseous refrigerant discharged from the first-stage impeller at point 2 and the vaporized refrigerant from the flash cooler at point 7 is an adiabatic process. The heat balance at the mixing point before the second-stage impeller, as shown in Fig. 9.7*b*, is given as

$$h_3 = (1 - x)h_2 + xh_7 \tag{9.31}$$

where h_2 = enthalpy of gaseous refrigerant discharged from first-stage impeller, Btu/lb (kJ/kg)
h_3 = enthalpy of mixture at point 3, Btu/lb (kJ/kg)
h_7 = enthalpy of saturated vapor refrigerant from flash cooler at point 7, Btu/lb (kJ/kg)

Coefficient of Performance

For 1 lb (kg) of refrigerant flowing through the condenser, the amount of refrigerant flowing through the evaporator is $(1 - x)$ lb (kg). The refrigeration effect q_{rf} per lb (kg) of refrigerant flowing through the condenser, Btu/lb, (kJ/kg), can be expressed as

$$q_{rf} = (1 - x)(h_1 - h_9) \tag{9.32}$$

where h_1 = enthalpy of saturated vapor leaving evaporator, Btu/lb (kJ/kg)
h_9 = enthalpy of refrigerant entering evaporator, Btu/lb (kJ/kg)

Total work input to the compressor (including the first- and second-stage impeller) W_{in} per lb (kg) of refrigerant flowing through the condenser, Btu/lb (kJ/kg), is

$$W_{\text{in}} = (1 - x)(h_2 - h_1) + h_4 - h_3 \tag{9.33}$$

where h_4 = enthalpy of the hot gas discharged from the second-stage impeller, Btu/lb (kJ/kg). The coefficient of performance of the two-stage compound system with a flash cooler COP_{ref} is

$$\begin{aligned} \text{COP}_{\text{ref}} &= \frac{q_{rf}}{W_{\text{in}}} \\ &= \frac{(1 - x)(h_1 - h_9)}{(1 - x)(h_2 - h_1) + (h_4 - h_3)} \end{aligned} \tag{9.34}$$

The mass flow rate of refrigerant at the condenser $\dot{m}_r$, lb/h (kg/s), is

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{Q_{rc}}{(1 - x)(h_1 - h_9)} \tag{9.35}$$

where Q_{rc} = refrigeration capacity, Btu/h (W).

Characteristics of Two-Stage Compound System with Flash Cooler

In a two-stage compound system with a flash cooler, a portion of the liquid refrigerant is flashed into vapor going directly to the second-stage suction inlet, so less refrigerant is compressed in the first-stage impeller. Furthermore, the remaining liquid refrigerant is cooled to the saturated temperature corresponding to the interstage pressure, which is far lower than the subcooled liquid temperature in a single-stage system. The increase in refrigeration effect and the drop in compression work input lead to a higher COP_{ref} than in a single-stage system.

Although the initial cost of a two-stage compound system is higher than that for a single-stage system, the two-stage compound system with a flash cooler is widely used in large central hydronic air conditioning systems because of the high COP_{ref}.

Example 9.2. For the same 500-ton (1758-kW) centrifugal vapor compression system as in Example 9.1, a two-stage compound system with a flash cooler is used instead of a single-stage centrifugal compressor. Vapor refrigerant enters the first-stage impeller at a dry saturated state, and the subcooled liquid refrigerant leaves the condenser at a temperature of 90°F (32.2°C). Both compression processes at the first-stage impeller and the second-stage impeller are assumed to be isentropic. Evaporating pressure is 76.17 psia (525kPa abs.), and the condensing pressure is 196.5 psia (1355 kPa abs.). Other conditions remain the same as in Example 9.1. Calculate

1. The fraction of liquid refrigerant vaporized in the flash cooler
2. The refrigeration effect per lb (kg) of refrigerant flowing through the condenser
3. The total work input to the compressor

4. The coefficient of performance of this refrigerating system
5. The mass flow rate of refrigerant flowing through the condenser
6. The percentage of saving in energy consumption compared with the single-stage vapor compression system

Solution

1. Based on the data calculated in Example 9.1, enthalpy of vapor refrigerant entering the first-stage impeller $h_1 = 107.72$ Btu/lb and the enthalpy of the subcooled liquid refrigerant leaving the condenser $h_{5'} = 36.162$ Btu/lb, as shown in Fig. 9.6*b*. From Eq. (9.28) and the given data, the interstage pressure can be calculated as

$$p_i = \sqrt{p_{\text{con}} p_{\text{ev}}} = \sqrt{196.5 \times 76.17} = 122.34 \text{ psia}$$

From the Table of Properties of Saturated Liquid and Vapor for HCFC-22 in *ASHRAE Handbook 1989*, *Fundamentals*, for $p_i = 122.34$ psia, the corresponding interstage saturated temperature T_i in the two-phase region is 63.17°F.

From Eq. (9.20), the enthalpy of liquid refrigerant at saturated temperature 63.17°F is

$$h_8 = h_9 = 10.409 + 0.26851(63.17) + 0.00014794(63.17)^2 + 5.3429(63.17)^3$$

$$= 10.409 + 16.961 + 0.59 + 0.135 = 28.095 \text{ Btu/lb}$$

Also, from Eq. (9.21), the enthalpy of the saturated vapor refrigerant at a temperature of 63.17°F is

$$h_7 = 104.465 + 0.098445(63.17) - 0.0001226(63.17)^2 - 9.861 \times 10^{-7}(63.17)^3$$

$$= 104.465 + 6.219 - 0.489 - 0.249 = 109.946 \text{ Btu/lb}$$

Then, from Eq. (9.30), the fraction of vaporized refrigerant in the flash cooler is

$$x = \frac{h_{5'} - h_8}{h_7 - h_8}$$

$$= \frac{36.162 - 28.095}{109.946 - 28.095} = 0.09856$$

2. From Eq. (9.32), the refrigeration effect is

$$q_{rf} = (1 - x)(h_1 - h_9) = (1 - 0.09856)(107.72 - 28.095)$$

$$= 71.78 \text{ Btu/lb } (167 \text{ kJ/kg})$$

3. From Eq. (9.22), the enthalpy differential $h_2 - h_1$ corresponding to a saturated temperature differential $T_{s2} - T_{s1} = 63.17 - 35 = 28.17$°F in the two-phase region is

$$h_2 - h_1 = -0.18165 + 0.21502(28.17) - 0.0012405(28.17)^2 + 8.1982 \times 10^{-6}(28.17)^3$$

$$= -0.182 + 6.057 - 0.984 + 0.183 = 5.074 \text{ Btu/lb}$$

Similarly, from Eq. (9.22), the enthalpy differential $h_4 - h_3$ corresponding to a saturated temperature differential $T_s - T_i = 95 - 63.17 = 31.83$°F is

$$h_4 - h_3 = -0.020 + 0.16352(31.83) - 0.00035106(31.83)^2 + 1.9177 \times 10^{-6}(31.83)^3$$

$$= -0.020 + 5.205 - 0.356 + 0.062 = 4.891 \text{ Btu/lb}$$

Then, from Eq. (9.33), the total work input to the compressor is calculated as

$$W_{\text{in}} = (1 - x)(h_2 - h_1) + h_4 - h_3$$
$$= (1 - 0.09856)(5.074) + 4.891 = 9.465 \text{ Btu/lb } (22.0 \text{ kJ/kg})$$

4. The coefficient of performance of this two-stage compound system is

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{\text{in}}} = \frac{71.78}{9.465} = 7.58$$

5. The mass flow rate of refrigerant flowing through the condenser can be evaluated as

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{500 \times 12{,}000}{71.78} = 83{,}588 \text{ lb/h } (37{,}916 \text{ kg/h})$$

6. From the results in Example 9.1, the power input to the single-stage system is 330.2 hp. The power input to the two-stage compound system is

$$P_{\text{in}} = \frac{500 \times 200}{42.41 \times 7.58} = 311.1 \text{ hp } (232 \text{ kW})$$

Energy saving compared with the single-stage system is calculated as

$$\frac{330.2 - 311.1}{330.2} = 0.058, \quad \text{or} \quad 5.8\%$$

9.14 THREE-STAGE COMPOUND SYSTEM WITH A TWO-STAGE FLASH COOLER

To reduce the energy consumption of refrigeration systems in air conditioning, the three-stage compound system with a two-stage flash cooler became a standard product in the 1980s. Fig. 9.9 shows the schematic diagram and refrigeration cycle of this system.

Flow Processes

In Fig. 9.9, vapor refrigerant enters the first-stage impeller of the centrifugal compressor at a dry saturated state, point 1. After the first-stage compression process, at point 2, it mixes with the vaporized refrigerant coming from the low-pressure flash cooler at point 12. At point 3, the mixture enters the second-stage impeller. After the second-stage compression process at point 4, it mixes again with vaporized refrigerant from the high-pressure flash cooler at point 9. The mixture, at point 5, is then compressed to the condensing pressure in the third-stage impeller. At point 6, the hot gas enters the condenser, condenses to liquid, and subcools to a temperature below the condensing temperature. Subcooled liquid refrigerant leaves the condenser at point 7′ and flows through a two-stage flash cooler and the associated throttling devices, in which a small portion of liquid refrigerant is successively flashed into vapor at interstage pressure p_{i1}, point 9, and interstage pressure p_{i2}, point 12. The liquid-vapor mixture enters the evaporator at point 14, and the remaining liquid refrigerant evaporates into vapor completely in the evaporator.

Fraction of Refrigerant Vaporized in Flash Cooler

Based on the heat balance of the refrigerants entering and leaving the high-pressure flash cooler, as shown in Fig. 9.10*a*, the fraction of liquid refrigerant vaporized in the high-pressure flash cooler

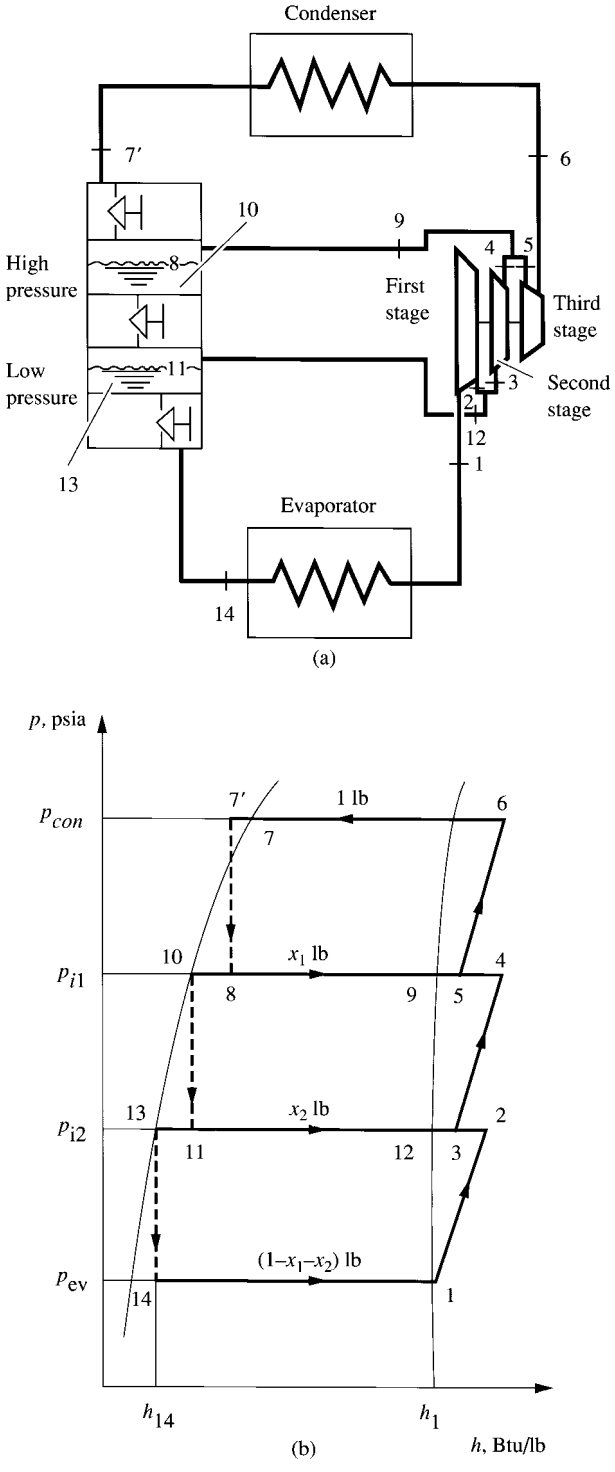


FIGURE 9.9 Three-stage compound system with a two-stage flash cooler: (*a*) schematic diagram; (*b*) refrigeration cycle.

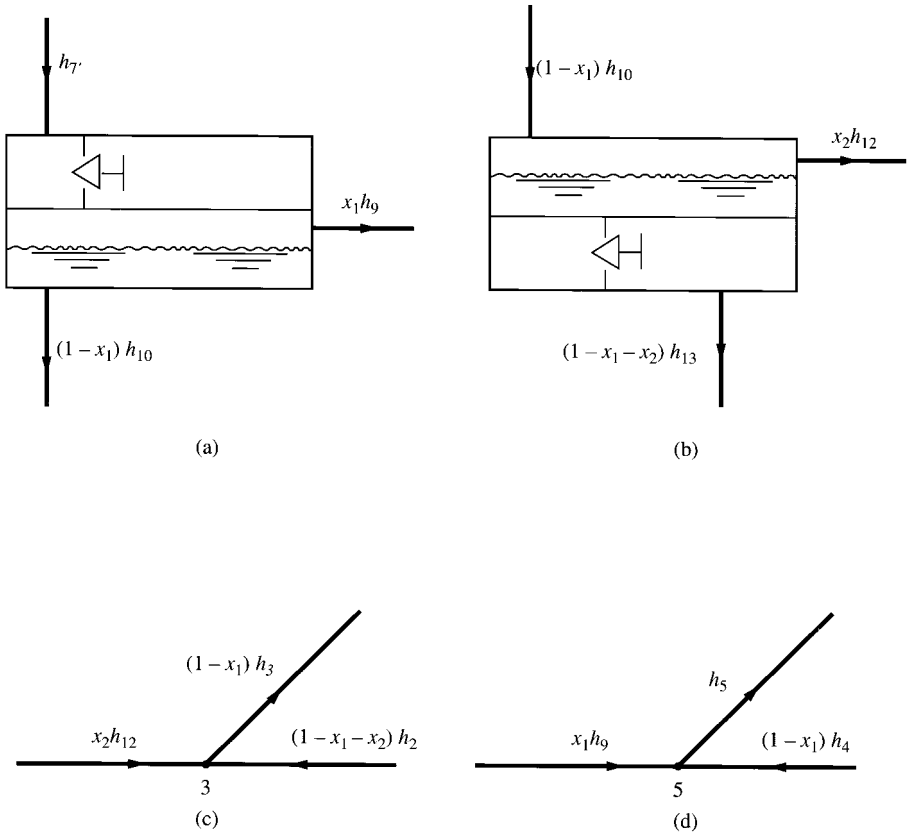


FIGURE 9.10 Heat balance of refrigerants entering and leaving the high-pressure and low-pressure flash cooler and mixing points. (*a*) High-pressure flash cooler; (*b*) low-pressure flash cooler; (*c*) at the mixing point before entering second-stage impeller; (*d*) at the mixing point before entering third-stage impeller.

x_1 at interstage pressure p_{i1} can be calculated as

$$h_{7'} = (1 - x_1)h_{10} + x_1h_9$$

Therefore,

$$x_1 = \frac{h_{7'} - h_{10}}{h_9 - h_{10}} \tag{9.36}$$

where $h_{7'}$, h_9, h_{10} = enthalpies of refrigerants at points 7′, 9, and 10, respectively, Btu/lb (kJ/kg). In the same manner, the heat balance of the refrigerant entering and leaving the low-pressure flash cooler, as shown in Fig. 9.10*b*, may be expressed as

$$(1 - x_1)h_{10} = x_2h_{12} + (1 - x_1 - x_2)h_{13}$$

where h_{12}, h_{13} = enthalpies of the refrigerant at points 12, and 13, respectively, Btu/lb (kJ/kg). The fraction of liquid refrigerant vaporized in the low-pressure flash cooler x_2 at an interstage pressure p_{i2} can be evaluated as

$$x_2 = \frac{(1 - x_1)(h_{10} - h_{13})}{h_{12} - h_{13}} \tag{9.37}$$

Coefficient of Performance of Three-Stage System

From a heat balance of the refrigerants entering and leaving the mixing point before the inlet of the second-stage impeller, as shown in Fig. 9.9*c*, the enthalpy of the mixture at point 3, h_3 (Btu/lb or kJ/kg) is given as

$$h_3 = \frac{(1 - x_1 - x_2)h_2 + x_2 h_{12}}{1 - x_1} \tag{9.38}$$

where h_2 = enthalpy of the gaseous refrigerant discharged from the first-stage impeller at point 2, Btu/lb (kJ/kg). As shown in Fig. 9.9*d*, the enthalpy of the mixture of vapor refrigerants at point 5, h_5, Btu/lb (kJ/kg), can also be evaluated as

$$h_5 = (1 - x_1)h_4 + x_1 h_9 \tag{9.39}$$

where h_4 = enthalpy of the gaseous refrigerant discharged from the second-stage impeller at point 4, Btu/lb (kJ/kg). The refrigeration effect q_{rf} in Btu/lb of refrigerant flowing through the condenser is given as

$$q_{rf} = (1 - x_1 - x_2)(h_1 - h_{14}) \tag{9.40}$$

where h_1, h_{14} = enthalpies of the refrigerants leaving the evaporator at point 1 and entering the evaporator at point 14, respectively, Btu/lb (kJ/kg). Total work input to the three-stage compressor W_{in}, Btu/lb (kJ/kg) of refrigerant flowing through the condenser, is given by

$$W_{\text{in}} = (1 - x_1 - x_2)(h_2 - h_1) + (1 - x_1)(h_4 - h_3) + h_6 - h_5 \tag{9.41}$$

where h_6 = enthalpy of the hot gas discharged from the third-stage impeller at point 6, Btu/lb (kJ/kg).

From Eqs. (9.40) and (9.41), the coefficient of performance of this three-stage system with a two-stage flash cooler is

$$\begin{aligned} \text{COP}_{\text{ref}} &= \frac{q_{rf}}{W_{\text{in}}} \\ &= \frac{(1 - x_1 - x_2)(h_1 - h_{14})}{(1 - x_1 - x_2)(h_2 - h_1) + (1 - x_1)(h_4 - h_3) + (h_6 - h_5)} \end{aligned} \tag{9.42}$$

A three-stage compound system with a two-stage flash cooler often has a further energy saving of about 2 to 5 percent compared to a two-stage compound system with a flash cooler.

The mass flow rate of the refrigerant flowing through the condenser $\dot{m}_r$, lb/h (kg/h), can be calculated as

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{q_{rc}}{(1 - x_1 - x_2)(h_1 - h_{14})} \tag{9.43}$$

9.15 TWO-STAGE COMPOUND SYSTEM WITH A VERTICAL INTERCOOLER

When an evaporating temperature in the range of −10 to −50°F (−23.3 to −45.6°C) is required, a two-stage compound system using reciprocating or screw compressors is usually applied. Figure 9.11 shows the schematic diagram and the refrigeration cycle of a two-stage compound system with a vertical coil intercooler. The subcooled liquid refrigerant from the receiver at point 5′ is divided into two streams. One stream enters the coil inside the intercooler. The other stream enters its shell after throttling to point 6, the interstage pressure.

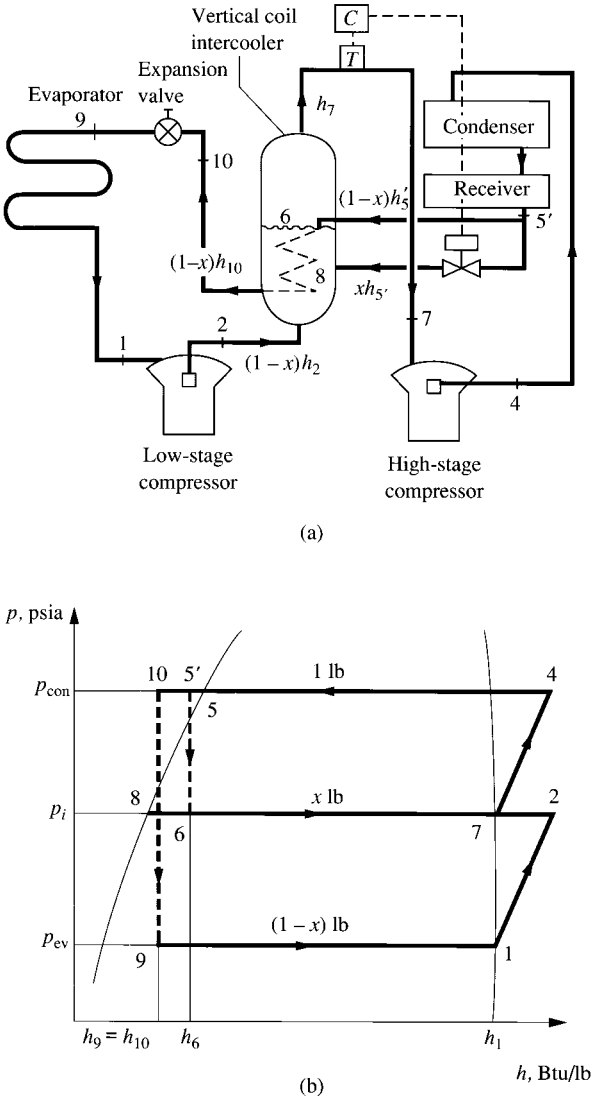


FIGURE 9.11 Two-stage compound system with a vertical coil intercooler: (*a*) schematic diagram; (*b*) refrigeration cycle.

In the intercooler shell, some of the liquid refrigerant vaporizes to saturated vapor at point 7, drawing latent heat from the liquid in the coil at point 5′, further subcooling it to point 10. This subcooled liquid is throttled by the expansion valve at point 10 and then evaporates to saturated vapor at point 1 in the evaporator. Vapor refrigerant from the evaporator at point 1 enters the low-stage compressor. The compressed hot gas at point 2 discharges into the intercooler, mixing with the liquid from the receiver at the interstage pressure. The liquid level in the intercooler is controlled by the saturated temperature at the interstage pressure in the intercooler. The saturated vapor from the vertical coil intercooler at point 7 enters the high-stage compressor. At point 4, hot gas is

discharged from the high-stage compressor and then condensed and subcooled to point 5′ in the condenser.

In this system, x is the fraction of liquid refrigerant vaporized in the intercooler, and h_{10} is the enthalpy of the liquid refrigerant that has been subcooled in the vertical coil. Based on the heat balance of the refrigerants entering and leaving the intercooler, as shown in Fig. 9.11*a*,

$$(1 - x)h_{5'} + xh_5 + (1 - x)h_2 = h_7 + (1 - x)h_{10}$$

Then

$$x = \frac{h_2 - h_7 + h_{5'} - h_{10}}{h_2 - h_{10}} \tag{9.44}$$

This type of system is often used in low-temperature refrigerated cold storage and other industrial applications. Ammonia is often used as the refrigerant.

Comparison between Flash Cooler and Vertical Coil Intercooler

Hot gas discharged from the low-stage compressor is always desuperheated to a nearly saturated vapor state at the interstage pressure in the vertical coil intercooler. This process is more appropriate for a refrigerant like ammonia, which has a high discharge temperature. In flash coolers, desuperheating is caused by the mixing of flashed vapor and hot gas, and will not result in a dry saturated state. Therefore, flash coolers are usually used in refrigeration systems using HCFCs or HFCs.

The liquid refrigerant flowing inside the coils of a vertical coil can be maintained at a slightly lower pressure than condensing pressure, whereas the pressure of liquid refrigerant in the flash cooler is decreased to the interstage pressure. Some refrigerant may be preflashed before the throttling device, causing a waste of refrigerating capacity. For a flash cooler, the available pressure drop in the throttling device is lower.

9.16 CASCADE SYSTEMS

A cascade system consists of two separate single-stage refrigeration systems: a lower system that can better maintain lower evaporating temperatures and a higher system that performs better at higher evaporating temperatures. These two systems are connected by a cascade condenser in which the condenser of the lower system becomes the evaporator of the higher system as the higher system's evaporator takes on the heat released from the lower system's condenser.

It is often desirable to have a heat exchanger between the liquid refrigerant from the cascade condenser and the vapor refrigerant leaving the evaporator of the lower system. The liquid refrigerant can be subcooled to a lower temperature before entering the evaporator of the lower system, as shown in Fig. 9.12*a*. Because the evaporating temperature is low, there is no danger of too high a discharge temperature after the compression process of the lower system.

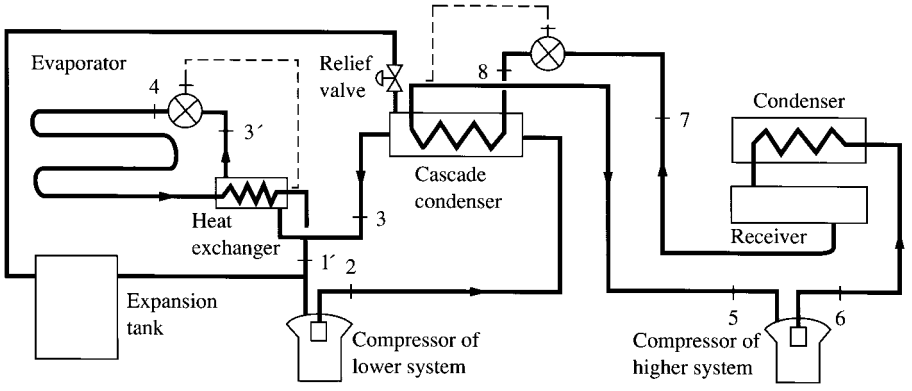
When a cascade system is shut down while the temperature of the ambient air is 80°F (26.7°C), the saturated vapor pressure of the refrigerant increases. For a lower system using HFC-125 as the refrigerant, this saturated pressure may increase to 208.91 psia (1440 kPa abs.). For safety reasons, a relief valve at the cascade condenser connects to an expansion tank, designed to store the refrigerant from the lower system in case of shutdown. For extremely low evaporating temperatures, a multistage compression system may be used in either the lower or higher system of a cascade system.

Advantages and Disadvantages

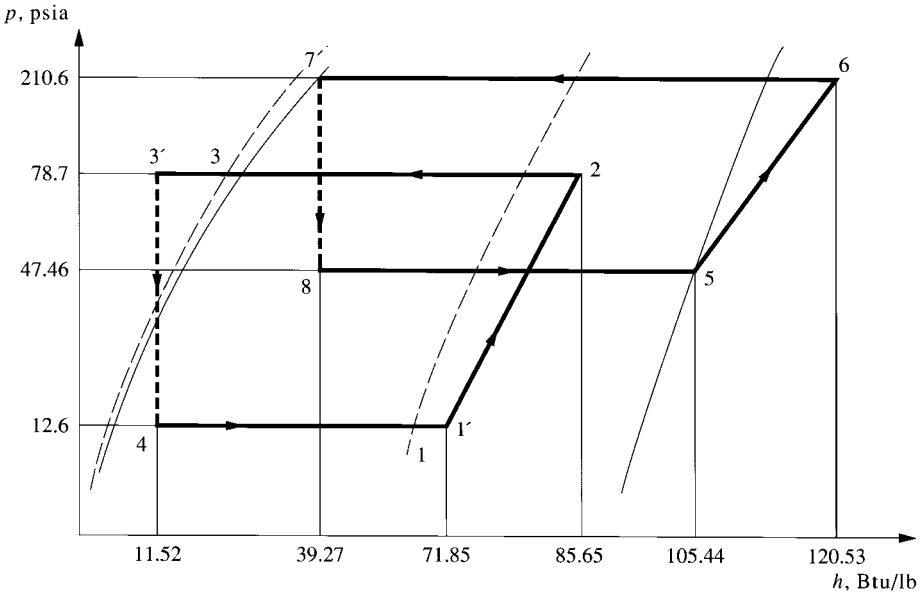
The main advantage of a cascade system is that different refrigerants, equipment, and oils can be used for the higher and the lower systems. This is especially helpful when the evaporating

temperature required in the lower system is less than −100°F (−73°C). The specific volume of the suction vapor v_{suc} is extremely important in such temperature applications. A greater v_{suc} always requires a large compressor and more space. For instance, at −80°F (−62.2°C), saturated vapor of HCFC-22 has a v_{suc} of 9.70 ft^3/lb (0.605 m^3/kg), whereas for HFC-125 v_{suc} is only 4.67 ft^3/lb (0.291 m^3/kg).

One disadvantage of a cascade system is the overlap of the condensing temperature of the lower system and the evaporating temperature of the higher system for heat transfer in the condenser. The overlap results in higher energy consumption. Also a cascade system is more complicated than a compound system.

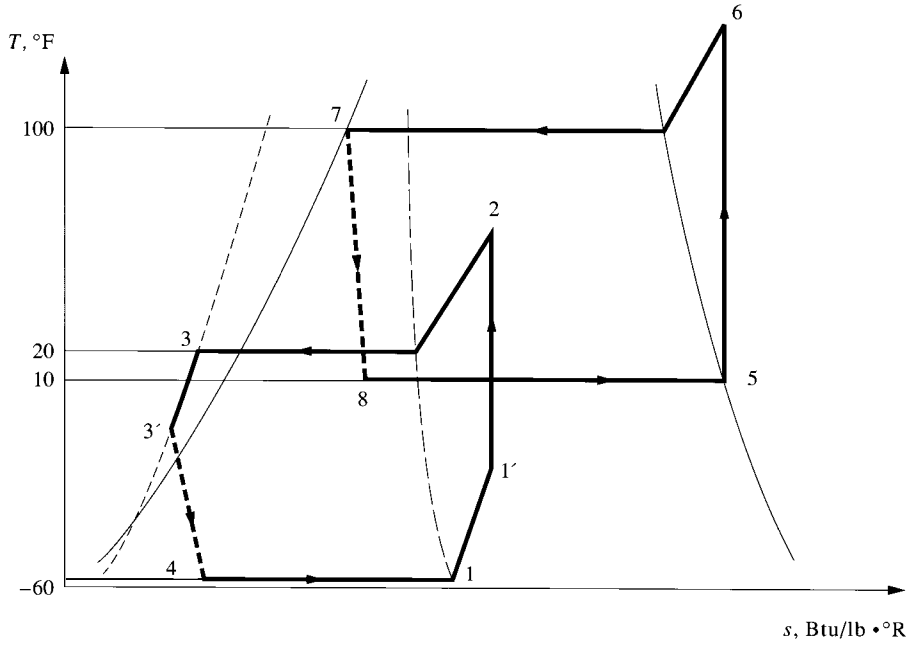


(a)



(b)

FIGURE 9.12 Cascade system: (*a*) schematic diagram; (*b*) *p-h* diagram; (c) *T-s* diagram.



(c)

FIGURE 9.12 (*Continued*)

Performance of Cascade System

The performance of the cascade system can be measured in terms of 1 lb (kg) of refrigerant in the lower system, for the sake of convenience. In Fig. 9.12*b* and *c*, which shows the refrigeration cycles on *p-h* and *T-s* diagrams, the lower system is indicated by points 1, 2, 3′, and 4, and the higher system by points 5, 6, 7, and 8. The desired low-temperature refrigeration effect of the cascade system can be calculated as

$$q_{rf} = h_1 - h_4 \tag{9.45}$$

If the heat transfer between the cascade condenser and the surroundings is ignored, then the heat released by the condenser of the lower system $\dot{m}_l(h_2 - h_3)$ is equal to the refrigerating load on the evaporator of the higher system $\dot{m}_h(h_5 - h_8)$, that is

$$\dot{m}_l(h_2 - h_3) = \dot{m}_h(h_5 - h_8)$$

where $\dot{m}_l, \dot{m}_h =$ mass flow rate of refrigerants in lower and higher systems respectively, lb/h (kg/s). Therefore, the ratio of the mass flow rate of refrigerant in the higher system to the mass flow rate in the lower system is

$$\frac{\dot{m}_h}{\dot{m}_l} = \frac{h_2 - h_3}{h_5 - h_8} \tag{9.46}$$

The mass flow rate of refrigerant in the lower system $\dot{m}_l$, lb/h (kg/s), is

$$\dot{m}_l = \frac{q_{rc,l}}{q_{rf,l}} \tag{9.47}$$

where $q_{rc,l}$ = refrigeration capacity of lower system, Btu/h (W)
$q_{rf,l}$ = refrigeration effect of lower system, Btu/lb (J/kg)

Total work input to the compressors in both higher and lower systems W_{in}, Btu/lb (J/kg), of refrigerant in the lower system can be evaluated as

$$W_{\text{in}} = (h_2 - h_{1'}) + \frac{\dot{m}_h(h_6 - h_5)}{\dot{m}_l} \tag{9.48}$$

where $h_{1'}$ = enthalpy of vapor refrigerant leaving heat exchanger, Btu/lb (J/kg). The coefficient of performance of the cascade system is given as

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{\text{in}}} = \frac{\dot{m}_l(h_1 - h_4)}{\dot{m}_l(h_2 - h_{1'}) + \dot{m}_h(h_6 - h_5)} \tag{9.49}$$

Example 9.3. A cascade system for an environmental chamber is designed to operate at the following conditions during summer:

	Lower system	Higher system
Refrigerant	HFC-125	HCFC-22
Evaporating temperature,°F	−60	10
Evaporating pressure, psia	12.75	47.46
Condensing temperature,°F	20	100
Condensing pressure, psia	78.5	210.6
Suction temperature,°F	−20	
Subcooling after heat exchanger,°F	1	
Refrigerating load, Btu/h	100,000	

Vapor refrigerant enters the compressor of the higher system at a dry saturated state, and liquid refrigerant leaves the condenser of the higher system without subcooling. Ignore the pressure losses in the pipelines and valves, and assume that the compression processes for both the higher and lower systems are isentropic.

Based on the data from the Properties of Saturated Liquid and Saturated Vapor of HFC-125 listed in *ASHRAE Handbook 1997*, *Fundamentals*, the following formulas can be used to calculate the enthalpies of the refrigerants:

Enthalpy of refrigerant HFC-125 at saturated liquid state at temperature T_{sl} between 0 and 50°F (−17.8 and 10°C) h_{sl}, Btu/lb (kJ/kg), is

$$h_{sl} = 11.04 + 0.300T_{sl}$$

Enthalpy of refrigerant HFC-125 at saturated vapor state at temperature T_{sv} between −80 and −30°F (−62.2 and −34.4°C) h_{sv}, Btu/lb (kJ/kg), is

$$h_{sv} = 63.30 + 0.14[T_{sv} - (-80)]$$

Enthalpy difference of vapor refrigerant HFC-125 along the constant-entropy line when $T_{\text{con}} <$ 30°F (−1.1°C) and $T_{\text{ev}} >$ 80°F (26.7°C) is

$$\Delta h_{sv} = 0.175(T_{\text{con}} - T_{\text{ev}})$$

Here T_{con} represents the condensing temperature and T_{ev} the evaporating temperature, both in °F (°C). The specific heat of saturated vapor of HFC-125 c_{pv}, in the temperature range of −80 to −40°F (−62.2 to −40°C) is 0.14 Btu/lb·°F (586 J/kg·°C). The specific heat of saturated liquid

of HFC-125 c_{pl} in the temperature range of 0 to 50°F (−17.8 to 10°C) is 0.300 Btu/lb · °F (1256 J/kg · °C).

Calculate:

1. The refrigeration effect per pound (kilogram) of refrigerant in the lower system
2. Total work input to the compressors
3. Coefficient of performance of this cascade system
4. Mass flow rates of the refrigerants in the higher and lower systems

Solution

1. From the given values, the enthalpy of the saturated vapor refrigerant at −60°F is

$$\begin{aligned} h_1 &= 63.30 + 0.14[T_{sv} - (-80)] \\ &= 63.30 + 0.14[-60 - (-80)] = 66.10 \text{ Btu/lb} \end{aligned}$$

Enthalpy of saturated liquid refrigerant at a condensing temperature of 20°F is

$$\begin{aligned} h_3 &= 11.04 + 0.30T_{sl} \\ &= 11.04 + 0.30(20) = 17.04 \text{ Btu/lb} \end{aligned}$$

and from Eq. (9.26)

$$\begin{aligned} h_{1'} = h_4 &= h_3 - c_{pl}(T_3 - T_{3'}) \\ &= 17.04 - 0.30(20 - 1) = 11.34 \text{ Btu/lb} \end{aligned}$$

Then the refrigeration effect of the lower system is

$$q_{rf,l} = h_1 - h_4 = 66.12 - 11.34 = 54.78 \text{ Btu/lb } (127.4 \text{ kJ/kg})$$

2. Enthalpy of superheated vapor refrigerant after the heat exchanger in the lower system is

$$\begin{aligned} h_{1'} &= h_1 + c_{pv}(\text{degree of superheat}) \\ &= 66.12 + 0.14[-20 - (-60)] = 71.72 \text{ Btu/lb} \end{aligned}$$

Enthalpy difference $(h_2 - h_{1'})$ along the constant-enthalpy line of the lower system is

$$\begin{aligned} h_2 - h_{1'} &= 0.175\,(T_{\text{con}} - T_{lv}) \\ &= 0.175[20 - (-60)] = 14.0 \text{ Btu/lb} \end{aligned}$$

Enthalpy of the hot gas discharged from the compressor of the lower system is

$$\begin{aligned} h_2 &= h_{1'} + (h_2 - h_{1'}) \\ &= 71.72 + 14.0 = 85.72 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.20), the enthalpy of saturated vapor refrigerant HCFC-22 in the higher system is

$$\begin{aligned} h_5 &= 104.465 + 0.098445T_s - 0.0001226T_s^2 - 9.861 \times 10^{-7}T_s^3 \\ &= 104.465 + 0.098445(10) - 0.0001226(10)^2 - 9.861 \times 10^{-7}(10)^3 \\ &= 104.465 + 0.984 - 0.012 - 0.001 = 105.436 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.22), the enthalpy difference along the constant-entropy line of the higher system is

$$\begin{aligned} h_6 - h_5 &= -0.18165 + 0.21502(T_{s2} - T_{s1}) - 0.0012405(T_{s2} - T_{s1})^2 + 8.1982 \times 10^{-6} \\ &\quad (T_{s2} - T_{s1})^3 \\ &= -0.18165 + 0.21502(100 - 10) - 0.0012405(100 - 10)^2 + 8.1982 \times 10^{-6} \\ &\quad (100 - 10)^3 \\ &= 15.09 \text{ Btu/lb} \end{aligned}$$

According to Eq. (9.20), the enthalpy of liquid refrigerant HCFC-22 at condensing temperature 100°F in the higher system is

$$\begin{aligned} h_7 = h_8 &= 10.409 + 0.26851T_s + 0.00014794\,(T_s^2) + 5.3429 \times 10^{-7}\,T_s^3 \\ &= 10.409 + 0.26851(100) + 0.00014794(100^2) + 5.3429 \times 10^{-7}(100^3) = 39.27 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.46), the ratio of the mass flow rate of refrigerant in the higher system to that of the lower system is

$$\frac{\dot{m}_h}{\dot{m}_l} = \frac{h_2 - h_3}{h_5 - h_8} = \frac{85.72 - 17.04}{105.44 - 39.27} = 1.04$$

Then, from Eq. (9.48), the total work input to the compressors is

$$\begin{aligned} W_{\text{in}} &= (h_2 - h_{1'}) + \frac{\dot{m}_h(h_6 - h_5)}{\dot{m}_l} \\ &= 14.0 + 1.04(15.09) = 29.69 \text{ Btu/lb } (69.06 \text{ kJ/kg}) \end{aligned}$$

3. Coefficient of performance of the cascade system is

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{\text{in}}} = \frac{54.78}{29.69} = 1.85$$

4. Mass flow rate of refrigerant in the lower system is

$$\dot{m}_l = \frac{Q_{rc,l}}{q_{rf,l}} = \frac{100{,}000}{54.78} = 1825 \text{ lb/h } (828 \text{ kg/h})$$

The mass flow rate of refrigerant in the higher system is

$$\dot{m}_h = 1.04\,\dot{m}_l = 1.04(30.42) = 1898 \text{ lb/h } (861 \text{ kg/h})$$

9.17 AIR EXPANSION REFRIGERATION CYCLES

Thermodynamic Principle

According to the steady flow energy equation

$$h_1 + \frac{v_1^2}{2g_c \times 778} + q_{\#} = h_2 + \frac{v_2^2}{2g_c \times 778} + W$$

For an adiabatic process, $q_{\#} = 0$. Assuming that the difference between the kinetic energy $v_1^2/(2g_c)$ and $v_2^2/(2g_c)$ is small compared with the enthalpy difference $h_1 - h_2$, it can be ignored. For air expanding in a turbine and producing work, the above equation can be rewritten as

$$W = h_1 - h_2$$

Usually air can be considered a perfect gas, so

$$\Delta h = c_p \Delta T_R \tag{9.50}$$

where c_p = specific heat of air at constant pressure, Btu/lb · °R (kJ/kg · K)
T_R = absolute temperature, °R (K)

and

$$W = c_p(T_{R1} - T_{R2}) \tag{9.51}$$

When the expansion process is a reversible adiabatic process, the temperature-pressure relationship can be expressed as

$$\frac{T_{R1}}{T_{R2}} = \left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} \tag{9.52}$$

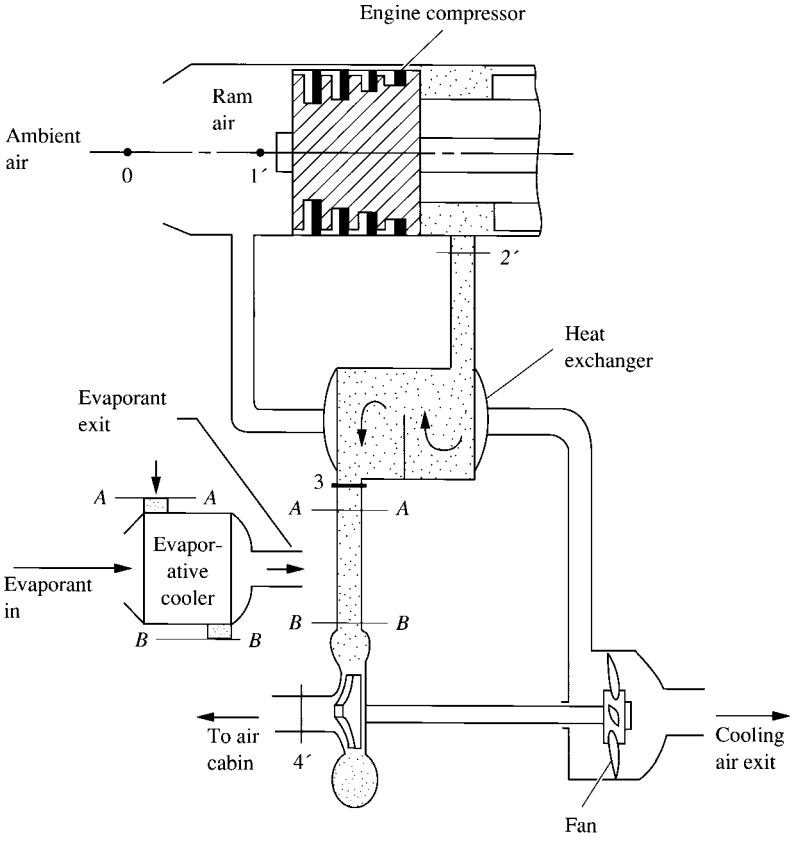
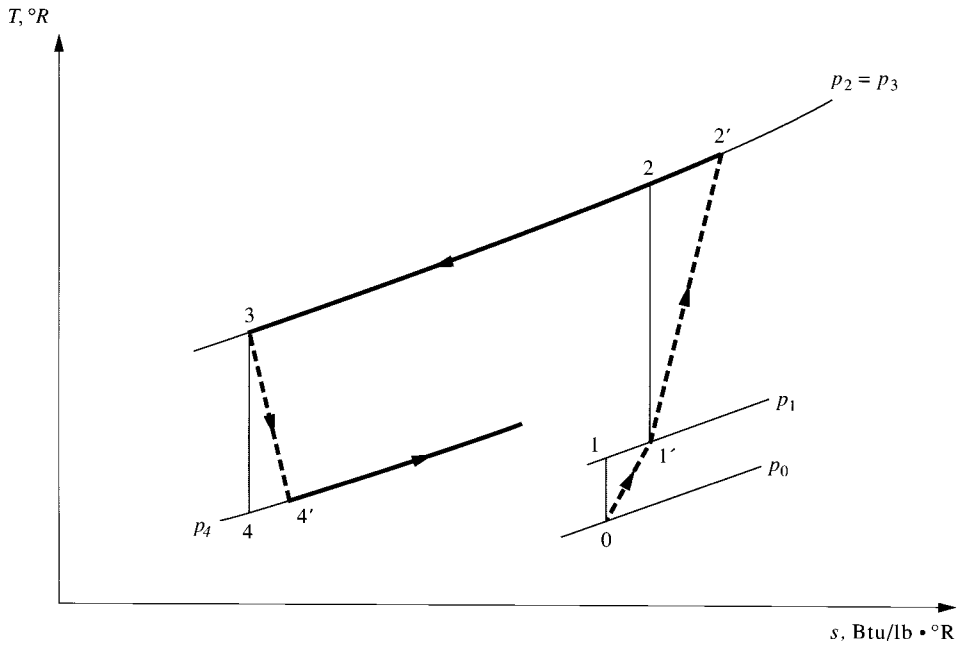


FIGURE 9.13 Air expansion refrigeration system and refrigeration cycle. (*a*) refrigeration system; (*b*) air expansion refrigeration cycle.



(b)

FIGURE 9.13 (*Continued*)

where p_1, p_2 = air pressure at inlet and outlet of turbine, psia (kPa abs.)
γ = ratio of specific heat at constant pressure to specific heat at constant volume

Equations (9.51) and (9.52) demonstrate that when air expands in a turbine and produces work, its absolute temperature T_{R1} drops to T_{R2}. The decrease in absolute temperature is proportional to the pressure ratio $(p_1/p_2)^{(\gamma-1)/\gamma}$ and the work produced W. The greater the pressure drop in the turbine, the lower the temperature at the outlet of the turbine. This type of refrigeration has been used for air conditioning systems in many aircraft.

Flow Processes of a Basic Air Expansion System for Aircraft

Figure 9.13*a* is a schematic diagram of a basic air expansion refrigerating system, and Fig. 9.13*b* depicts the air expansion refrigeration cycle. In Fig. 9.13, ambient air that surrounds an aircraft flying at subsonic speed and at high altitude, at point 0 with pressure p_0, is rammed into the engine scoop. Before entering the engine, the pressure of the air at point $1'$ is increased to p_1. When ram air enters the engine compressor, it is compressed to pressure p_2 and bleeds to a heat exchanger at point $2'$. In the heat exchanger, the bleed air is cooled at constant pressure to point 3 by another stream of ram air extracted by a fan. Bleed air is then expanded in a turbine and cooled to point $4'$. This cold air is supplied to the cabin and flight deck for air conditioning. Cabin air is then discharged to the atmosphere after absorbing the sensible and latent heat from these conditioned spaces. Air expansion refrigeration cycles for aircraft are open cycles because there is no recirculated air.

Another alternative is to add an evaporative cooler between section *A-A* and *B-B*, as shown in Fig. 9.13, to cool the supply air a few degrees more. An evaporant such as water or alcohol can be used to extract heat from the supply air by indirect contact.

The stream of ram air, extracted by a fan driven by the cooling turbine to cool the bleed air, is also discharged to the atmosphere at the cooling air exit. The work output of the turbine is mainly used to drive the fan.

Air Expansion Cycle

The air expansion refrigeration cycle is different from a vapor compression cycle because the refrigerant, air, remains in gaseous form, does not evaporate, and condenses. Air is also not recirculated and is used directly as the supply air for cooling the conditioned spaces.

Work input to the engine compressor W_c, Btu/lb (kJ/kg), is given by

$$W_c = c_p(T_{R2'} T_{R1'}) \tag{9.53}$$

where $T_{R1'}$, $T_{R2'}$ = absolute temperature of ram air and air after compression process, respectively, °R (K). The rate of heat transfer in the heat exchanger between the bleed air and the ambient air for cooling Q_{ex}, Btu/h (W), can be calculated as

$$Q_{ex} = \dot{m}_b c_p(T_{R2'} - T_{R3}) \tag{9.54}$$

where T_{R3} = absolute temperature of air after heat exchanger, °R (K)
$\dot{m}_b$ = mass flow rate of bleed air, lb/h (kg/s)

Work output from the turbine W_o during the expansion process, Btu/lb (J/kg), can be calculated as

$$W_o = c_p(T_{R3} - T_{R4'}) \tag{9.55}$$

where $T_{R4'}$ = absolute temperature of cold supply air at exit of turbine, °R (K). The amount of sensible heat Q_{sen} absorbed by the cold supply air in the air cabin or in the flight deck, Btu/h (W), is

$$Q_{sen} = \dot{m}_b c_p(T_{Rr} - T_{R4'}) \tag{9.56}$$

where T_{Rr} = absolute temperature of air cabin, °R (K). The relationship between temperature and pressure for an isentropic compression process is

$$\frac{T_{R2}}{T_{R1'}} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} \tag{9.57}$$

For an isentropic expansion process, the relationship between temperature and pressure is

$$\frac{T_{R3}}{T_{R4}} = \left(\frac{p_3}{p_4}\right)^{(\gamma-1)/\gamma} \tag{9.58}$$

where T_{R2}, T_{R4} = absolute temperature of air after isentropic compression and isentropic expansion processes, °R (K).

The isentropic efficiency for the compressor η_c is

$$\eta_c = \frac{c_p(T_{R2} - T_{R1'})}{c_p(T_{R2'} - T_{R1'})} = \frac{T_{R2} - T_{R1'}}{T_{R2'} - T_{R1'}} \tag{9.59}$$

Similarly, isentropic efficiency for the turbine η_t is

$$\eta_t = \frac{T_{R3} - T_{R4'}}{T_{R3} - T_{R4}} \tag{9.60}$$

If η_c, η_t, $T_{R1'}$, T_{R3}, and pressure ratios p_2/p_1 and p_3/p_4 are known values, $T_{R2'}$, and $T_{R4'}$, can be determined by using Eqs. (9.57) to (9.60).

The effectiveness of the heat exchanger ε can be evaluated as

$$\varepsilon = \frac{c_p \dot{m}_b (T_{R2'} - T_{R3})}{c_p \dot{m}_b (T_{R2'} - T_{Rt1})} = \frac{T_{R2'} - T_{R3}}{T_{R2'} - T_{Rt1}} \tag{9.61}$$

where $T_{R2'}$, T_{R3} = absolute temperature of bleed air entering and leaving heat exchanger, °R (K)
T_{Rt1} = absolute total temperature of cooling ram air entering heat exchanger, measured at outer heat-transfer surface at rest, °R (K)

From Eq. (9.61), ε depends mainly on the construction of the heat exchanger and the ratio $\dot{m}_b/\dot{m}_c$, if the difference of c_p between bleed and cooling ram air is ignored. Here $\dot{m}_c$ indicates the mass flow rate of cooling ram air, lb/h (kg/s).

If air at a velocity v with an absolute temperature T_R finally reaches a total temperature T_{Rt} when brought to rest adiabatically, then, based on the steady flow energy equation;

$$c_p T_R + \frac{v_2}{2g_c \times 778} = c_p T_{Rt}$$

or

$$T_{Rt} = T_R + \frac{v^2}{2c_p(778g_c)} \tag{9.62}$$

When the air velocity $v < 200$ ft/s (61 m/s), the term $v^2/[(2c_p)(778g_c)]$ is small compared with T_R and can be ignored. Hence, only T_{Rt} is used for cooling ram air.

Net work input to the air expansion refrigeration cycle is calculated as

$$W_{\text{net}} = W_c - W_t = c_p(T_{R2'} - T_{R1'}) - c_p(T_{R3} - T_{R4'}) \tag{9.63}$$

If W_t is used to drive the fan to extract the cooling air, then the coefficient of performance of the basic air expansion refrigeration cycle can be calculated as

$$\text{COP}_{\text{ref}} = \frac{c_p(T_{Rr} - T_{R4'})}{c_p(T_{R2'} - T_{R1'})} = \frac{T_{Rr} - T_{R4'}}{T_{R2'} - T_{R1'}} \tag{9.64}$$

The main advantage of an air expansion refrigeration system for an aircraft is its simple construction and light weight. The main drawbacks are the lower COP_{ref} and the need for an additional air conditioning and refrigeration system when the aircraft is on the ground.

9.18 REFRIGERATION SYSTEMS—CLASSIFICATIONS AND DEVELOPMENTS

Classifications

As discussed in Sec. 9.1, based on the type of energy input and refrigeration process, refrigeration systems can be classified as vapor compression, absorption, or air/gas expansion systems. In the 1990s, the absorption system has a share of only less than 8 percent in new installations compared

with centrifugal chillers in the United States. Applications of air or gas expansion refrigeration systems are limited in aircraft and cryogenics. Vapor compression systems dominate in both comfort and processing air conditioning.

Vapor compression refrigeration systems can also be classified according to

- The type of compressor used, such as reciprocating, rotary, scroll, screw, or centrifugal systems.
- The type of evaporator: direct-expansion (DX) cooler or liquid chillers. In a DX cooler system, space air is directly cooled and dehumidified by the expanded and evaporated refrigerant in a DX coil. A liquid chiller is a factory-assembled refrigeration package in which chilled water is produced in its evaporator. Air is then cooled and dehumidified by chilled water in air-handling units (AHUs) or in fan coils.
- The size (tonnage in capacity) of the refrigeration system, such as small (< 2.5 ton, or 8.8 kW), medium, and large (≥ 75 tons or 264 kW).

For a refrigeration system, the type of compressor, system size, and whether a DX coil or a water chiller is used are often interrelated. A centrifugal chiller is often a large-tonnage refrigeration system. A small refrigeration system with a cooling capacity of less than 2.5 tons (8.8 kW) is often an air-cooled DX cooler using a reciprocating, rotary, or scroll compressor. Refrigeration systems can therefore be primarily classified as follows:

- DX (direct-expansion) cooler systems:
  - Air-cooled, reciprocating DX coolers
  - Air-cooled, scroll DX coolers
  - Air-cooled rotary DX coolers
- Liquid chillers
  - Centrifugal chillers
  - Screw chillers
  - Scroll chillers
  - Reciprocating chillers
- Liquid overfeed systems
- Multistage systems
- Absorption systems
- Air or gas expansion systems

Among these, DX coolers have a share of about 70 percent based on the air conditioned floor area in commercial buildings in 1992. For air conditioned homes, more than 90 percent use DX coolers. Chillers have a share of about 28 percent of floor space in air conditioned commercial buildings. According to the U.S. domestic market research 1997 estimate provided by BSRIA/Ducker in *Air Conditioning, Heating and Refrigeration News*, September 22, 1997, for a total number of nearly 18,460 chillers sold, the large-tonnage centrifugal and screw chillers with a capacity > 75 tons (264 kW) had a share of 28 percent in units and a value of about \$505 million. Reciprocating, screw, and scroll chillers with a capacity < 75 tons (264 kW) using step unloading had a share of 61 percent in units and a value of about \$380 million. Absorption and gas-engine chillers had a value of about \$66 million.

Refrigeration systems are also classified according to their evaporating temperatures T_{ev} as

Low-temperature systems	$-40°\text{F} \leq T_{ev} \leq 0°\text{F}$ $(-40°\text{C} \leq T_{ev} \leq -18°\text{C})$
Medium-temperature systems	$0°\text{F} < T_{ev} \leq 32°\text{F}$ $(-18°\text{C} < T_{ev} \leq 0°\text{C})$
High-temperature systems	$T_{ev} > 32°\text{F}$ $(T_{ev} > 0°\text{C})$

ASHRAE Standard 15-1994 classifies refrigeration systems as direct systems and indirect systems based on the method employed for extracting or delivering heat. A direct refrigeration system is one in which the evaporator or condenser of the refrigeration system is in direct contact with air or other media to be cooled or heated. An indirect refrigeration system uses a secondary coolant to cool the air or other substance after the coolant is cooled by the refrigeration system.

According to the probability that leakage of refrigerant will enter the occupied space, ASHRAE Standard 15-1994 divides refrigeration systems into the following two categories:

High-probability system. Any refrigeration system in which a leakage of refrigerant from a failed connection, seal, or component could enter the occupied space (the space is frequently occupied by people) is a high-probability system. A direct system is a high-probability system.

Low-probability system. This is a refrigeration system in which leakage of refrigerant from a failed connection, seal, or component cannot enter the occupied space. An indirect system that uses secondary coolants, such as chilled water, brines, or glycols, is a low-probability system.

Recent Developments

In the United States in the 1990s, refrigeration systems underwent the following major developments:

- The production of CFC and halons in developed countries ceased from January 1, 1996. However, the conversion of CFCs to non–ozone depletion HFCs and HCFCs was slower than expected.
- For small and medium-size refrigeration systems, the direct-expansion systems using reciprocating, rotary, scroll, and screw compressors dominate in the individual and packaged air conditioning systems. For large-size refrigeration systems, centrifugal and screw chillers are widely adopted.
- Energy-efficient rotary, scroll, and screw compressors are gradually replacing reciprocating compressors. According to *Air Conditioning, Heating and Refrigerating News*, May 6, 1996, of the compressors in new packaged units and room air conditioners 35 percent are scroll compressors. Based on BSRIA/Ducker analyses for the U.S. domestic market in 1996, rotary compressors dominated the room air conditioner market of less than 2 tons (7 kW), scroll compressors dominated the small chillers, screw compressors had a two-thirds share at the top of medium-size units (3 to 75 tons, or 10 to 264 kW), and reciprocating chillers showed a share of 70 percent of the units with a cooling capacity around 5 tons (18 kW).
- The energy use of new centrifugal compressors dropped considerably from an average of 0.85 kW/ton (4.14 COP) in the late 1970s in the United States to 0.60 kW/ton (5.86 COP) in 1996. In 1997, the lowest-energy-use new centrifugal chiller is 0.49 kW/ton (7.17 COP).

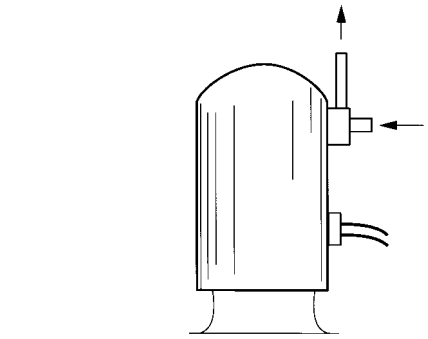
The rotary, scroll, screw, and centrifugal compressors will form a series of rotary refrigeration products to meet mainly the requirements of new and retrofit, small, medium, and large refrigeration systems in the twenty-first century. Air conditioning becomes the largest user of refrigeration.

9.19 REFRIGERATION COMPRESSORS

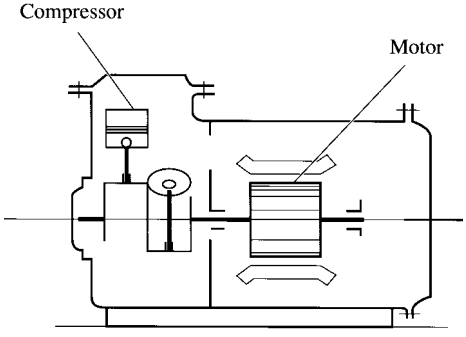
A refrigeration compressor is the heart of a vapor compression refrigeration system. Its function is to raise the pressure of the refrigerant and provide the primary force to circulate the refrigerant. The refrigerant thus produces the refrigeration effect in the evaporator, condenses into liquid form in the condenser, and throttles to a lower pressure through the throttling device.

Positive Displacement and Non–Positive Displacement Compressors

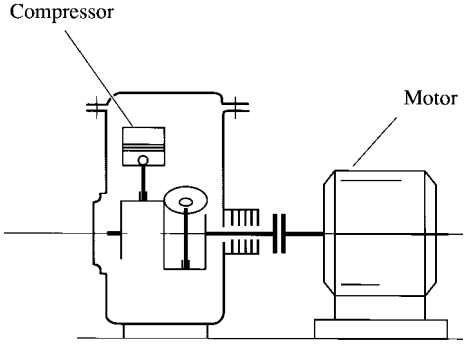
According to the characteristics of the compression process, currently used refrigeration compressors can be classified as positive displacement compressors or non–positive displacement



(a)



(b)



(c)

FIGURE 9.14 Hermetic and open-type compressors: (*a*) hermetic; (*b*) semihermetic; and (*c*) open.

compressors. A positive displacement compressor increases the pressure of the vapor refrigerant by reducing the internal volume of the compression chamber through mechanical force applied to the compressor. This type of refrigeration compressor mainly includes reciprocating, scroll, screw, and rotary compressors.

The only type of non–positive displacement refrigeration compressor widely used in refrigeration systems is the centrifugal compressor. In a centrifugal compressor, the increase of the pressure of the vapor refrigerant depends mainly on the conversion of dynamic pressure to static pressure.

Hermetic, Semihermetic, and Open Compressors

In a hermetic compressor, the motor and the compressor are "sealed" or "welded" in the same housing, as shown in Fig. 9.14*a*. Hermetic compressors have two advantages: They minimize leakage of refrigerant, and the motor can be cooled by the suction vapor flowing through the motor windings, which results in a small and cheaper compressor-motor assembly.

Motor windings in hermetic compressors must be compatible with the refrigerant and lubrication oil, resist the abrasive effect of the suction vapor, and have high dielectric strength. Welded compressors are usually used for small installations from $<$ 1 hp to 24 hp ($<$ 0.7 kW to 18 kW).

Semihermetic compressors are also known as *accessible* hermetic compressors (see Fig. 9.14*b*). The main advantage that semihermetic compressors have over hermetic compressors is accessibility for repair during a compressor failure or for regular maintenance. Other features are similar to those of the hermetic compressor. Most of the medium compressors are semihermetic.

In an open compressor, the compressor and the motor are enclosed in two separate housings, as shown in Fig. 9.14*c*. An open compressor needs shaft seals to minimize refrigerant leakage. In most cases, an enclosed fan is used to cool the motor windings by using ambient air. An open compressor does not need to evaporate the liquid refrigerant to cool the hermetic motor windings. Compared with hermetic compressors, open compressors may save 2 to 4 percent of the total power input. Many very large refrigeration compressors are open compressors.

Direct Drive, Belt Drive, and Gear Drive

Hermetic compressors are driven by motor directly or driven by gear trains. Both semihermetic and open compressors can be driven directly, driven by gear trains, or driven by motor through V belts. The purpose of a gear train is to increase the speed of the compressor. Gear drive is compact in size and rotates without slippage. Like belt drive, gear drive needs about 3 percent more power input than direct-drive compressors. Some large open compressors may be driven by steam turbine, gas turbine, or diesel engine instead of electric motor.

9.20 PERFORMANCE OF COMPRESSORS

Volumetric Efficiency

The *volumetric efficiency* η_v of a refrigeration compressor is defined as

$$\eta_v = \frac{V_{a,v}}{V_{\text{dis}}} \tag{9.65}$$

where $V_{a,v}$ = actual induced volume of suction vapor at suction pressure, ft^3 (m^3)
V_{dis} = theoretical displacement of compressor, ft^3 (m^3)

Factors that influence the η_v of the compressor include:

- *Clearance volume and compression ratio* R_{com}. Both factors affect the volume of reexpansion gas trapped in the clearance volume.

- *Heating effect*. When vapor refrigerant enters the compressor, heat absorbed by the vapor results in a heating effect that increases the specific volume of the refrigerant and, therefore, the $V_{a,v}$ value.
- *Leakage*. Refrigerant leaks through the gap and the clearance across the high- and low-pressure sides of the compressor, such as the clearance between the piston ring and the cylinder in a reciprocating compressor.

Motor, Mechanical, and Compression Efficiency

Motor efficiency η_{mo} is defined as

$$\eta_{\text{mo}} = \frac{P_{\text{com}}}{P_{\text{mo}}} \tag{9.66}$$

where P_{com}, P_{mo} = power input to shaft of compressor and to motor, respectively, hp (kW). *Mechanical efficiency* η_{mec} is defined as

$$\eta_{\text{mec}} = \frac{W_v}{W_{\text{com}}} \tag{9.67}$$

where W_v, W_{com} = work delivered to vapor refrigerant and to compressor shaft, Btu/lb (kJ/kg). Compression efficiency η_{cp} is defined as

$$\eta_{cp} = \frac{W_{\text{isen}}}{W_v} \tag{9.68}$$

where W_{isen} = work required for isentropic compression, Btu/lb (kJ/kg). Here, $W_{\text{isen}} = h_2 - h_1$, where h_1 and h_2 represent the enthalpy of intake vapor refrigerant and of discharged hot gas, respectively, in an isentropic compression process, in Btu/lb (kJ/kg). The difference between work input on the compressor shaft and work delivered to the vapor refrigerant $W_{\text{com}} - W_v$ is mainly caused by the friction loss and turbulent loss of refrigerant flow.

Compressor efficiency η_{com} is the product of η_{cp} and η_{mec}, that is,

$$\eta_{\text{com}} = \eta_{cp}\eta_{\text{mec}} \tag{9.69}$$

Isentropic and Polytropic Analysis

The *isentropic efficiency* η_{isen} of a compressor is defined as

$$\eta_{\text{isen}} = \frac{h_2 - h_1}{h_{2'} - h_1} \tag{9.70}$$

where $h_{2'}$ = enthalpy of discharged hot gas if compression process is not isentropic, Btu/lb (kJ/kg). The difference between $h_{2'}$ and h_2 implies, first, the deviation of a reversible polytropic process from an isentropic process and, second, the deviation of an irreversible polytropic process from a reversible process. Isentropic efficiency is equal to compressor efficiency, or $\eta_{\text{isen}} = \eta_{\text{com}}$.

The actual power input to the compressor P_{com}, hp, can be calculated as

$$P_{\text{com}} = \frac{\dot{m}_r(h_{2'} - h_{1'})}{42.41} = \frac{\dot{m}_r(h_2 - h_1)}{42.41\,\eta_{\text{isen}}} = \frac{\dot{m}_r(h_2 - h_1)}{42.41\,\eta_{\text{com}}} \tag{9.71}$$

In Eq. (9.71), $\dot{m}_r$ represents the mass flow rate of refrigerant, lb/min (kg/min). For reciprocating compressors, $\dot{m}_r$ can be calculated as

$$\dot{m}_r = \dot{V}_p \eta_v \rho_{\text{suc}} \tag{9.72}$$

where $\dot{V}_p$ = piston displacement, cfm (or m^3/min, $\dot{V}_p$ to be covered in Chap. 11)
ρ_{suc} = density of suction vapor, lb/ft^3 (kg/m^3)

Although the actual compression processes for most compressors are irreversible polytropic processes, for simplicity, an isentropic analysis is often used. In other words, η_{isen} is a widely used efficiency index for refrigeration compressors, and actual power input to the compressor P_{com} is usually calculated by Eq. (9.71).

Energy Use Index

According to ASHRAE/IESNA Standard 90.1-1999, the following are the current energy use indices for refrigeration compressors, packaged units, heat pumps, and chillers:

- *Energy efficiency ratio* (EER) is defined as the ratio of the net cooling capacity of a refrigeration compressor, a packaged unit, or other device, in Btu/h, to the electric power input to that device, in W, under designated operating conditions.
- SEER indicates the *seasonal energy efficiency ratio*. This is the total cooling capacity of a device during its normal annual usage period, in Btu, divided by the total electric energy input during the same period, in watt-hours.
- IPLV indicates the *integrated part-load value*. This is a single index of merit that is based on part-load EER or COP. It expresses part-load efficiency for refrigeration compressors, packaged units, and heat pumps based on the weighted operation at various load capacities.
- HSPF indicates the *heating season performance factor*. This is the total heating output of a heat pump during its normal annual usage period for heating, in Btu, divided by the total electric energy input to the equipment during the same period, in watt-hours.
- kW/ton indicates the electric power consumption of a refrigerating compressor per ton of refrigeration output. It is a clear and widely used energy index.

From Eq. (9.71) and because $\dot{m}_r = Q_{rc}/q_{rf}$, the kW/ton for a hermetic refrigeration compressor can be calculated as

$$\text{kW/ton} = \frac{3.516\ W_{\text{in}}}{q_{rf}\eta_{\text{isen}}\eta_{\text{mo}}} = \frac{3.516}{\text{COP}_{\text{ref}}} \tag{9.73}$$

where W_{in} = work input to compressor (including single-, two-, or three-stage) per lb of refrigerant flowing through condenser, Btu/lb (kJ/kg)
q_{rf} = refrigeration effect per lb of refrigerant flowing through condenser, Btu/lb (kJ/kg)
η_{isen} = isentropic efficiency of compressor
η_{mo} = motor efficiency

From Eq. (9.73), primary factors that affect the energy use of the kW/ton index for a refrigeration compressor are as follows:

- Difference between the condensing and evaporating temperature $T_{\text{con}} - T_{\text{ev}}$
- Isentropic efficiency η_{isen} or compressor efficiency η_{com}
- Refrigeration effect of the specific refrigerant

The following are the EERs indicating the energy use of a hermetic refrigeration compressor and motor, and the equivalent COP and kW/ton values:

EER	COP	kW/ton
6	1.758	2.0
8	2.344	1.5
10	2.930	1.2
12	3.516	1.0
15	4.395	0.8
20	5.860	0.6
30	8.790	0.4

EER is widely used for reciprocating and scroll compressors in air-cooled, direct-expansion refrigeration systems. The coefficient of performance and kW/ton are usuallly used for water-cooled centrifugal chillers. They are often rated at different operating conditions. Water-cooled chillers always show a more energy-efficient index than air-cooled DX systems.

9.21 SAFETY REQUIREMENTS AND MACHINERY ROOM

Refrigerant Safety

Refrigerant hazards stemming from leaks in the pipe joints, the rupture of system components, and the burning of escaping refrigerant depend on the type of refrigerant, the occupancy classification, and the refrigeration system. As discussed in Sec. 9.3, refrigerants can be classified into six safety groups that range from lower toxicity and no flame propagation (safety group A1) to higher toxicity and higher flammability (safety group B3). The type of occupancy may be one of the following five categories:

- Institutional or health care, such as hospitals and nursing homes
- Public assembly, such as auditoriums and department stores
- Residential, including hotels and apartments
- Commercial, such as offices, restaurants, and markets
- Industrial, such as factories and warehouses

Application Rules for High-Probability Systems

A tightly sealed refrigerant system is always necessary to reduce refrigerant leaks that may produce refrigerant hazards. In addition, limiting the quantity of refrigerant in a refrigeration system per occupied space, thereby reducing the possible leaks from joints and seals in a high-probability system, is an effective means of reducing the hazards of refrigerants for the safety of people and property.

ASHRAE Standard 15-1994 specifies a number of rules and requirements for various refrigerants in high-probability refrigeration systems. These rules are described in the following paragraphs.

1. Any refrigeration system in a room air conditioner or packaged terminal air conditioner (PTAC), or any small packaged unit for which the refrigerant charge does not exceed 6.6 lb (3 kg), is considered to meet the system refrigerant safety application requirements.

2. Refrigerants that belong to the A1 safety group (HCFC-22, HFC-134a, etc.) and are used in high-probability refrigeration systems have the following restrictions:

For institutional or health care occupancies (except in kitchens, laboratories, or mortuaries), the following conditions apply:

- The quantity of refrigerant HCFC-22 in the largest refrigeration system $m_{\text{HCFC-22}}$ is limited to 4.7 lb/1000 ft^3 (2.13 kg/28.3 m^3) of occupied space.
- The quantity of refrigerant HFC-134a, or $m_{\text{HFC-134a}}$, is limited to 8 lb/1000 ft^3 (3.6 kg/28.3 m^3) of occupied space. A flame-sustaining device in an occupied space must be provided with a hood to exhaust combustion products to open air if the refrigeration system contains more than 1 lb (0.45 kg) of A1 refrigerant (except R-744 carbon dioxide).

For public assembly, residential, and commercial occupancies, the following apply:

- The value of $m_{\text{HCFH-22}}$ in a refrigeration system is limited to 9.4 lb/1000 ft^3 (4.3 kg/28.3 m^3) of occupied space.
- The value of $m_{\text{HFC-134a}}$ in a refrigeration system is limited to 16 lb/1000 ft^3 (7.3 kg/28.3 m^3) of occupied space.

In industrial applications, refer to ANSI/ASHRAE Standard 15-1994 for details.

3. Refrigerants in the B1 safety group, such as HCFC-123, that are used in high-probability refrigeration systems should not be allowed for applications involving air conditioning for human comfort. The quantity of refrigerants in a refrigeration system is limited to under 0.4 lb/1000 ft^3 (0.18 kg/28.3 m^3) occupied space for commercial occupancy and 0.2 lb/1000 ft^3 (0.09 kg/28.3 m^3) occupied space for institutional occupancy.

4. Refrigerants in the B2 safety group, such as R-717 ammonia, that are used in high-probability refrigeration systems are not allowed to be used for institutional occupancy as well as comfort air conditioning in other occupancies. The following rules apply: When R-717 is used in high-probability refrigeration systems in process air conditioning (including food storage), $m_{\text{R-717}}$, lb (kg), the maximum permissible quantity, is limited to the following values:

	Institutional	Public	Residential	Commercial
Sealed absorption system:				
In public hallways or lobbies	0	0	3.3 (1.5)	3.3 (1.5)
In adjacent outdoor locations	0	0	22 (10)	22 (10)
Other than hallways or lobbies	0	6.6 (3)	6.6 (3)	22 (10)
Individual and packaged systems:				
Other than hallways or lobbies	0	0	6.6 (3)	22 (10)

Application Rules for Low-Probability Systems

ASHRAE Standard 15-1994 states the requirements for low-probability refrigeration systems that have complete, factory-assembled chillers tested by an approved and nationally recognized laboratory and that are used in health care, public assembly, residential, commercial, and industrial buildings. The following paragraphs describe these requirements.

1. For A1 safety group refrigerants, if $m_{\text{HCFC-22}}$ in the largest refrigeration system exceeds 9.4 lb/1000 ft^3 (4.3 kg/28.3 m^3) of occupied space or $m_{\text{HFC-134a}}$ exceeds 16 lb/1000 ft^3 (7.3 kg/28.3 m^3) of occupied space, all refrigerant-containing parts, except pipes and parts located outdoors, must be installed in a machinery room that meets general safety requirements.

2. For B1 safety group refrigerants, if $m_{\text{HCFC-123}}$ in the largest refrigeration system exceeds 0.4 lb/1000 ft^3 (0.18 kg/28.3 m^3) of occupied space, all refrigerant-containing parts, except pipes and parts located outdoors, must be installed in a machinery room that meets general safety requirements.

3. For B2 safety group refrigerants such as R-717 ammonia, the requirements are as follows: When $m_{\text{R-717}}$ in the largest refrigeration system exceeds 0.022 lb/1000 ft^3 (0.01 kg/28.3 m^3) of

occupied space, all refrigerant-containing parts, except pipes and those parts outdoors, must be installed in a machinery room that meets special requirements; and $m_{\text{R-717}}$ in the refrigeration system must not exceed 550 lb (250 kg) for institutional occupancy. For details, refer to ASHRAE Standard 15-1994 and local codes.

Refrigerating Systems of 100 hp (74.6 kW) or Less

Separated fan rooms with light construction and with controlled access by authorized personnel may contain refrigeration machinery. Refer to ASHRAE Standard 15-1994 and Section 16.6.

Refrigerating Machinery Room

A refrigerating machinery room is an enclosure with tightly fitted doors to safely house compressors, refrigeration components, and other types of mechanical equipment. A refrigerating machinery room must be designed so that it is easily accessible, with adequate space for proper servicing, maintenance, and operation. A refrigerating machinery room must have doors that open outward and are self-closing if they open into the building; there must be an adequate number of doors to allow easy escape in case of emergency.

According to ASHRAE Standard 15-1994, a refrigerating machinery room must meet the following requirements:

- It must be ventilated to the outdoors by means of mechanical ventilation using power-driven fans or multiple-speed fans. The minimum volume flow rate of mechanical ventilation $V_{m,v}$, cfm (L/s), required to exhaust the refrigerant from leaks or ruptured component is

$$\dot{V}_{m,v} = 100\, m_{\text{ref}}^{0.5} \tag{9.74}$$

  where m_{ref} is the mass of refrigerant in the largest system, in pounds. When a refrigeration system is located outdoors, its location is more than 20 ft (6 m) from any building opening, and it is enclosed by a penthouse or other open structure, then both mechanical and natural ventilation can be used. The minimum free-aperture section A_{fa}, ft^2 (m^2), required for natural ventilation of the machinery room is

$$A_{\text{fa}} = m_{\text{ref}}^{0.5} \tag{9.75}$$

  Provisions for venting catastrophic leaks and component ruptures should be considered.
- For safety group A1 refrigerants such as HCFC-22 and HFC-134a, the machinery room must include an oxygen sensor to warn the operator when oxygen levels drop below 19.5 percent by volume. For other refrigerants, a vapor detector should be installed to actuate the alarm and the mechanical ventilation system.
- There must be no open flames that use combustion air from the machinery room except matches, lighters, leak detectors, and similar devices.

A refrigerating machinery room of special requirements must meet the following specifications in addition to the general requirements:

- Inside the room, there must be no flame-producing device or hot surface continuously operated at a surface temperature exceeding 800°F (427°C).
- There must be an exit door that opens directly to the outdoors or to a similar facility.
- Mechanical ventilation for ammonia must be either continuously operating or equipped with a vapor detector that actuates a mechanical ventilation system automatically at a detection level not exceeding 4 percent by volume.

- It must be provided with remote pilot control panel immediately outside the machinery room to control and shut down the mechanical equipment in case of emergency.

Because the refrigerating machinery room itself is a fire compartment, the building structure and its material (including the door, wall, ceiling, and floor) should meet National Fire Protection Association (NFPA) fire codes. Refrigerating machinery rooms are different from fan rooms where air-handling units with fans are mounted inside. In a refrigerating machinery room, there are general and special requirements. The installation of mechanical ventilation and an oxygen or vapor sensor is mandatory. A central refrigeration plant must be a kind of machinery room.

For more details, refer to ASHRAE Standard 15-1994.

Storage of Refrigerants

Refrigerants are usually stored in cylinders during transport and while on site. During storage, the pressure of the liquid refrigerant must be periodically checked and adjusted. Excessive pressure may cause an explosion. According to Interstate Commerce Commission (ICC) regulations, liquid refrigerants must not be stored above 130°F (54.4°C), although the containers are designed to withstand up to 3 times the saturated pressure at 130°F (54.4 °C). If a container bursts, liquid refrigerant flashes into vapor. Such a sudden expansion in volume could cause a violent explosion inside a building, blasting out windows, walls, and roofs.

Containers should never be located near heat sources without sufficient ventilation. They must also not be put in a car or truck in direct sunlight. The valve of the container is attached by thread only. If the threads are damaged, the force of escaping vapor could propel the container like a rocket.

According to ASHRAE Standard 15-1994, in addition to the refrigerant charge in the system and receiver, refrigerant stored in a machinery room shall not exceed 330 lb (150 kg). The receiver is a vessel used to store refrigerant after the condenser when necessary.

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CHAPTER 10
REFRIGERATION SYSTEMS: COMPONENTS

10.1 EVAPORATORS 10.2

10.2 DX COILS—WET COILS 10.3
- Construction and Installation 10.3
- Two-Region Model 10.4
- Simultaneous Heat and Mass Transfer 10.5
- DX Coil Effectiveness 10.6
- Face Velocity and Air-Side Pressure Drop of Wet Coil 10.7
- DX Coils at Part-Load Operation 10.8
- Selection of DX Coils 10.10

10.3 FLOODED LIQUID COOLER 10.12
- Construction 10.12
- Heat Transfer in a Flooded Liquid Cooler 10.14
- Fouling Factor 10.14
- Pool Boiling and Forced-Convection Model 10.15
- Parameters That Influence Performance of Flooded Liquid Cooler 10.16
- Part-Load Operation of Flooded Liquid Cooler 10.17

10.4 DIRECT-EXPANSION LIQUID COOLER AND LIQUID OVERFEED COOLER 10.18
- Direct-Expansion Liquid Cooler 10.18
- Liquid Overfeed Cooler 10.18

10.5 CONDENSATION AND CONDENSERS 10.20
- Condensation Process 10.20
- Total Heat Rejection 10.21
- Types of Condenser 10.22

10.6 WATER-COOLED CONDENSERS 10.22
- Types of Water-Cooled Condenser 10.22
- Shell-and-Tube Condensers 10.22
- Heat Transfer in Shell-and-Tube Condensers 10.24
- Parameters That Influence the Performance of Shell-and-Tube Condensers 10.25
- Part-Load Operation 10.26

10.7 AIR-COOLED CONDENSERS 10.26
- Construction of Air-Cooled Condenser 10.26
- Heat-Transfer Processes and Temperature Curves 10.26
- Cooling Air Temperature Rise and Volume Flow 10.28
- Condenser Temperature Difference 10.28
- Condensing Temperature, Subcooling, and Oil Effect 10.29
- Clearance, Warm Air Recirculation, and Dirt Clogging 10.29
- Low Ambient Air Control 10.29
- Selection of Air-Cooled Condenser 10.30

10.8 EVAPORATIVE CONDENSERS 10.30
- Condensation Process 10.30
- Heat Transfer in Evaporative Condensation 10.30
- Water Spraying 10.32
- Cooling Air 10.32
- Site Location and Low Ambient Air Control 10.32
- Selection and Installation 10.33

10.9 COOLING TOWERS 10.34
- Counterflow Induced-Draft Cooling Towers 10.34
- Crossflow Induced-Draft Cooling Towers 10.34
- Counterflow Forced-Draft Cooling Towers 10.35

10.10 THERMAL ANALYSIS OF COOLING TOWERS 10.36
- Technological Terms 10.36
- Tower Coefficient 10.36
- Heat- and Mass-Transfer Process for Counterflow Cooling Tower 10.37
- Tower Capacity, Size, and Coefficient 10.37

10.11 COOLING TOWER PERFORMANCE 10.40
- Factors That Affect Cooling Tower Performance 10.40
- Range and Water Circulating Rate 10.40
- Tower Coefficient and Water-Air Ratio 10.41
- Approach 10.41
- Outdoor Wet-Bulb Temperature 10.41
- Fill Configuration and Water Distribution 10.42
- Construction Materials 10.43

10.12 PART-LOAD OPERATION AND OPTIMUM CONTROL 10.43
- Part-Load Operation 10.43
- Optimum Control 10.43

10.13 OPERATING CONSIDERATIONS FOR COOLING TOWERS 10.46
- Recirculation and Interference 10.46
- Freeze Protection and Fogging 10.46
- Blowdown 10.47
- Legionnaires' Disease 10.47
- Maintenance of Cooling Towers 10.47

10.14 HEAT REJECTION SYSTEMS 10.48
Types of Heat Rejection System 10.48
Comparison between Various Systems 10.48
ASHRAE / IESNA Standard 90.1-1999 10.50
10.15 REFRIGERANT FLOW CONTROL DEVICES 10.51
Thermostatic Expansion Valves 10.51
Liquid Charge in the Sensing Bulb 10.53
Electric Expansion Valves 10.55
Float Valves 10.56
Capillary Tube 10.57
REFERENCES 10.58

10.1 EVAPORATORS

The evaporator is one of the main components of a refrigeration system, in which refrigerant evaporates for the purpose of extracting heat from the surrounding air, chilled water, or other substances. In vapor compression refrigeration systems, the evaporator is also an indirect-contact heat exchanger.

Evaporators can be classified into three categories, depending on the medium or substance to be cooled:

- An air cooler is an evaporator that cools the air directly in a refrigerated space or piece of equipment (such as a packaged unit). Conditioned air is then distributed through air distribution systems. In an air cooler, the refrigerant flows inside the metal tubes or finned tubes while air flows over them.
- In a liquid cooler, chilled water is cooled to a lower temperature and is pumped to remote air-handling units, fan coils, or other terminals for air conditioning or other applications.
- An evaporator can be used to produce ice directly, such as an ice maker in an ice harvester of an ice storage system. These are discussed in Chap. 31.

A liquid cooler is different from a chiller. A liquid cooler is an evaporator, a component of a refrigeration system, whereas a chiller is a refrigeration package to produce chilled water.

The refrigerant feed for air and liquid coolers and ice makers can be classified mainly into the following three types:

- *Dry-expansion or direct-expansion (DX).* In evaporators with dry- or direct-expansion refrigerant feed, liquid refrigerant is fed through an expansion valve and a distributer, flows inside the tubes in a finned coil, the evaporator, and is completely vaporized and superheated to a certain degree before reaching the exit of the evaporator.
- *Flooded refrigerant feed.* In evaporators with flooded refrigerant feeds, liquid refrigerant is fed through a throttling device and vaporizes outside the tubes within a shell. The refrigerant-side surface area is always wetted by the liquid refrigerant, which results in a higher surface heat-transfer coefficient.
- *Liquid overfeed.* In liquid overfeed evaporators, liquid refrigerant is fed by a mechanical or gas pump and is then overfed to each evaporator. The inner surface in an overfeed evaporator is also wetted by liquid refrigerant.

The various combinations of cooler types and refrigerant feeds make up the following evaporator groups:

- DX cooler, which is a DX coil, acts as an air cooler with direct-expansion refrigerant feed
- Flooded shell-and-tube liquid cooler, or simply flooded liquid cooler

- Shell-and-tube liquid cooler with direct-expansion refrigerant feed, or simply direct-expansion liquid cooler
- Liquid overfeed cooler
- Direct-expansion ice maker

Among these, DX coils are most widely used. Flooded liquid coolers are widely used in large chillers, and direct-expansion liquid coolers are often used in reciprocating or screw chillers. Liquid overfeed coolers are sometimes adopted for processing air conditioning systems. Ice makers are usually used in thermal storage systems and for industrial applications.

10.2 DX COILS—WET COILS

In a direct-expansion or dry-expansion DX coil for a comfort air conditioning system, the evaporating temperature T_{ev}, °F (°C), of refrigerant HCFC-22, HFC-134a, HFC-404A, HFC-410A, HFC-407A, or HFC-407C inside the coil tubes is usually 37 to 52°F (2.8 to 11.1°C). In such a T_{ev}, the surface temperature of the coil is usually lower than the dew point of the air entering the coil. Condensation occurs on the outside surface of the DX coil, so the coil becomes a wet coil. The air conditioning process of a DX coil is always a cooling and dehumidifying process as well as a heat- and mass-transfer process.

Construction and Installation

Copper tubes and aluminum fins are widely used for DX coils using halocarbon refrigerants. The diameter of the copper tubes is usually $\frac{3}{8}$ to $\frac{5}{8}$ in. (10 to 16 mm). Aluminum fins of a fin spacing often from 12 to 18 fins/in. (1.4 to 2 fins/mm) are typically 0.006 in. (0.15 mm) thick. Corrugated plate fins are often used for additional turbulence and increase in heat transfer. On the inner surface of the copper tubes, microfins typically with a spacing of 60 fins/in. (0.42 fins/mm) and a height of 0.008 in. (0.2 mm) are widely used to enhance the boiling heat transfer. The Eckels and Pate (1991) experiments showed that compared with smooth tubes, copper tubes with inner microfins had a heat-transfer enhancement factor from 1.5 to 2.5 during in-tube evaporation or condensation for refrigerant HFC-134a.

During the analysis and calculation of the performance of a water-cooling, water-heating, or DX coil, A_o/A_i, fin spacing, and tube spacing are the coil construction parameters that determine coil performance. Here A_o indicates the total outside surface area of the coil, and A_i the inner surface area of the coil, ft^2 (m^2). The following coil construction parameters have been used by many manufacturers to optimize the rate of heat transfer, air-side pressure drop, and manufacturing cost:

Longitudinal tube spacing S_L	1.083 in. (27.5 mm)
Transverse tube spacing S_T	1.25 in. (31.8 mm)
Outside nominal diameter of copper tube D_o	0.5 in. (23 mm)
Aluminum fin thickness F_t	0.006 in. (0.15 mm)

Table 10.1 lists the coil construction parameters for smooth fin, based on the data given above. Parameters are for a face area of 1 ft^2 (0.305 m^2) and a coil depth of one row, or 1.083 in. (27.5 mm). If the edge effect, in both longitudinal and transverse directions, is included, then the values of A_o/A_p, A_o/A_i, and F_s may be slightly higher. Here F_s indicates coil core surface area parameter, and it is discussed in detail in Sec. 15.9. For corrugated fins, A_o/A_p, A_o/A_i, and F_s should be multiplied by a corrugated factor $F_{c,a}$ of 1.1 to 1.2.

The ratio of free flow (or minimum flow) area to face area A_{min}/A_a is usually 0.54 to 0.6. For a smaller fin spacing, such as 12 to 15 fins/in. (1.7 to 2 fins/mm), $A_{min}/A_a = 0.58$ to 0.60.

TABLE 10.1 Finned-Tube Coil Construction Parameters

Outside diameter of copper tube D_o	0.528 in.
Inner diameter of copper tube D_i	0.496 in.
Aluminum fin thickness F_t	0.006 in.
Longitudinal tube spacing	1.083 in.
Transverse tube spacing	1.25 in.

Fin spacing S_f						
Fins/in.	in.	A_o/A_p[†]	A_o/A_i	A_f/A_o[‡]	F_s	S_f/F_t
8	0.125	7.85	7.95	0.873	9.91	20.8
10	0.100	9.68	9.68	0.896	12.07	16.7
12	0.0833	11.54	11.40	0.913	14.21	13.9
14	0.0714	13.46	13.17	0.925	16.37	11.9
15	0.0667	14.47	14.03	0.928	17.48	11.1

[†]A_p is area of the primary surfaces (outside surface of the copper tubes).
[‡]A_f is area of fins.
Note: For corrugated plate fins, A_o/A_p, A_o/A_i, and F_s should be multiplied by a factor of 1.10 to 1.20.

For even coolant distribution and to reduce pressure drop, flow paths of refrigerant in a DX coil are always divided into a number of refrigerant circuits, according to the finned width of the coil. Refrigerant is usually supplied through a thermostatic expansion valve. It then flows through a distributor, which distributes the refrigerant evenly, as shown in Fig. 10.1. Venturi-type distributors are usually used. The distributor is connected to various circuits through copper distributor tubes of diameter typically between $^1/_4$ and $^5/_{16}$ in. (6.5 and 16 mm). Equal length of distributor tube and approximately equal circuit lengths ensure even distribution of refrigerant.

To create a counterflow arrangement between air and refrigerant, the suction header should be installed nearest to the air entering the side of the coil.

For a DX coil, copper finned tubes should be reasonably level to ensure proper air venting and condensate drainage. When two or more DX coils are banked vertically, there must be an individual drain pipe and drain pan for each coil.

Two-Region Model

Refrigerant enters the DX coil at a liquid-vapor two-phase state. It is cooled to evaporating temperature T_{ev} and gradually evaporated along the flow paths until it is completely vaporized. Because of the pressure drop of the refrigerant as it flows through the coil, the evaporating temperature T_{ev} gradually drops along the refrigerant circuit as the evaporating pressure decreases, as shown in Fig. 10.1.

For simplification, the flow path of the refrigerant and its corresponding outer surface in a DX coil is divided into two regions.

Two-Phase Region. In the two-phase region, the quality of the refrigerant $x_r < 1$. The pressure drop of the liquid refrigerant is negligible, and liquid refrigerant is evaporated at a uniform temperature.

In the two-phase region, the boiling heat-transfer coefficient h_{boil} of refrigerant HCFC-22, Btu/h·ft²·°F (W/m²·°C), inside a DX coil is mainly a function of the mass flow rate and quality of the refrigerant. Figure 10.2 shows the boiling heat-transfer coefficient of HCFC-22 in a tube with an outside diameter of 0.665 in. (17 mm) at 40°F (4.4°C) when the quality of refrigerant is varied.

Schlager et al. (1989) found that naphthenic mineral oil with a viscosity of 150 SSU enhances the surface heat-transfer coefficient of refrigerant HCFC-22 inside a microfin tube up to 25 percent

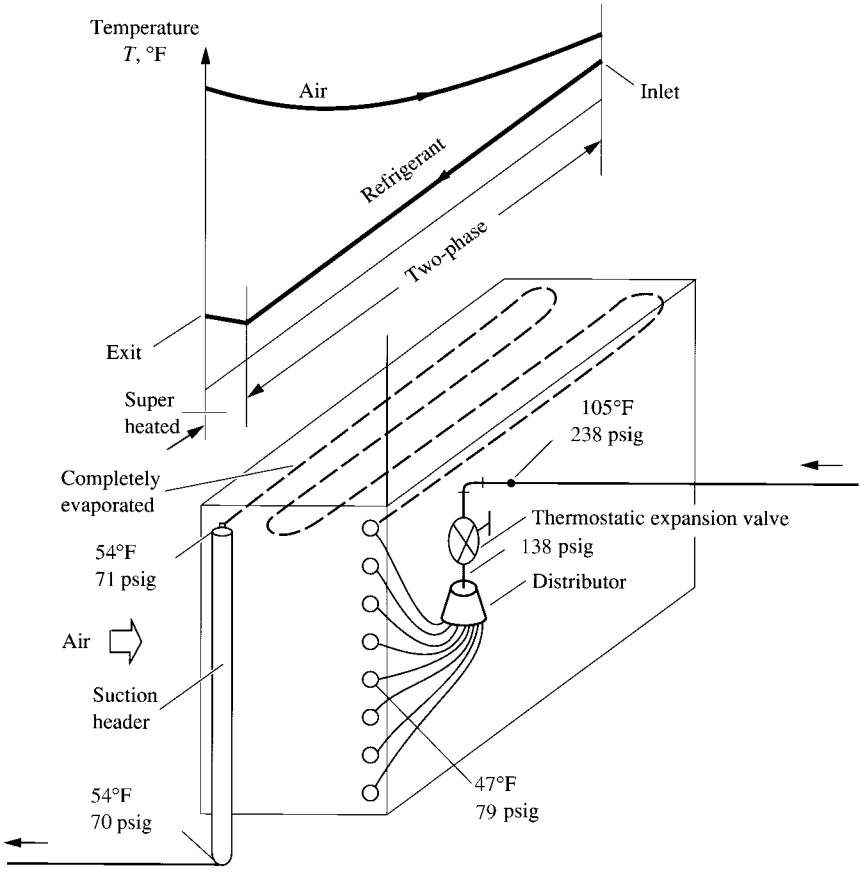


FIGURE 10.1 Temperature and pressure drop across a typical DX coil using HCFC-22 as refrigerant. (All pressures are measured at gauge pressure.)

at an oil concentration of 3 percent. For 300-SSU oil, the surface heat-transfer coefficient inside the microfin tube increases slightly and then decreases about 20 percent when the oil concentration increases from 0 to 3 percent.

Superheated Region. In the superheated region, the quality of refrigerant $x_r = 1$. The refrigerant-side heat-transfer coefficient is low. Because of the relatively high outer surface temperature caused by the superheated refrigerant, the outer surface of the DX coil corresponding to the superheated region may be dry. The superheated region is small. For simplification, the rate of heat transfer (including both sensible and latent heat) of the whole DX coil is often multiplied by a degrading factor F_{super} to account for the existence of the superheated region.

Simultaneous Heat and Mass Transfer

In a DX coil, the driving potentials for the simultaneous heat and mass transfer during cooling and dehumidifying are the enthalpy difference between the ambient air and the saturated air film at the

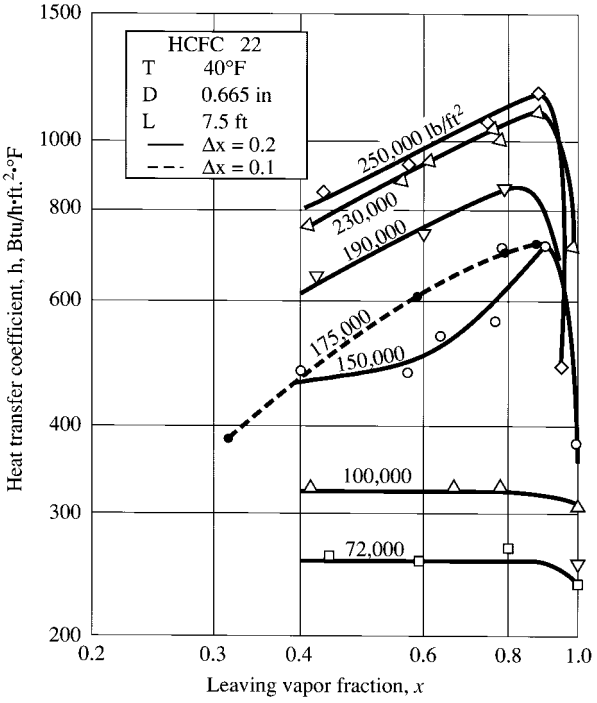


FIGURE 10.2 Boiling heat-transfer coefficient of HCFC-22. (*Source: ASHRAE Handbook 1981, Fundamentals. Reprinted with permission.*)

interface of condensate and air (in Btu/lb or J/kg) and the temperature difference between the air and the evaporating refrigerant inside the tubes.

This heat and mass transfer $Q_{c,\text{wet}}$, Btu/h (W), is more conveniently calculated as

$$Q_{c,\,\text{wet}} = \varepsilon_{\text{wet}}\, 60 \dot{V}_a \rho_a (h_{ae} - h_{s,\,r}) \tag{10.1}$$

where h_{ae} = enthalpy of entering air, Btu/lb (J/kg)
$h_{s,r}$ = enthalpy of saturated air film at coil surface corresponding to evaporating temperature of refrigerant inside tubes, Btu/lb (J/kg)
$\dot{V}_a$ = air volume flow rate, cfm [m^3/(60 s)]
ρ_a = air density, lb/ft^3 (kg/m^3)

DX Coil Effectiveness

The *effectiveness* of a heat exchanger ε is defined as the ratio of the actual rate of heat transfer between the hot and cold fluids to the maximum possible rate of heat transfer. Because the heat energy of the condensate is small compared to that of air and water streams, it can be ignored. DX coil effectiveness ε_{wet} can be calculated as

$$\varepsilon_{\text{wet}} = \frac{h_{ae} - h_{al}}{h_{ae} - h_{s,\,r}}$$

or

$$h_{al} = h_{ae} - \varepsilon(h_{ae} - h_{s,r}) \tag{10.2}$$

where h_{al} = enthalpy of air leaving the coil, Btu/lb (kJ/kg). After h_{al} is determined, the dry-bulb and wet-bulb temperatures of the leaving air can be found from the cooling and dehumidifying curve of the psychrometric chart with the same entering air condition.

For convenience, the refrigerant in a DX coil is assumed to evaporate at a constant temperature, so $C_{\min}/C_{\max} = 0$. Here $C_{\min}$ and $C_{\max}$ indicate the smaller and greater value of heat capacities between air and water, Btu/h·°F (W/°C). The wet coil effectiveness is calculated as

$$\varepsilon_{\text{wet}} = 1 - \exp(\text{-NTU}) \tag{10.3}$$

The number of transfer units NTU for a DX coil can also be calculated as

$$\text{NTU} = \frac{U_o A_o}{C_a} \tag{10.4}$$

where U_o = overall heat-transfer coefficient based on outside surface of coil, Btu/h·ft²·°F (W/m²·°C)
A_o = total outside surface area of coil, ft² (m²)
C_a = heat capacity rate of moist air, Btu/h·°F (W/°C)

and

$$C_a = 60\,\dot{V}_a \rho_a c_{pa} \tag{10.5}$$

where c_{pa} = specific heat of moist air, Btu/h·°F (J/kg·°C)
ρ_a = density of moist air, lb/ft³ (kg/m³)

Also

$$U_o A_o = \frac{1}{1/(\eta_s h_o A_o) + 1/(h_i A_i)} \tag{10.6}$$

where h_o, h_i = heat-transfer coefficient of outer surface of coil (including fins and tubes) and inner surface of tubes, Btu/h·ft²·°F (W/m²·°C)
A_i = inner surface area of coil, ft² (m²)

In Eq. (10.6), η_s represents the finned surface efficiency and is given as

$$\eta_s = \frac{1 - A_f}{A_o(1 - \eta_f)} \tag{10.7}$$

where A_f = area of fins, ft² (m²). The fin efficiency η_f can be calculated as

$$\eta_f = \frac{\text{actual heat transfer}}{\text{heat transfer when fins are at base temperature}} \tag{10.8}$$

Face Velocity and Air-Side Pressure Drop of Wet Coil

Selection of face velocity for wet coils must consider the carryover of the condensate water droplets, the influence of coil face velocity on heat-transfer coefficients, the pressure drop of the coil, and the size of the coil. If air velocity distribution is even over the entire surface of the DX coil, the carryover is usually not significant at face velocities lower than 500 fpm (2.5 m/s) for

smooth plate fins or lower than 550 fpm (2.75 m/s) for corrugated fins [to prevent wet surfaces formed in an air distribution system that nourishes the mold growth in fiberglass duct liners, a DX coil face velocity of 500 fpm (2.5 m/s) or less is preferable]. When the supply fan is located upstream from the DX coil (a blow-through fan and coil arrangement), there may be areas where the face velocity is far higher than in the rest of the coil.

For wet DX coils, a rough estimate of air-side pressure drop for 15 fins/in. (1.7 fins/mm) corrugated fins is as follows:

	Fin configuration	
Face velocity, fpm (m/s)	Simple	Corrugated
500 fpm (2.5 m/s)	0.15 in. WC (37 Pa)/row	0.23 in. WC (57 Pa)/row
600 fpm (3 m/s)	0.25 in. WC (62 Pa)/row	0.35 in. WC (87 Pa)/row

The air-side pressure drop for a three-row 15-fins/in.(1.7 fins/mm) corrugated fin coil from one manufacturer may be 50 percent higher than that from another manufacturer, even at the same operating conditions and same unit cooling capacity. It is important to carefully compare the coil performances of different manufacturers.

The lower limit of coil face velocity at design load depends on initial and operating costs. For DX coils in VAV systems, reduction of face velocity in part-load operation should also be considered. Face velocities between 350 and 450 fpm (1.75 and 2.25 m/s) may be justified in many cases.

DX Coils at Part-Load Operation

The refrigerant circuits in a DX coil of capacity greater than 15 tons are usually divided into two or more sections, each with its own expansion valve, distributor, and suction header. The refrigerant circuits are controlled in three ways, as shown in Fig. 10.3.

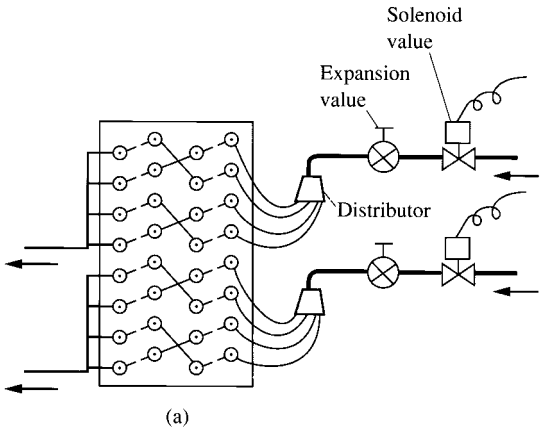
Face Control. The refrigerant circuits are divided into upper and lower sections, as shown in Fig. 10.3*a*. One section can be shut off by deenergizing its solenoid valve during capacity reduction. Face control reduces the temperature of the conditioned air, and requires thorough mixing of conditioned and bypass airstreams downstream from the DX coil.

Row Control. Row control is more effective when applied to a six-row DX coil, as shown in Fig. 10.3*b*. One distributor connects to the refrigerant circuits of two rows; another distributor connects to the remaining four rows. The active refrigerant circuits can then be operated at 33, 66, or 100 percent capacity.

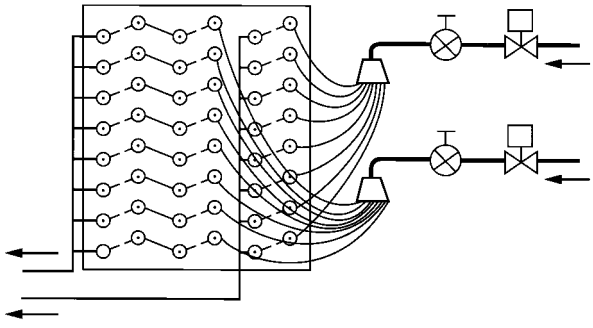
Intertwined Face Control. An intertwined face control (shown in Fig. 10.3*c*) always has a full face active coil even when the refrigerant supply to one of the distributors is cut off by the solenoid valve. When one distributor is inactive, the fin efficiency η_f and the heat capacitance ratio of wet coil C_{wet} are changed, and the capacity of the DX coil is reduced to about 70 percent.

In the selection of a DX refrigeration system, it is important to check the minimum refrigerant temperature in the DX coil during the part-load operation. The minimum saturated temperature of the refrigerant $T_{r,\,min}$, °F (°C), for a DX coil at a face velocity about 500 fpm (2.5 m/s) must not drop below 32°F (0°C). Otherwise, frost will form on the coil surface. Frost formation blocks the air passage and severely reduces the rate of heat transfer. At part-load operation, the evaporating temperature is lowered because less refrigerant is evaporated.

For DX coils using HCFC-22 or other oil-miscible refrigerants, the refrigrant pipe should be designed so that the refrigerant-oil mixture can be returned in the pipe riser at minimum load operation. This is covered in detail in Chap. 11.



(a)



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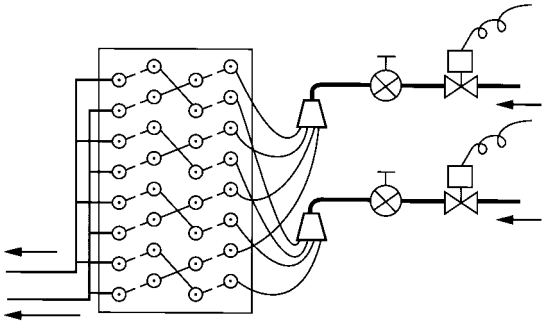


(c)

FIGURE 10.3 Control of DX coils at part-load operation. (*a*) Face control; (*b*) row control; (*c*) intertwined face control.

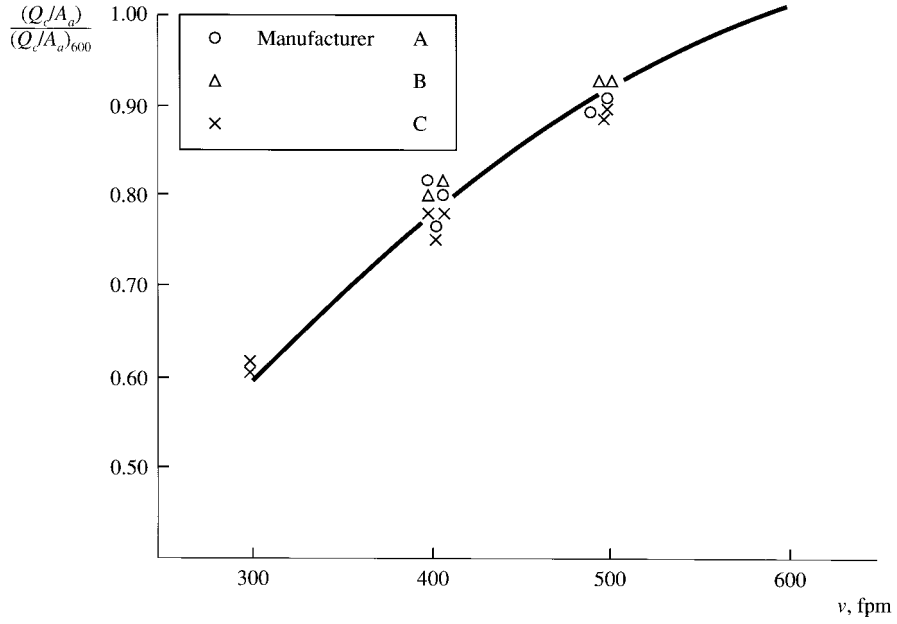


FIGURE 10.4 Variation of cooling capacity versus face velocity.

When a DX coil is installed in a variable-air-volume system, the air volume flow rate and face velocity are decreased during part-load operation. Figure 10.4 shows the relationship between the cooling capacity of the DX coil at a specific face velocity and the cooling capacity at a face velocity of 600 fpm (3 m/s), denoted by $(Q_c/A_a)/(Q_c/A_a)_{600}$. Here, Q_c indicates the coil capacity and A_a the face area of the coil. The curve is plotted according to data from the catalogs of three manufacturers at an air entering condition of 80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb, and 45°F (7.2°C) suction temperature, or 45°F (7.2°C) entering chilled water temperature and 55°F (12.8°C) leaving. When the face velocity drops from 600 to 400 fpm (3 to 2 m/s), $(Q_c/A_a)/(Q_c/A_a)_{600}$ drops from 1.0 to about 0.8.

For a VAV packaged system using a DX coil, it is important to have effective multistep capacity unloading and refrigeration circuit cutoff controls during part-load operation to maintain T_{ev} above 32°F (0°C). This is discussed in Chap. 11.

Selection of DX Coils

The cooling capacity (refrigeration capacity) of a DX coil is actually the cooling capacity of the corresponding DX refrigeration system as well as the cooling capacity of the packaged air conditioning system. For comfort air conditioning, today a fixed pattern of DX coil, including number of rows, fins per inch, and tubing spacing, is adopted for each model and size of packaged unit. The trend is to use a denser fin spacing, such as 14 to 17 fins/in. (1.5 to 1.8 fins/mm). For the cooling and dehumidifying of the mixture of outdoor and recirculating air in air conditioning, a two-, three-, or four-row DX coil is often used. For a 100 percent outdoor air makeup unit, a four- or six-row DX coil is often used. Check also the coil face velocity so that it does not exceed 550 fpm (2.75 m/s).

The selection of the equipment for the DX refrigeration system including the selection of the DX coil is discussed in later chapters.

Example 10.1. A DX coil in a packaged conditioning system has a supply volume flow rate of 5500 cfm (2595 L/s). Air enters the coil at a dry-bulb temperature of 80°F (26.7°C) and a wet-bulb temperature of 67°F (19.4°C). Calculate the total cooling capacity of this DX coil at full load and the conditions of the air leaving the coil, using the following data:

- A three-row, 15-fins/in. (1.7-fins/mm) coil with the same coil construction parameters shown in Table 10.1
- Boiling heat-transfer coefficient for HCFC-22 inside the copper tubes of 700 Btu/h·ft²·°F (3975 W/m²·°C) at $T_{ev} = 45°F$ (7.2°C)
- Surface effectiveness for wet fins $\eta_s = 0.76$
- Heat-transfer coefficient of the outer surface of the coil (corrugated fin) $h_o = 12.90$ Btu/h·ft²·°F (73.25 W/m²·°C)

Solution. For a DX coil using corrugated plate fins, select a face velocity $v_a = 550$ fpm (2.75 m/s) to provide an efficient heat transfer and a reasonable air-side pressure drop, and avoid condensate carryover. Face area A_a for the DX coil can then be calculated as

$$A_a = \frac{\dot{V}_s}{v_a} = \frac{5500}{550} = 10 \text{ ft}^2$$

For a 15-fins/in. (1.7-fins/mm) DX coil, from Table 10.1, the coil core surface area parameter $F_s = 17.48$. At full-load operation, for a face area of 10 ft², and a multiplying factor of 1.15 for the corrugated surface, the outer surface area of this DX coil is

$$A_o = F_s N_r A_a = 17.48 \times 1.15 \times 3 \times 10 = 603 \text{ ft}^2 \text{ (56 m}^2\text{)}$$

For corrugated fins, $A_o/A_i = 14.03 \times 1.15$. The inner area is therefore

$$A_i = \frac{603}{14.03 \times 1.15} = 37.4 \text{ ft}^2 \text{ (3.5 m}^2\text{)}$$

As given, for a wet surface $\eta_s = 0.76$. Then, from Eq. (10.6),

$$U_o A_o = \frac{1}{1/(\eta_s h_o A_o) + 1/(h_r A_i)}$$

$$= \frac{1}{1/(0.76 \times 12.90 \times 603) + 1/(700 \times 37.4)} = 4995 \text{ Btu/h} \cdot °F$$

The heat capacity rate for conditioned air is calculated as

$$C_a = 60 \dot{V}_a \rho_a c_{pa} = 60 \times 5500 \times 0.075 \times 0.243 = 6014 \text{ Btu/h} \cdot °F$$

The number of transfer units is

$$\text{NTU} = \frac{U_o A_o}{C_a} = \frac{4995}{6014} = 0.831$$

From Eq. (10.3), the effectiveness for the DX coil can be calculated as

$$\varepsilon_{\text{wet}} = 1 - \exp(\text{-NTU}) = 1 - e^{-0.831} = 1 - 0.436 = 0.564$$

Considering the influence of the superheated region, effectiveness of the DX coil should be multiplied by a degrading factor $F_{\text{super}} = 0.95$. Then

$$\varepsilon_{\text{DX}} = 0.0564(0.95) = 0.536$$

From the psychrometric chart, $h_{ae} = 31.6$ Btu/lb and the enthalpy of saturated air film at evaporating temperature $T_{ev} = 45°F$ is 17.65 Btu/lb. Then the total cooling capacity of the DX coil is

$$Q_c = 60\dot{V}_s\rho_a\varepsilon(h_{ae} - h_{s,r})$$
$$= 60 \times 5500 \times 0.075 \times 0.536(31.6 - 17.65) = 185{,}060 \text{ Btu/h (54,223 W)}$$

The cooling capacity is

$$\frac{Q_c}{A_a} = \frac{185{,}060}{10 \times 1000} = 18.5 \text{ MBtu/h} \cdot \text{ft}^2 (58.36 \text{ kW/m}^2)$$

The enthalpy of the conditioned air leaving the DX coil h_{al} is

$$h_{al} = h_{ae} - \Delta h = h_{ae} - \varepsilon(h_{ae} - h_{s,r}) = 31.6 - 0.536(31.6 - 17.65) = 24.1 \text{ Btu/lb (56 kJ/kg)}$$

Draw a cooling and dehumidifying curve from the air entering condition, 80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb temperature on the psychrometric chart. Determine the air leaving condition on the curve since $h_{al} = 24.1$ Btu/lb (56 kJ/kg). From psychrometric chart, air leaves at 57.6°F (14.2°C) dry-bulb and 56.2°F (13.4°C) wet-bulb temperature with a relative humidity of 93 percent.

10.3 FLOODED LIQUID COOLER

Construction

Most medium-size and large liquid coolers are shell-and-tube flooded liquid coolers. In a flooded liquid cooler, several straight tubes are aligned in a parallel staggered arrangement, usually held in place at both ends by tube sheets, as shown in Fig. 10.5*a*. Chilled water circulates inside the tubes, which are submerged in a refrigerant-filled shell.

Liquid-vapor refrigerant, usually at a quality x around 0.15 in air conditioning applications, is fed into the bottom of the shell. It is evenly distributed over the entire length of the tubes. As the refrigerant boils and bubbles rise, the upper part becomes increasingly bubbly. Vapor refrigerant is discharged from the opening at the top of the cooler. A dropout area, or eliminator, is sometimes installed to separate the liquid refrigerant from the vapor. The amount of refrigerant fed to the flooded liquid cooler is controlled by a low-pressure-side float valve, or a multiple-orifice throttling device, which is discussed in Chaps. 11 and 13.

When halocarbons are used as the refrigerants, copper tubes are always used because they provide higher thermal conductivity and do not react with halocarbons. Tube diameters of $^3/_4$ and 1 in. (19 and 25 mm) are often used, such as either internally enhanced copper tubing of 1-in.- (25-mm-) diameter or smooth bore of $^3/_4$-in. (19-mm) diameter, and the number of tubes inside the shell varies from 50 to several thousand. Integral fins are extruded on the outer surface of the tubes to increase the outer surface area. Other tube geometry on the outside surface is also used to enhance the boiling heat-transfer coefficient, especially to coupling of a high-voltage and low current electric field with the fluid field called electrohydrodynamics, and the boiling and condensing heat-transfer coefficient may be increased in excess of tenfold. Fin spacing ranges from 19 to 40 fins/in. (0.6 to 1.3 fins/mm), typically 26 fins/in. (1 fin/mm) at a height from 0.06 in. (1.5 mm). The outer-to-inner surface ratio of integral finned tubes ranges from 2.5 to 3.5. Spiral grooves or other enhancements may be added to the inner surface of the tubes to increase the inner surface heat-transfer coefficient. By increasing the turbulence, dirt and suspended solids are prevented from settling on the inner surface during operation.

Possible water flow arrangements in flooded shell-and-tube liquid coolers are shown in Fig. 10.5*b*. A one-pass arrangement has the highest chilled water flow rate, two-pass has a lower

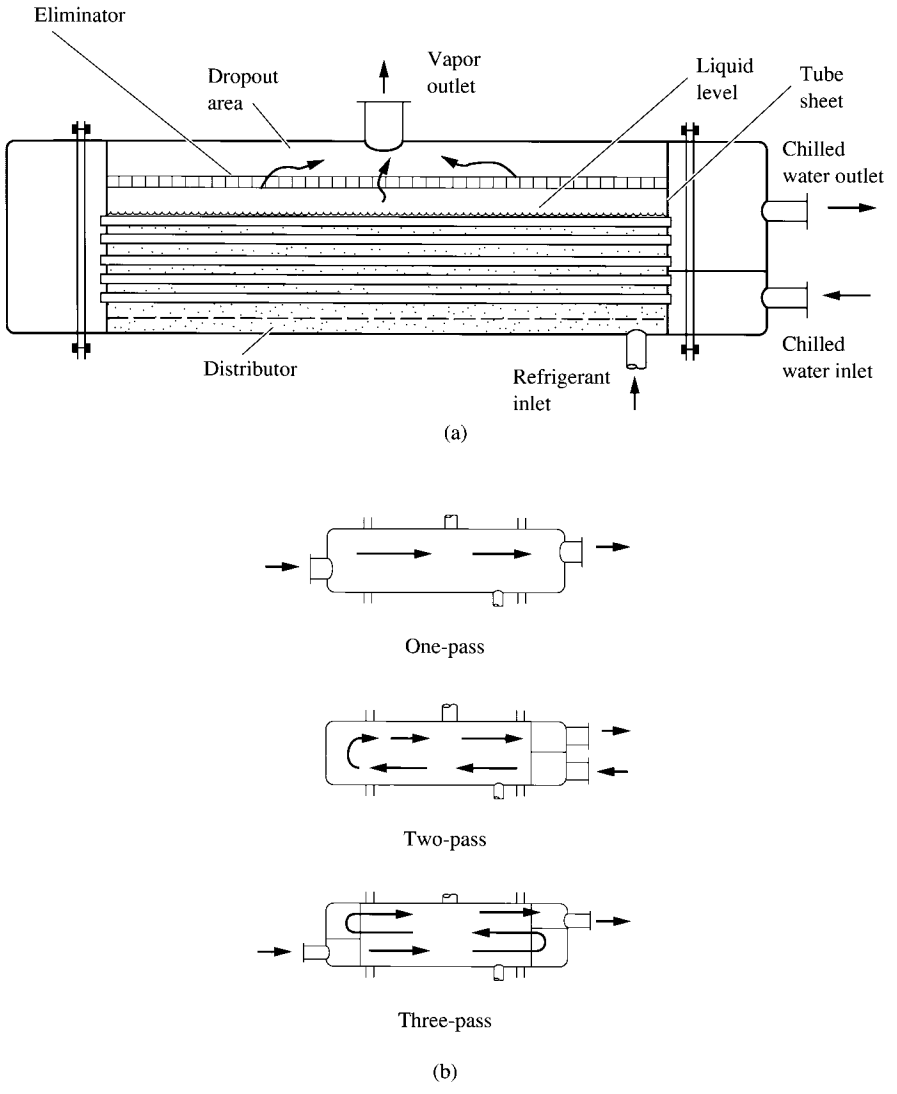


FIGURE 10.5 Flooded shell-and-tube liquid cooler: (*a*) schematic diagram; (*b*) passages of water flow.

rate, and three-pass has the lowest chilled water flow rate. The two-pass water flow arrangement, with the water inlet and outlet on the same side, is the standard arrangement. Performance analyses showed that there is no significant difference between a two-pass arrangement with the water inlet and outlet located side by side and one with the inlet at the bottom and outlet at the top. Water velocity inside the copper tubes is usually between 4 and 12 ft/s (1.2 and 3.6 m/s); water velocity exceeding 12 ft/s (3.6 m/s) may cause erosion. Water-side pressure drop is usually maintained at or below 10 psi (70 kPa) to optimize pump energy consumption. Because of the greater flow passage for refrigerant within the shell, the pressure loss on the refrigerant side is far lower than the pressure loss in DX coils.

Flooded liquid coolers provide a large surface area and higher heat-transfer coefficient, need minimal space and headroom, and are easily maintained. They are most widely used in large refrigeration plants.

Heat Transfer in a Flooded Liquid Cooler

In flooded liquid coolers, the refrigerant is assumed to evaporate at a uniform temperature. The rate of heat transfer between the chilled water and the refrigerant Q, Btu/h (W); U_o and h_i, Btu/h·ft²·°F (W/m²·°C); ε; and NTU can be calculated from Eqs. (7.1), (9.25), (10.1) through (10.4), and (10.6) and summarized as follows:

$$\begin{aligned} Q &= 60\dot{m}_w c_{pw}\,(T_{ee} - T_{el}) \\ &= 500\dot{V}_{\text{gal}}(T_{ee} - T_{el}) \\ &= A_o\,U_o\,\Delta T_m \\ &= \dot{m}_r(h_{rl} - h_{re}) \end{aligned}$$

$$U_o = \frac{1}{1/h_o + L_t A_o/(A_i k_t) + A_o R_f/A_i + A_o/(A_i h_i)} \tag{10.9}$$

$$\frac{h_i D_i}{k_w} = 0.023\,\text{Re}_D^{0.8}\,\text{Pr}^n$$

$$\varepsilon = 1 - \exp(\text{-NTU})$$

$$\text{NTU} = \frac{U_o A_o}{c_{pw}}$$

$$Q = 60\dot{m}_w c_{pw}\varepsilon(T_{ee} - T_{\text{ev}})$$

where h_{re}, h_{rl} = enthalpy of refrigerant entering and leaving liquid cooler, Btu/lb (kJ/kg or J/kg)
T_{ee}, T_{el} = chilled water entering and leaving liquid cooler, °F (°C)
ΔT_m = log-mean temperature difference between chilled water and refrigerant, °F (°C)
$\dot{m}_r$ = mass flow rate of refrigerant, lb/min [kg/(60 s)]
h_o = average heat-transfer coefficient of refrigerant-side surface, Btu/h·ft²·°F (W/m²·°C)
h_i = inner surface heat-transfer coefficient, Btu/h·ft²·°F (W/m²·°C)
D_i = inner diameter of tube, ft (m)
L_t = thickness of tube, ft (m)
k_w = thermal conductivity of water, Btu/h·ft·°F (W/m·°C)
k_t = thermal conductivity of metal tube, Btu/h·ft·°F (W/m·°C)
T_{ev} = evaporating temperature, °F (°C)

For the analysis of heat transfer in a flooded liquid cooler, both the log-mean temperature difference and the NTU method can be used. However, NTU method is preferred because it clearly describes the effectiveness and characteristics of the liquid cooler.

Fouling Factor

Fouling factor R_f, h·ft²·°F/Btu (m²·°C/W), is the additional resistance caused by the dirty film of scale, rust, silt, or other deposits on the surface of the tube. The fouling factor may be defined as

$$R_f = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}} \tag{10.10}$$

where U_{dirty}, U_{clean} = overall heat-transfer coefficient through tube wall with dirty or clean surface respectively, Btu/h·ft²·°F (W/m²·°C).

In 1982, *ASHRAE Journal* published research results on the fouling of heat-transfer surfaces such as evaporators and condensers in air conditioning. The study showed that with a certain water treatment and in the absence of biological growth and suspended particulates, long-term fouling did not exceed 0.0002 h·ft²·°F/Btu (0.000035 m²·°C/W) and short-term fouling did not exceed 0.0001 h·ft²·°F/Btu (0.000018 m²·°C/W).

The Air-Conditioning and Refrigeration Institute (ARI) Standard 550-88 specifies the following:

Field fouling allowance	0.00025 h·ft²·°F/Btu (0.000044 m²·°C/W)
ARI Rating Standard (new evaporators and condensers)	0

The ARI Standard recommends specifying 0.00025 h·ft²·°F/Btu (0.000044 m²·°C/W) total fouling for closed-loop liquid coolers and for condensers served by well-maintained cooling towers.

Pool Boiling and Forced-Convection Model

If a surface is in contact with a liquid, and if the temperature of the surface is maintained above the saturation temperature of the liquid, boiling may occur. The rate of heat transfer depends mainly on the temperature difference between the surface giving heat and the saturation temperature of the liquid at a specified pressure.

When the surface is submerged below the free surface of the liquid, the boiling process is known as *pool boiling*. When a mixture of liquid and vapor is forced through a tube or a passage and the surface surrounding the mixture is above the saturation temperature of the liquid, forced-convection boiling may occur inside the tube or passage.

When the temperature of a surface that releases heat is a few degrees higher than the temperature of the liquid, molecules within the thin layer of superheated liquid adjacent to the surface tend to break away from the surrounding liquid molecules. They form nuclei, which grow into vapor bubbles. As the number and size of these bubbles increase, the bubbles flow upward to the surface of the liquid and form forced-convection boiling. This process of combined nucleate boiling and forced convection occurs in flooded shell-and-tube liquid coolers.

Chen (1966) developed a forced-convection model that can be applied to flooded shell-and-tube liquid coolers to calculate the boiling heat-transfer coefficient on the outer surface of the tube h_b, Btu/h·ft²·°F (W/m²·°C), as follows:

$$h_b = h_{nb} + h_{tp} = Sh_{nbl} + Fh_{fc} \tag{10.11}$$

where h_{nbl} = single-tube pool-boiling coefficient, Btu/h·ft²·°F (W/m²·°C)
S = suppression factor
h_{fc} = single-phase forced-convection coefficient for liquid phase flowing alone in tube bundle, Btu/h·ft²·°F (W/m²·°C)
F = forced-convection multiplier

The experimental results of Webb et al. (1990) showed that for CFC-11 and copper tubes with integral fins, the forced-convection portion $h_{nb} = Sh_{nbl}$ is 1.6 times larger than the nucleate boiling portion $h_{tp} = Fh_{fc}$ at the first row (bottom row) of the tube bundle and 5.3 times larger at the top row. In tubes with high-flux outer boiling surfaces with porous coating, the capacity of the liquid cooler is 36 percent higher than in tubes with integral fins.

Because the average heat-transfer coefficient of a tube's outer surface h_o, Btu/h·ft²·°F (W/m²·°C), is a function of the temperature difference between the tube wall and the refrigerant $T_s - T_r$, also h_o is a function of the heat flux at the tube's outer surface Q_{ev}/A_o, Btu/h·ft² (W/m²). Here Q_{ev} represents the rate of heat transfer through the tube surfaces of the liquid cooler or the

cooling capacity of a specific cooler, Btu/h (W), and A_o the outer surface area of the tubes, ft^2 (m^2). Heat-transfer coefficient h_o in a flooded liquid cooler can be calculated as

$$h_o = C\left(\frac{Q_{\text{ev}}}{A_o}\right)^n \tag{10.12}$$

Constant C and exponent n are mainly determined by the outer surface conditions of the tubes and the wettability of different refrigerants on tubes.

Parameters That Influence the Performance of Flooded Liquid Cooler

Evaporating Temperature T_{ev}. In comfort air conditioning systems, chilled water usually leaves the liquid cooler at a temperature T_{el} from 42 to 45°F (5.6 to 7.2°C) to provide the desirable air off-coil temperature T_{al}. In ice storage and cold air distribution systems (which are discussed in Chap. 31), brine, a mixture of chilled water and glycol, may leave the liquid cooler at a temperature of 24 to 34°F (−4.4 to 1.1°C).

In a conventional chilled water system, if the value of T_{el} is fixed, the choice of optimum temperature difference $T_{el} - T_{\text{ev}}$ determines the evaporating temperature in the flooded liquid cooler, as shown in Fig. 10.6. A smaller $T_{el} - T_{\text{ev}}$ means a lower log-mean temperature difference ΔT_m between the chilled water and refrigerant. For a specific refrigeration capacity, a smaller ΔT_m needs a larger liquid cooler, a higher T_{ev}, and a lower power input to the compressor. If a larger $T_{el} - T_{\text{ev}}$ is chosen, the liquid cooler can be smaller, but the T_{ev} value will be lower and compressor power input will be higher. A compromise between energy cost and initial investment should be made through life-cycle cost analysis.

Most liquid cooler manufacturers have adapted a value of $T_{el} - T_{\text{ev}}$ between 6 and 10°F (3.3 and 5.6°C). For energy-saving models, $T_{el} - T_{\text{ev}}$ may be as low as 4 to 7°F (2.2 to 3.9°C).

Temperature Difference $T_{ee} - T_{el}$. For each ton (12,000 Btu/h, or 3.516 kW) of cooling capacity of a liquid cooler, the water flow rate $\dot{V}_{\text{gal}}$ (gpm, or L/s) at various temperature differences of chilled water entering and leaving $T_{ee} - T_{el}$ can be calculated from Eq. (10.9) as follows:

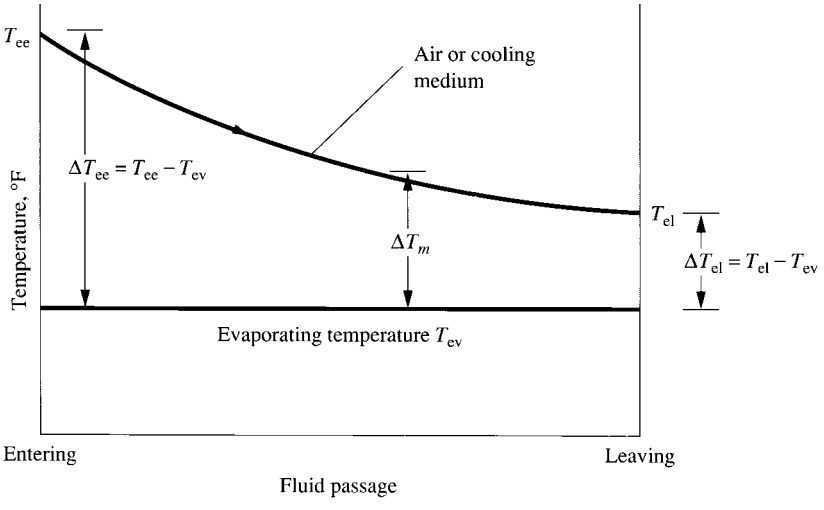


FIGURE 10.6 Heat-transfer process in a flooded shell-and-tube cooler.

$T_{ee} - T_{el}$, °F	$\dot{V}_{gal}$, gpm
8	3
10	2.4
15	1.6
20	1.2
24	1.0

Temperature difference $T_{ee} - T_{el}$ has a direct impact on pump power. In addition, for a given T_{el}, a greater $T_{ee} - T_{el}$ always means a greater log-mean temperature difference ΔT_m and a greater $T_{ee} - T_{ev}$ and, therefore, a greater cooler capacity and higher off-coil temperature T_{al}, as shown in Fig. 10.6. Here the log-mean temperature difference is calculated as

$$\Delta T_m = \frac{T_{ee} - T_{el}}{\ln[(T_{ee} - T_{ev})/(T_{el} - T_{ev})]} \tag{10.13}$$

At full load, the chilled water temperature increase from pump power and piping heat gain is usually negligible. Therefore, temperature difference $T_{ee} - T_{el}$ is often equal to the temperature difference of chilled water leaving and entering the water cooling coils $T_{wl} - T_{we}$ at design full load, and is closely related to the conditioned air temperature leaving the coil T_{al}. Many liquid coolers use a $T_{ee} - T_{el}$ between 12 and 24°F (6.7 and 13.3°C). For $T_{ee} - T_{el} = 20$°F, the water flow rate $\dot{V}_{gal} = 1.2$ gpm/ton (0.0215 L/s·kW) refrigeration at full load.

The minimum chilled water temperature leaving the liquid cooler should be 38°F (3.3°C), and the mass flow rate of chilled water flowing through the flooded liquid cooler should be approximately constant to prevent freezing. If the chilled water temperature is to be lower than 38°F (3.3°C), brine should be used, as in ice storage systems.

Oil Effect. In flooded liquid coolers, vapor velocity above the tube bundle is usually insufficient to return the oil to the compressor. Oil concentration in the refrigerant tends to increase during operation. If oil concentration rises above 5 percent, it may degrade the rate of heat transfer. Generally, oil-rich liquid is taken and returned to the compressor continuously.

Cooling Capacity. The cooling capacity of a flooded liquid cooler also represents the capacity of the refrigeration system. From Eq. (10.10), for a liquid cooler with specified T_{el}, T_{ee}, and T_{ev}, the cooling capacity Q_{ev}, Btu/h, depends on the following factors:

- Tubes' outer surface area A_o and number of transfer units NTU
- Heat flux at the tubes' outer surface Q_{ev}/A_o
- Heat-transfer coefficients h_o and h_i and overall heat-transfer coefficient U_o
- Heat capacity rate of chilled water

Recent developments in flooded shell-and-tube liquid coolers have emphasized high-flux and enhanced boiling coefficient tubes to increase cooling capacity.

Part-Load Operation of Flooded Liquid Cooler

When a flooded liquid cooler in a comfort air conditioning system is at part-load operation, the outer surface area A_o is the same as in full-load operation, so the reduction of its cooling capacity is mainly caused by the following:

1. A decrease in heat flux and, therefore, a lower outer surface boiling heat-transfer coefficient h_o
2. A smaller difference in $T_{ee} - T_{ev}$ caused by the drop in T_{ee} and increase in T_{ev}

3. A smaller log-mean temperature difference resulting from a higher T_{ev}
4. A decrease of effectiveness ε

10.4 DIRECT-EXPANSION LIQUID COOLER AND LIQUID OVERFEED COOLER

Direct-Expansion Liquid Cooler

In a direct-expansion liquid cooler (shown in Fig. 10.7), liquid refrigerant evaporates inside the copper tubes while chilled water fills the shell. An expansion valve and sometimes a distributor are used for each group of refrigerant circuits connected to the same suction header and compressor.

In a direct-expansion liquid cooler, various inner surface configurations and enhancements are used to increase the boiling heat transfer. To provide optimum velocity and a higher rate of heat transfer on the water side of the liquid cooler, baffle plates are used to guide the water flow in the shell in multipass arrangements.

Direct-expansion liquid coolers are usually used for refrigerating systems equipped with multiple compressors. In a direct-expansion liquid cooler, refrigerant circuits may be connected to a single header and compressor or to two separate headers and multiple compressors.

Liquid Overfeed Cooler

A liquid overfeed cooler can be an air cooler or a liquid cooler. Liquid refrigerant is fed to multiple evaporators by a mechanical pump or by a gas pump using the high-pressure gas discharged from the compressor. Liquid refrigerant is fed at a mass flow rate 2 to 6 times greater than the actual evaporating rate, which causes liquid recirculation, as shown in Fig. 10.8. Liquid recirculation pro-

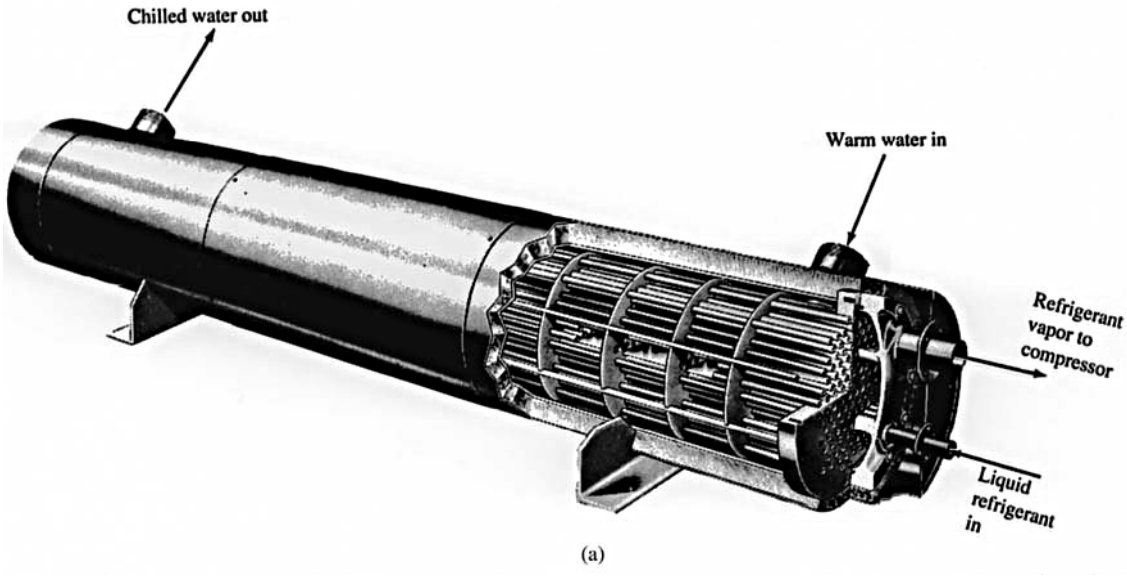


FIGURE 10.7 Direct-expansion liquid cooler: (*a*) cut-away view of a typical DX liquid cooler. (*Source*: *Trane Company. Reprinted with permission.*)

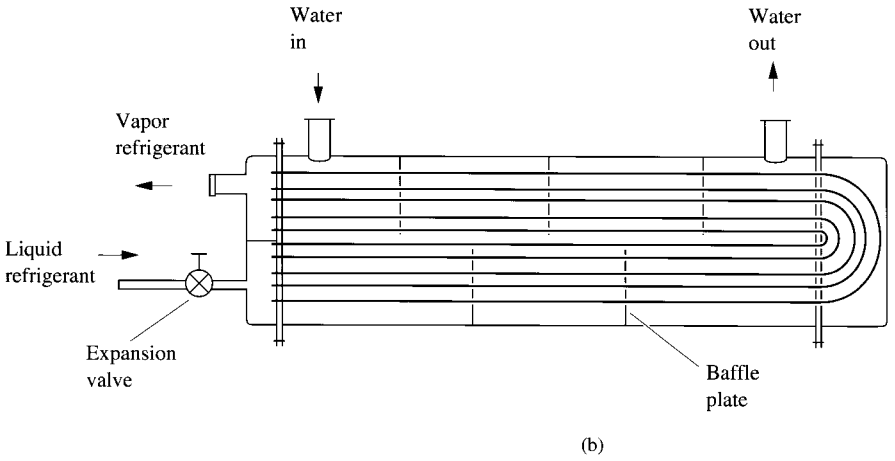


FIGURE 10.7 (*Continued*) (*b*) Schematic diagram.

vides a sufficient volume of liquid in the tubes and ensures that the inner surface of the tubes is fully wetted throughout its length.

Compared with DX coils, liquid overfeed air coolers have the following advantages:

- They have higher heat-transfer coefficient on the refrigerant side, eliminating the decline of the heat-transfer coefficient at the dryout area when the quality of the refrigerant is raised to 0.85 to 0.90 in a DX coil.
- They allow nonuniform loading between refrigerant circuits or between evaporators, so that none are starved, with insufficient refrigerant supply.
- Air-refrigerant temperature difference is lower, which is especially beneficial for low-temperature systems.
- Evaporators can be conveniently defrosted.

Their main disadvantage is their higher initial cost.

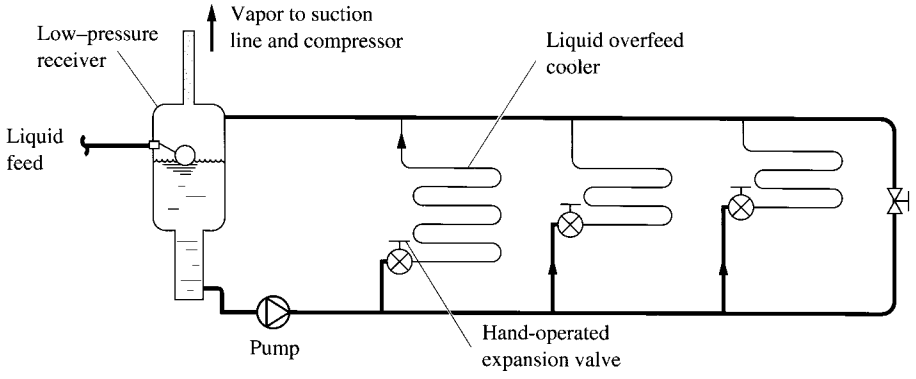


FIGURE 10.8 Liquid overfeed cooler using a mechanical pump.

During the design of a liquid overfeed cooler, the following choices should be carefully considered:

Mechanical Pump or Gas Pump. Gas pumps generally require more energy than mechanical pumps. According to Cole's (1986) analysis, based on a typical plant of 275 tons (967 kW) of cooling capacity and an evaporating temperature of 20°F (−6.7°C), the energy requirement of a gas pumping system is about 7 percent of the total plant power, and that of mechanical pump is about 2 percent. However, the difference in energy consumption is a small percentage of the total plant power, so in many circumstances, gas pumps are still used.

Among mechanical pump systems, centrifugal pumps are most popular. Rotary and gear pumps may be suitable in circumstances where cavitation is critical.

Down-Feed or Up-Feed. In down-feed circuits, refrigerant is fed to the highest tube of the evaporating coil. This arrangement improves oil return and eliminates static head. Up-feed circuits simplify liquid distribution. The relative locations of the evaporators and receivers become less important, and the system design and layout may be simplified.

Counterflow or Parallel Flow. Because of its lack of superheat, as in DX coils, a parallel flow arrangement between air and refrigerant in a liquid overfeed air cooler has a small performance advantage over the counterflow arrangement. Also, frosting occurs evenly in parallel flow, whereas frosting is heavier in the first and second rows of the finned-tube coil in a counterflow arrangement.

Circulating Rate. Circulating rate or number n is defined as

$$n = \frac{\dot{m}_{cir}}{\dot{m}_{ev}} \tag{10.14}$$

where $\dot{m}_{cir}$ and $\dot{m}_{ev}$ = mass flow rate of refrigerant, circulated and evaporated, respectively, lb/min [kg/(60 s)]. According to *ASHRAE Handbook 1998, Refrigeration,* the minimum recommended circulating rate is as follows:

Ammonia (R-717)	
Down-feed (large-diameter tubes)	6 to 7
Up-feed (small-diameter tubes)	2 to 4
HCFC-22-Up-feed	3
HFC-134a	2

Liquid overfeed air coolers are widely used in low-temperature refrigeration plants for food storage and processing. Recently, overfeed liquid coolers have also been used in ice storage systems.

10.5 CONDENSATION AND CONDENSERS

Condensation Process

When saturated vapor comes into contact with a surface having a temperature below the saturation temperature, condensation occurs on the surface. There are two types of condensation:

1. The condensed liquid, often called the condensate, wets the surface and forms a film covering the entire surface. This type of condensation is called *filmwise condensation*.
2. The surface is not totally wetted by the saturated vapor, and the condensate forms liquid droplets that fall from the surface. This type of condensation is called *dropwise condensation*.

Compared with filmwise condensation, dropwise condensation has a greater surface heat-transfer coefficient because it has a greater surface area exposed to the saturation vapor. In practice, however, the entire surface of the condenser tubes becomes wet during refrigerant condensation. Filmwise condensate falls in droplets and often disturbs the condensate films in a tube bundle.

The heat-transfer process in a refrigerant condenser occurs in three stages:

1. Desuperheating of the hot gas
2. Condensing of the gas to liquid state and release of the latent heat
3. Subcooling of the liquid refrigerant

Although the surface heat-transfer coefficient is lower on the hot gas side during desuperheating, there is a greater temperature difference between the hot gaseous refrigerant and the cooling medium during desuperheating. Subcooling only occupies a small portion of the condenser's surface area. Therefore, for simplification, an average heat-transfer coefficient is used for the whole condenser's surface area, and the condensation of refrigerant is assumed to occur at the condensing temperature.

Total Heat Rejection

The capacity of a condenser is rated according to its total heat rejection Q_{rej}, Btu/h (W). Total heat rejection Q_{rej} is defined as the total heat removed from the condenser during desuperheating, condensation, and subcooling of the refrigerant in the condenser, and it is expressed as

$$Q_{\text{rej}} = 60\dot{m}_r(h_2 - h_{3'}) \tag{10.15}$$

where $\dot{m}_r$ = mass flow rate of refrigerant in condenser, lb/min [kg/(60 s)]
$h_2, h_{3'}$ = enthalpy of hot gas entering condenser and enthalpy of subcooled liquid leaving condenser, Btu/lb (J/kg)

In refrigerating systems using hermetic compressors, the heat released by the motor is absorbed by the refrigerant. If the heat gains and losses from the ambient air at the evaporator, condenser, suction line, and discharge line are ignored, then

$$Q_{\text{rej}} = \frac{Q_{rl} + 2545P_{\text{com}}}{\eta_{\text{mot}}} \tag{10.16a}$$

where Q_{rl} = refrigeration load at evaporator, Btu/h
P_{com} = power input to compressor, hp
η_{mot} = efficiency of hermetic compressor motor

For a refrigerating system equipped with open compressors, $\eta_{\text{mot}} = 1$.

For water-cooled condensers used in refrigeration systems using chilled water as the cooling medium, chilled water pump power heat gain and chilled water piping heat gain should be added to the refrigeration load by multiplying by a water heat gain factor $F_{w,h}$. Its magnitude varies from a water system with a small plant-building loop $F_{w,h} = 1.05$ to a campus-type water system $F_{w,h} = 1.10$. Equation (10.16*a*) can then be modified to

$$Q_{\text{rej}} = \frac{F_{w,h}Q_{rl} + 2545\,P_{\text{com}}}{\eta_{\text{mot}}} \tag{10.16b}$$

Heat rejection factor F_{rej} is defined as the ratio of the total heat rejection to the refrigeration load of the evaporator, and it can be calculated as

$$F_{\text{rej}} = \frac{Q_{\text{rej}}}{Q_{rl}} = F_{w,h} + \frac{2545\,P_{\text{com}}}{Q_{rl}\,\eta_{\text{mot}}} \tag{10.17}$$

Here the power input to the compressor $P_{com} = \dot{m}_r W_{in}$, and W_{in} represents the work input. Therefore, F_{rej} depends mainly on the difference between the condensing and evaporating pressures, hence F_{rej} is always greater than 1.

Types of Condenser

A condenser is a major system component of a refrigeration system. It is also an indirect-contact heat exchanger in which the total heat rejected from the refrigerant is removed by a cooling medium, usually air or water. As a result, the gaseous refrigerant is cooled and condensed to liquid at the condensing pressure. Liquid refrigerant is often subcooled to a temperature up to 15°F (8.3°C) below the saturated temperature at condensing pressure to conserve energy.

Based on the cooling medium used, condensers used in refrigeration systems can be classified into the following three categories:

1. Water-cooled condensers
2. Air-cooled condensers
3. Evaporative condensers

10.6 WATER-COOLED CONDENSERS

Types of Water-Cooled Condenser

In a water-cooled condenser, condenser water or cooling water is used to remove condensing heat from the refrigerant. The condenser water is often the recirculating water from the cooling tower. It also could be from a lake, river, or well near the refrigeration plant. Groundwater discharged from the condenser may be treated and used for other purposes. If river or lake water is used as the condenser water, an effective water filter system, or sometimes a heat exchanger, may be necessary to prevent fouling of condenser tubes. Use of river, lake, or groundwater is a once-through method, as it cannot be recirculated and used as condenser water again.

If a refrigeration plant is near the seashore, sometimes seawater can be used as condenser water if it is more economical than water from the cooling tower. Seawater is corrosive to many metals. Copper-nickel alloy, stainless steel, and titanium are often used for water tubes in seawater condensers. An effective filter system is necessary to prevent dirt and impurities from blocking the water passages. Water treatment is often required to prevent the growth of living organisms. The whole seawater intake system may be very expensive. The use of seawater is also a once-through method.

Two types of water-cooled condensers are widely used for air conditioning and refrigeration purposes: shell-and-tube and double-tube condensers. A double-tube condenser consists of two tubes, one inside the other, as shown in Fig. 10.9. Water is pumped through the inner tube while refrigerant flows in the space between the inner and outer tubes in a counterflow arrangement. Because double-tube condensers provide limited condensing area, they are only used in small refrigeration systems such as indoor packaged units.

Shell-and-Tube Condensers

There are two types of shell-and-tube condensers: horizontal and vertical. Vertical shell-and-tube condensers usually incorporate a one-pass water flow arrangement. They consume more condenser water than the horizontal type and are often located outdoors because of their height. Vertical shell-and-tube condensers are sometimes used for ammonia refrigeration systems in industrial applications.

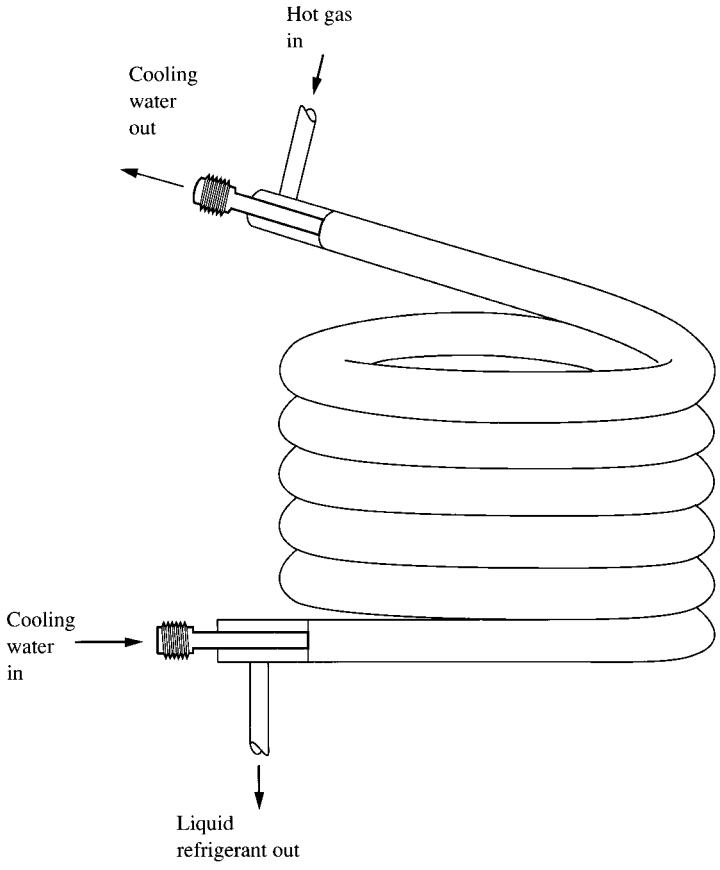
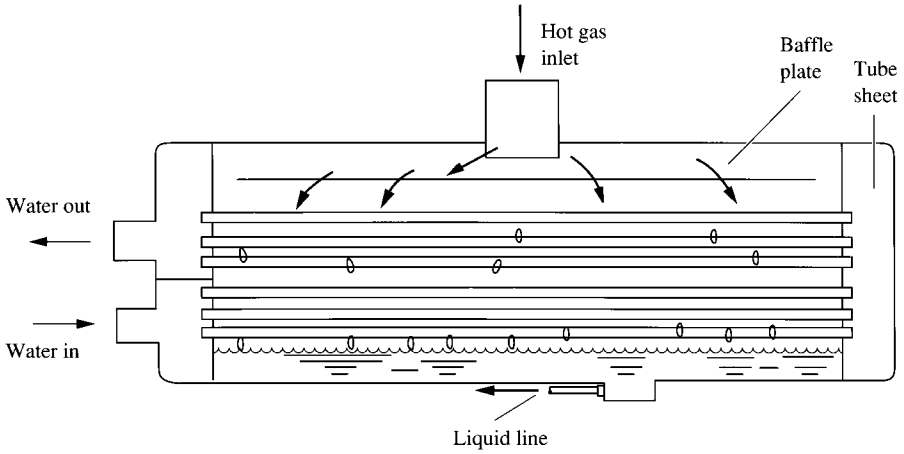


FIGURE 10.9 Double-tube condenser.

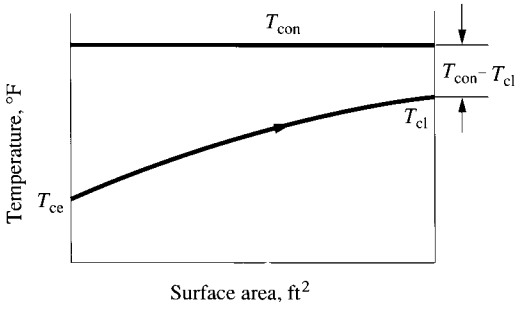
Horizontal shell-and-tube condensers are widely used in both comfort and process air conditioning systems. A horizontal shell-and-tube condenser using halocarbon refrigerant has an outer shell in which copper tubes are held in position by tube sheets, as shown in Fig. 10.10. As in the shell-and-tube liquid cooler, copper tubes with diameters of $^3/_4$ and 1 in. (19 and 25 mm) are often used. Integral external fins with spacing between 19 and 40 fins/in. (0.6 to 1.3 fins/mm) and at a height of 0.035 to 0.06 in. (0.9 to 1.5 mm) are usually used. Surface coating is employed to enhance the condensing coefficient of the tube's outer surface on the refrigerant side. Spiral grooves or other turbulators are added to the inner surface to increase the water-side heat-transfer coefficient.

Hot gas from the compressor enters at the top inlet and is distributed along the two sides of the baffle plate to fill the shell. Condenser water enters the copper tubes at the bottom of the condenser to provide effective subcooling. The water extracts heat from the hot gas and the liquid refrigerant and is discharged near the top of the shell. Water flow arrangements are the same as in shell-and-tube liquid coolers: one-pass, two-pass, and three-pass, with two-pass as the standard equipment.

After heat has been removed, the hot gas is desuperheated, condensed to liquid, and subcooled to a lower temperature. Because of the higher condensing pressure, the liquid refrigerant is forced into the liquid line. In a shell-and-tube condenser, usually about one-sixth of the shell's volume is filled with liquid refrigerant for subcooling. This liquid level prevents gas bubbles from entering the liquid line.



(a)



(b)

FIGURE 10.10 Shell-and-tube condenser with two-pass water flow arrangement: (*a*) schematic diagram; (*b*) heat transfer.

Heat Transfer in Shell-and-Tube Condensers

As in shell-and-tube liquid coolers, heat transfer in shell-and-tube condensers can be calculated by using Eq. (10.9) with the following modifications:

- Heat is rejected from the refrigerant in the condenser, whereas heat is absorbed in the liquid cooler.
- Evaporating temperature T_{ev} should be replaced by condensing temperature T_{con}, °F (°C).
- $T_{ee} - T_{el}$ should be replaced by $T_{cl} - T_{ce}$. Here T_{ce} and T_{cl} represent the condenser water temperature entering and leaving the condenser. Also $h_{rl} - h_{re}$ is replaced by $h_{re} - h_{rl}$, and $T_{ee} - T_{ev}$ is replaced by $T_{con} - T_{ce}$.

In 1915, Nusselt developed the correlation for laminar film condensation. Actual shell-side condensing coefficient h_{con}, $\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$ ($\text{W/m}^2 \cdot °\text{C}$), is needed to correct for vapor shear, inundation due to tube bundle, surface effects, and enhancements. These factors are usually determined by experiment.

It is important to recognize that the shell-side condensing coefficient h_{con} is a function of $1/(T_{con} - T_s)$. Here T_s is the surface temperature of tube wall. Therefore, the condensing coefficient can be calculated as

$$h_{con} = h_o = C_{con}\left(\frac{1}{Q_{rej}/A_o}\right)^{1/3} \tag{10.18}$$

where C_{con} = constant. The difference between the boiling and condensing heat-transfer coefficient is that the greater the heat flux Q/A_o, the higher the boiling heat-transfer coefficient h_b and the lower the condensing coefficient h_{con}.

Parameters That Influence the Performance of Shell-and-Tube Condensers

Temperature Difference $T_{cl} - T_{ce}$. Temperature difference $T_{cl} - T_{ce}$ determines the condenser water flow rate. When the condenser water is from the cooling tower, the temperature difference between the condenser water leaving and entering the shell-and-tube condenser $T_{cl} - T_{ce}$ has a direct impact on condensing temperature T_{con} and the compressor's power input, condenser water pump power, and cooling tower fan power input. The performance of the cooling tower must also be considered. A temperature difference $T_{cl} - T_{ce} = 8°F$ (4.4°C) means a condenser water flow rate of 3 gpm/ton (0.054 L/s·kW), and $T_{cl} - T_{ce} = 12°F$ (6.7°C) means a flow rate of 2 gpm/ton (0.036 L/s·kW).

Kirsner (1996) showed that for a three-stage centrifugal chiller using HCFC-123 as refrigerant, a 3 gpm/ton (0.054 L/s·kW) condenser water flow rate had a compressor's kW/ton of 0.596 (5.9 COP) at $T_{ev} = 37.6°F$ (3.1°C), whereas a 2 gpm/ton (0.036 L/s·kW) flow rate had a 0.644 kW/ton (5.46 COP), an increase of compressor power input of about 8 percent. At the same time, condenser water flow rate of 2 gpm/ton (0.036 L/s·kW) showed a saving of condenser water pump power of 3.4 percent and a cooling tower fan power saving of about 3 percent compared to the flow rate of 3 gpm/ton (0.054 L/s·kW). In addition, the initial cost of a condenser water flow rate of 2 gpm/ton was lower than the flow rate of 3 gpm/ton.

When lake, river, well, or seawater is used, the scarcity of water and water temperature are factors to be considered. A cost analysis is often required to determine the temperature difference $T_{cl} - T_{ce}$.

Condensing Temperature T_{con}. In a shell-and-tube condenser, T_{con} is closely related to the temperature of condenser water T_{ce} available. If $T_{cl} - T_{ce}$ has been selected, then T_{cl} is a fixed value and T_{con} is determined by choosing an optimum value of $T_{con} - T_{cl}$. This value is directly related to the size of the shell-and-tube condenser, as shown in Fig. 10.10*b*. As in the shell-and-tube liquid cooler, a smaller $T_{con} - T_{cl}$ means a lower T_{con} and a large condensing surface area, whereas a larger $T_{con} - T_{cl}$ indicates a higher T_{con} and a smaller condenser. Condenser manufacturers usually adopt a $T_{con} - T_{cl}$ value between 6 and 10°F (3.3 and 5.6°C), and it is 4 to 7°F (2.2 and 3.9°C) for energy-efficient models.

Effect of Oil. If the lubrication oil is miscible with the refrigerants (like many halocarbon refrigerants), experiments show that there is no significant effect on condensing coefficient h_{con} when oil concentration is lower than 7 percent. In actual operation, oil concentration is usually lower than 3 percent.

Subcooling. Subcooling increases the cooling capacity and coefficient of performance. However, subcooling also uses part of the surface area in the condenser to cool the liquid refrigerant at the bottom of the condenser. The surface heat-transfer coefficient on the subcooled liquid refrigerant side is smaller than the condensing coefficient h_{con}. Subcooling also depends on the temperature of entering condenser water T_{ce}. For a shell-and-tube condenser, it may vary from 2 to 8°F (1.1 to 4.4°C).

Condenser Capacity. Parameters that influence the condenser capacity Q_{rej} of a shell-and-tube condenser are mainly U_o, A_o, ΔT_m, and Q_{rej}/A_o. Size and capacity of the condenser must be matched with the evaporator and compressor. Condenser capacity is always rated at certain operating conditions.

Part-Load Operation

When the heat flux Q_{rej}/A_o reduces at the condensing surface of the shell-and-tube condenser during part-load operation, h_{con} increases. Therefore, the reduction of Q_{rej} at part-load operation in a shell-and-tube condenser is mainly caused by the drop in condensing temperature T_{con} and, therefore, a smaller ΔT_m between refrigerant and condenser water. At part-load operation, the degree of subcooling is also reduced because of the drop in condensing temperature T_{con}.

10.7 AIR-COOLED CONDENSERS

Construction of Air-Cooled Condenser

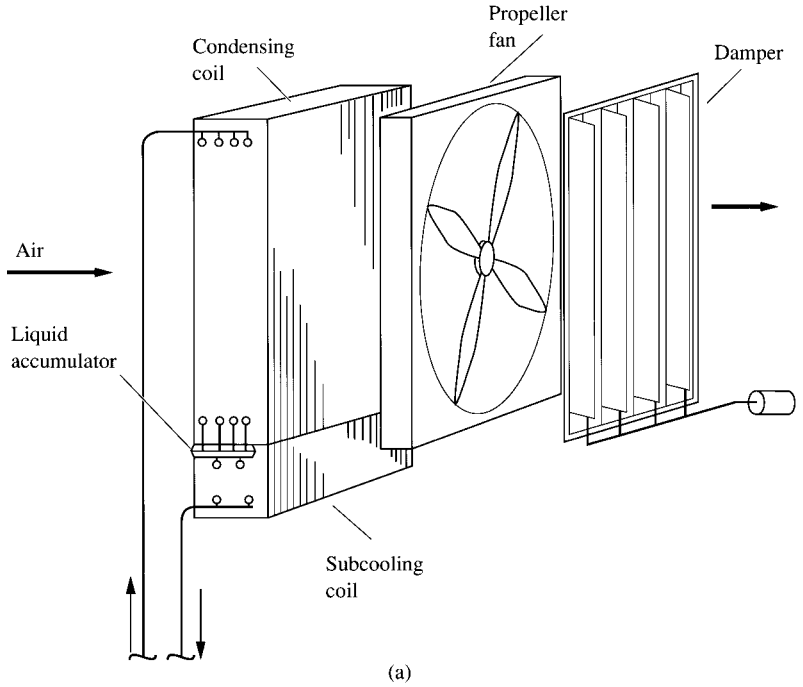
Air-cooled condensers use air to extract the latent heat of condensation released by the refrigerant during condensation. An air-cooled condenser generally consists of a condenser coil in which there are a main condensing coil section and a subcooling coil section connected in series, with several refrigerant circuits to condense the gaseous refrigerant to liquid, and a subcooled refrigerant at a lower level, as shown in Fig. 10.11. The condenser coil is usually equipped with copper tubes and aluminum fins when halocarbon is used as refrigerant. The diameters of the tubes are usually between $\frac{1}{4}$ and $\frac{3}{4}$ in. (6.5 and 19 mm), and the fin spacing is generally 12 to 20 fins/in. (1.3 to 2 fins/mm). On the inner surface of the copper tubes, microfins (typically 60 fins/in. or a fin spacing of 0.4 mm) with a height of 0.008 in. (0.2 mm) are used, as in DX coils. The condenser coil usually has two to three rows of tubes because of the lower air-side pressure drop provided by the propeller fan.

Hot gas from the compressor enters the refrigerant circuits at the top. This arrangement provides flexibility between the condensing and subcooling areas. As the condensate increases, part of the condensing area can be used as subcooling for the storage of liquid refrigerant. A receiver is necessary only when not all the liquid refrigerant can be stored in the condenser coil during the shutdown period of the refrigerant plant in winter. Cooling air is usually forced through the coil by a propeller fan, as shown in Fig. 10.11. A propeller fan has a lower fan total pressure and large volume flow rate, which make it more suitable for air-cooled condensers. Fans are usually located downstream from the coils in order to provide an even airstream through the coils. A damper may be installed after or before the fan to modulate the air volume flow rate. In a small air-cooled condenser, the coils, propeller fan, and damper may be installed in line horizontally. In large air-cooled condensers, condensing and subcooling coils are usually located on two sides, and the propeller fans and dampers are at the top of the unit.

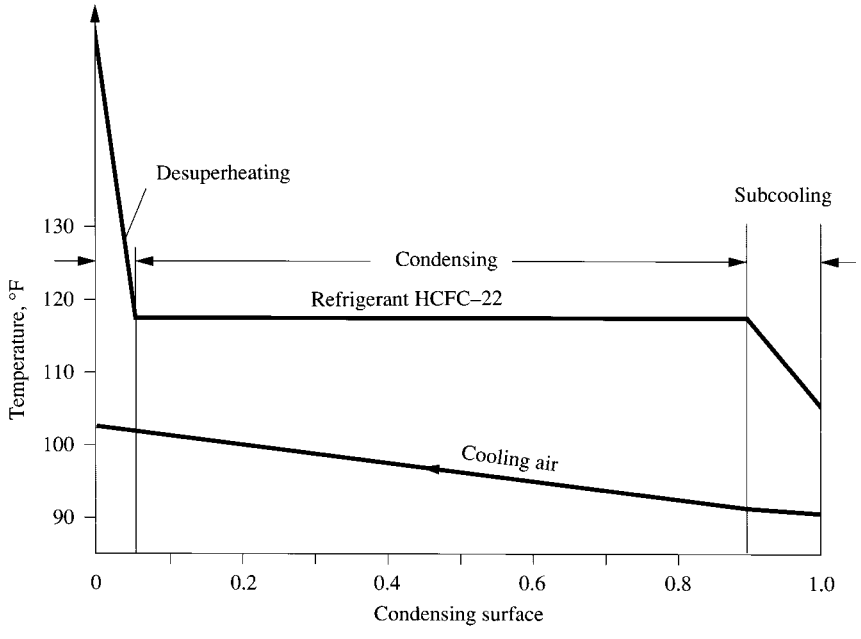
Heat-Transfer Processes and Temperature Curves

Heat-transfer processes in a typical air-cooled condenser using HCFC-22 as refrigerant are divided into three stages, as shown in Fig. 10.11*b*: desuperheating of hot gas, condensation of liquid refrigerant, and subcooling. Desuperheating uses about 5 percent of the condensing surface area, condensation uses 85 to 90 percent, and subcooling uses 5 to 10 percent.

For a condenser coil with a depth of two to three rows, airflow and refrigerant flow are usually in a combined counterflow and cross-flow arrangement. Air enters the condenser coil at a temperature equal to the summer outdoor design dry-bulb temperature T_o. In a year of average weather, T_o is the temperature whose annual cummulative frequency of occurrence is about $0.01 \times 8760 = 88$ h. If the air-cooled condenser is installed on the roof, an additional 3 to 5°F (1.7 to 2.8°C) should be



(a)



(b)

FIGURE 10.11 Air-cooled condenser: (*a*) typical construction; (*b*) temperature curves of refrigerant HCFC-22 and cooling air.

added to the summer outdoor design dry-bulb temperature to account for the temperature increase of the roof due to solar heat. Let $T_o = 90°F$ (32.2°C). For a typical condenser, entering air temperature is gradually raised to T_{al} of about 102°F when it leaves the condenser coil. Hot HCFC-22 gas may enter the top of the condenser coil at a temperature $T_{gas} = 170°F$ (76.7°C) and is desuperheated to a saturated temperature of about 115°F (46.1°C). Gaseous refrigerant condenses to liquid and releases the latent heat of condensation. As the refrigerant flows through the condensing coil, the condensing pressure drops, typically by about 8 psi (55 kPa), before entering the subcooling coil. Because no latent heat is released, the temperature of the subcooled liquid refrigerant drops sharply to 105°F (40.6°C) when it leaves the air-cooled condenser.

Equations to calculate heat transfer between refrigerant and air in an air-cooled condenser are similar to those for a water-cooled condenser, except that the temperature of condenser water should be replaced by the temperature of cooling air, and the air-side heat-transfer coefficient h_o is discussed in Chap. 17. The condensing coefficient for horizontal tubes can be found in *ASHRAE Handbook 1997*, *Fundamentals.*

Total heat rejection Q_{rej} of an air-cooled condenser in a DX packaged system can still be calculated by Eq. (10.16*a*). However, for air-cooled condensers, the refigeration load Q_{rl} in Eq. (10.16*a*) is equal to the DX coil load Q_c, both in Btu/h (W), and $F_{w,\,h} = 1$.

Cooling Air Temperature Rise and Volume Flow

For a specific Q_{rej}, temperature rise between air entering and leaving the air-cooled condenser $T_{al} - T_o$ depends mainly on the volume flow rate of cooling air per unit of total heat rejection $\dot{V}_{ca}/Q_{u,\,rej}$. Here Q_{rej} is often expressed in tons of refrigeration capacity at the evaporator; therefore, $\dot{V}_{ca}/Q_{u,\,rej}$ is expressed in cfm/ton (L/s·kW). A smaller $\dot{V}_{ca}/Q_{u,\,rej}$ usually results in a lower air-side heat-transfer coefficient h_o, a greater $T_{al} - T_o$, a higher T_{con}, a lower condenser fan power, and a greater log-mean temperature difference ΔT_m between refrigerant and air. Conversely, a larger $\dot{V}_{ca}/Q_{u,\,rej}$ and a smaller $T_{al} - T_o$ mean a lower T_{con}, a smaller ΔT_m, greater fan power consumption, and probably more noise from the propeller fans. Based on cost analysis, the optimum value of $\dot{V}_{ca}/Q_{u,\,rej}$ is usually 600 to 1200 cfm/ton (80.5 to 161 L/s·kW) refrigeration capacity at the evaporator. In comfort air conditioning systems, if the heat rejection factor $F_{rej} = 1.25$, air volume flow may be between 40 and 80 cfm per 1 MBtu/h total heat rejection. When $\dot{V}_{ca}/Q_{u,\,rej} = 900$ cfm/ton (120.8 L/s·kW) refrigeration capacity, $T_{al} - T_o$ is around 13°F (7.2°C). Fan power consumption in air-cooled condensers is usually 0.1 to 0.2 hp/ton (0.075 to 0.15 kW/ton) refrigeration capacity.

Condenser Temperature Difference

The condenser temperature difference (CTD) for an air-cooled condenser is defined as the difference in saturated condensing temperature corresponding to the refrigerant pressure at the inlet and the air intake dry-bulb temperature $T_{con,\,i} - T_o$. The total heat rejection Q_{rej} of an air-cooled condenser is directly proportional to its CTD. The Q_{rej} of an air-cooled condenser operated at a CTD of 30°F (16.7°C) is approximately 50 percent greater than if the same condenser is operated at a CTD of 15°F (8.3°C). On the other hand, for a specific value of Q_{rej}, an air-cooled condenser selected with a CTD of 15°F (8.3°C) is larger than a condenser elected with a CTD of 30°F (16.7°C).

Air-cooled condensers are rated at a specific CTD related to the evaporating temperature T_{ev} of the refrigeration system in which the air-cooled condenser is installed. Typical CTD values are as follows:

T_{ev}, °F	CTD, °F
45, for air conditioning	20 to 30
20	15 to 20
−20 to −40	10 to 15

A refrigeration system with a lower T_{ev} is more economical to equip with a large condenser with a small CTD.

Condensing Temperature, Subcooling, and Oil Effect

A high condensing temperature T_{con} means a higher condensing pressure p_{con} and a higher discharge temperature. High T_{con} and p_{con} result in a higher compressor power input and, more often, unsafe operating conditions. Both are undesirable. At a specific location, an air-cooled condenser always has a higher T_{con} than a water-cooled condenser.

A higher T_{con} and p_{con} may be caused by an undersized air-cooled condenser or a low $\dot{V}_{ca}/Q_{u,\,rej}$ value. It may be the result of a high T_o in a specific location or a high roof temperature. It may also be caused by a lack of cooling air or warm air recirculation, which is discussed below.

In air-cooled condensers, the degree of subcooling usually ranges from 10 to 20°F (5.6 to 11.1°C). Based on the results of Schlager et al. (1989) for HCFC-22 in microfin tubes, the condensing heat-transfer coefficient decreased by 15 to 20 percent with a 150-SSU (Saybolt seconds universal viscosity) and 300-SSU oil concentration of 2.6 percent. Experiments also showed a larger degradation of heat transfer with the more viscous oil.

Clearance, Warm Air Recirculation, and Dirt Clogging

Air-cooled condensers must provide clearance between the condenser coil and the wall, or adequate space between two condenser coils, as specified in the manufacturer's catalog, to provide sufficient cooling air. Sufficient clearance can also prevent recirculation of warm air discharged from air-cooled condensers. Generally, the clearance should not be less than the width of the condenser coil.

Recirculation of discharged warm air from the air-cooled condenser raises the average air temperature entering the condenser coil T_o and, therefore, increases T_{con} and reduces the capacity of the condenser. Recirculation of warm air may be caused by insufficient clearance, insufficient upward discharge air velocity from the condenser, and wind effect.

When an air-cooled condenser is located in industrial areas or in locations where outdoor air has a high concentration of dirt, solid particles, or sometimes pollens and seeds from trees and plants, then dirts and particles may clog the condenser coil and considerably reduce the volume flow of cooling air. In these circumstances, the surface of the condenser coil must be inspected and cleaned regularly.

Low Ambient Air Control

Lower condensing pressure saves energy. However, when the ambient air or outdoor air temperature is low, the condensing pressure may drop considerably. If the condensing pressure drops below a certain value because of low outdoor temperature at part-load operation, a reciprocating vapor compression refrigerating system using an expansion valve as the throttle device will not operate normally. The supply of liquid refrigerant to the evaporator may be reduced. If refrigerant feed is insufficient, frost may form on the outer surface of the coil. The strategy is to lower the condensing pressure to the minimum limit specified by the manufacturer for effective operation. Therefore, various methods of pressure control for air-cooled condensers have been developed to prevent the condensing pressure from falling below a specific value. These include the following:

- Duty cycling, or turning the fans on and off, so that air is not forced through part of the condensing coils. This reduces the cooling air volume flow as necessary.
- Modulating the air dampers to decrease the volume flow of the cooling air.
- Reducing the fan speed so that less air passes through the condensing coil.

Low ambient air control should follow the manufacturer's instructions. One manufacturer specifies the following low ambient air control scheme:

- Control energized at 65 ± 2°F (18.3 ± 1.1°C)
- Control deenergized at 70 ± 2°F (21.1 ± 1.1°C)
- Minimum entering outdoor temperature, 25°F (−3.9°C)

Selection of Air-Cooled Condenser

Total heat rejection Q_{rej} of air-cooled condensers varies at different cooling air intake temperatures. During the selection of an air-cooled condenser, the following steps should be taken:

- Calculate the total heat rejection of the refrigerating system.
- Compare the face area of the condenser coil at the same cooling air intake temperature.
- Choose an air-cooled condenser with the required Q_{rej} at a specific cooling air intake temperature with the appropriate T_{ev}.
- Select a condenser for comfort air conditioning whose $\dot{V}_{ca}/Q_{u,\text{rej}}$ value is between 800 and 1000 cfm/ton (107 and 134 L/s·kW) refrigeration capacity at the evaporator.
- Select an air-cooled condenser whose manufacturer specifies a lower low ambient air control limit.
- Check the sound power level of the propeller fans. Noise attenuation is always expensive.

10.8 EVAPORATIVE CONDENSERS

Condensation Process

An evaporative condenser uses the evaporation of water spray to remove the latent heat of condensation of the refrigerant during condensation. It is actually a simplified combination of a water-cooled condenser and cooling tower.

An evaporative condenser consists of a condensing coil, a water spray bank, a forced-draft or induced-draft fan, an eliminator, a circulating pump, a water basin, and a casing, as shown in Fig. 10.12. Water is sprayed over the outside surface of the condensing coil. Because of the evaporation of water, heat is extracted through the wetted surface of the coil. The rest of the spray falls and is collected in the water basin. Air enters from the inlet located just above the water basin. It moves up through the condensing coil, spray nozzles, and water eliminator; is extracted by the fan; and is finally discharged at the top outlet in a counterflow arrangement. Other airflow and water flow arrangements have also been developed.

The condensing coils are usually made of bare pipes of copper, galvanized steel, or stainless steel. The high heat-transfer coefficient of the wetted surface does not require fins to increase its outer surface area. Bare pipes are not easy to foul and are easier to clear.

Heat Transfer in Evaporative Condensation

The heat-transfer process in an evaporative condenser consists also of desuperheating, condensation, and subcooling. Heat transfer from the saturated air film to the ambient air, and from the condensing refrigerant through the metal tube to the wetted surface, can be calculated as

$$h_{\text{wet}} A_o (T_{os} - T_o) = h_i A_i (T_{\text{con}} - T_{is}) \tag{10.19}$$

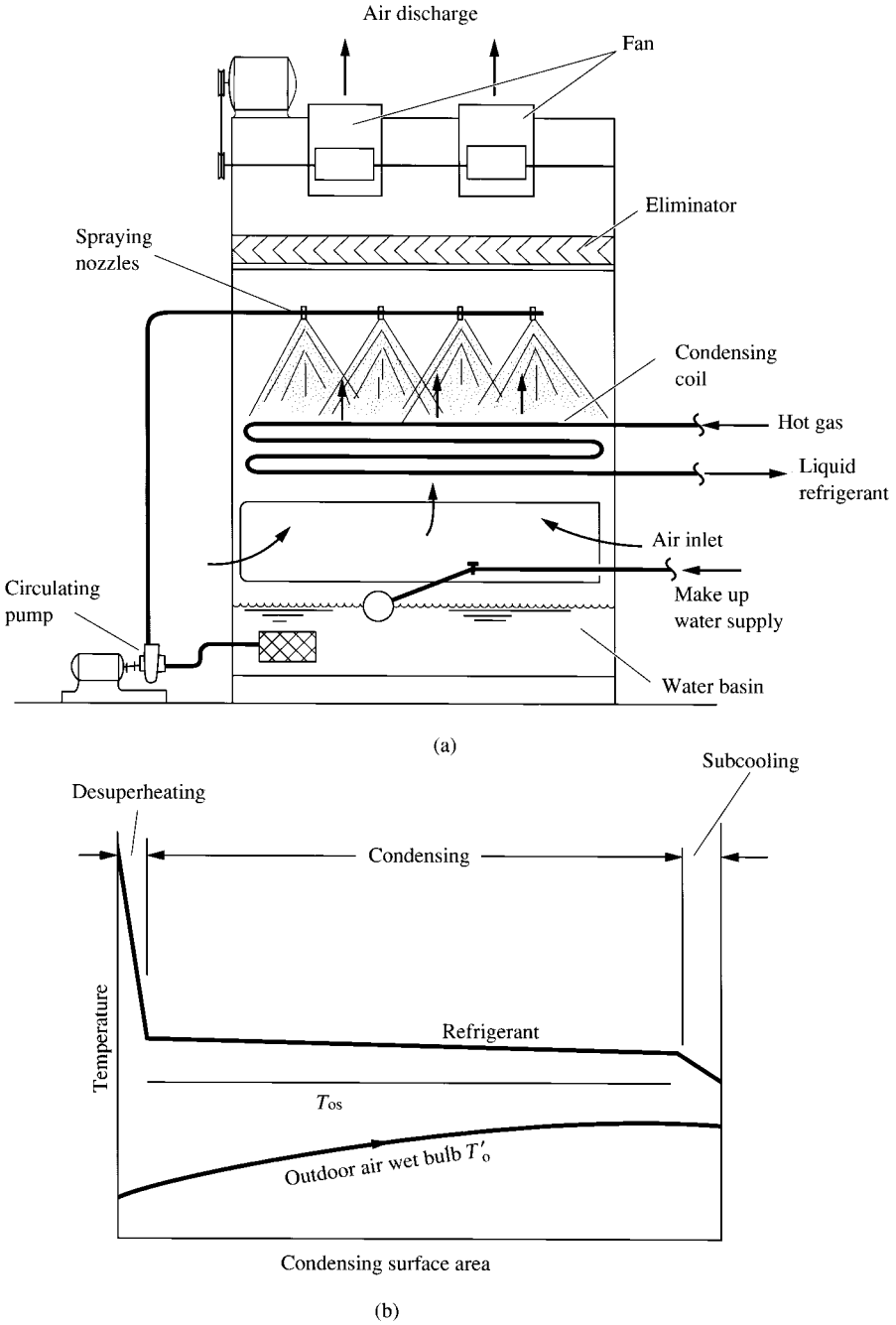


FIGURE 10.12 A typical evaporative condenser: (*a*) schematic diagram; (*b*) temperature curves.

where T_{os}, T_{is} = outer and inner surface temperature of tube, °F (°C)
T_o = ambient air temperature, °F (°C)
T_{con} = condensing temperature, °F (°C)
A_o, A_i = outer and inner surface area of tube, ft^2 (m^2)
h_i = inner surface heat-transfer coefficient, Btu/h·ft^2·°F (W/m^2·°C)

In Eq. (10.19), h_{wet} is the heat-transfer coefficient of the wet surface, Btu/h·ft^2·°F (W/m^2·°C). According to *ASHRAE Handbook 1996, HVAC Systems and Equipment*,

$$h_{wet} = \frac{m''}{c_{pa}} \frac{l}{R_{w'}} \tag{10.20}$$

Here m'' is the slope of the air enthalpy saturation curve, $m'' = dh_s/dT_s$, where h_s denotes the enthalpy of saturated air film and T_s the saturated temperature; $R_{w'}$ represents the R value of the wet film on copper tubes. And $R_{w'}$ is nearly equal to $1/h_o$. For convenience,

$$h_{wet} = \frac{m''h_o}{c_{pa}} \tag{10.21}$$

Because of the higher T_{os} in an evaporative condenser, for instance, $T_{os} = 80$°F (26.7°C), and the ratio $h_{wet}/h_o = m''/c_{pa} = 1.085/0.245 = 4.4$, so h_{wet} is often 4 or 5 times greater than h_o.

The number of transfer units, rate of heat transfer, coil effectiveness, and parameter U_oA_o can be calculated from Eq. (10.9).

Water Spraying

The water circulation rate in an evaporative condenser is far less than in the cooling tower for the same total heat rejection. Usually, it is around 1.6 to 2 gpm/ton refrigeration capacity at the evaporator.

In evaporative condensers, drift carry over the water eliminator depends mainly on the eliminator configuration and the face air velocity at the water spraying section. According to *ASHRAE Handbook 1996, HVAC Systems and Equipment*, an efficient eliminator can reduce drift loss to a range of 0.001 to 0.2 percent of the water circulation rate. The evaporated water is 1.6 to 2 gph (gallon per hour) per ton (1.7 to 2.2 kg/h·kW) of refrigeration. And the blowdown rate may be as low as one-half of the evaporation rate, and the total water consumption is about 2.4 gph/ton (2.6 L/h·kW) for air conditioning and 3 gph/ton (3.3 L/h·kW) for cold storage. According to Knebel (1997), typical pump power requirement is 0.02 hp/ton (0.004 kW/kw).

Cooling Air

In evaporative condensers, air should not be used for sensible cooling of the outer surface of the condensing coil. An air velocity of 400 to 700 fpm (2 to 3.5 m/s), typically 500 fpm (2.5 m/s), should be maintained when it flows over the condensing coil to provide a higher h_{wet} for the latent heat transfer between the wetted surface and the ambient air. Sufficient air volume flow rate is necessary to maintain a desirable enthalpy difference between the saturated film and air ($h_s - h_a$).

Both centrifugal and propeller fans are used in evaporative condensers to provide the airflow. According to Knebel (1997), typical propeller fan power requirement is 0.06 hp/ton (0.013 kW/kW).

Site Location and Low Ambient Air Control

Evaporative condensers are usually located outdoors. If they are located indoors, air is ducted outdoors and centrifugal fans are generally used to overcome the greater fan total pressure required.

The evaporative condenser should be located as near to the compressor as possible. If it is located at a distance from the compressor, the discharge refrigerant pipe should be selected to have a pressure drop of 2°F (1.1°C) of saturated temperature difference of the refrigerant between the compressor and the condenser.

If an evaporative condenser is located outdoors and will be operated during winter, its water basin and pump can be located indoors in a heated space. A water heater can be installed in the water basin, and the water should be drained back to the water basin as soon as the pump ceases to operate, to prevent freezing.

To prevent the condensing pressure from falling below a certain value at which the expansion valve or other system components are difficult to operate, the capacity of the evaporative condenser is reduced by means of fan cycling, dampers, and fan speed modulation at low ambient temperatures, as in air-cooled condensers.

Selection and Installation

The procedure for selecting and installing an evaporative condenser is as follows:

1. Before selecting an evaporative condenser, calculate the total heat rejection of the refrigerating system Q_{rej}, MBtu/h (kW).
2. According to the refrigerant used in the system, select the appropriate material for the condenser coil. Copper tubes should be used for halocarbon refrigerants, and stainless steel or galvanized steel should be used for ammonia.
3. Select a condenser coil that can provide a lower condensing temperature—below 95°F (35°C), if possible. This will save a lot of energy and reduce scale accumulation on the outer surface of the condenser coil significantly compared with $T_{con} = 100$ or 105°F (37.8 or 40.6°C).
4. Based on the location of the evaporative condenser and the sound control requirement, it is often best to select a centrifugal fan. Check the volume flow rate and the fan total pressure that can be provided by the fan to overcome the pressure drop of the condenser coil and eliminator or air duct, if any.
5. Check the flow rate of the water spray of the circulating pump.
6. For outdoor installation, select a casing and other system components that are weatherproof and corrosion-resistant.
7. Install the evaporative condenser in a location where there is ample space for outdoor air intake and discharge. The influence of nearby exhaust systems should be avoided.

Evaporative condensers have the advantage of lower condensing temperature and lower installation and energy costs. They are widely used in industrial applications and have been adopted more and more frequently in commercial buildings recently. The following reasons may explain why they were not previously popular in commercial and institutional applications:

- Monthly maintenance, inspection, and cleaning are required to prevent buildup of scale on the condenser coil.
- Water basin and water spraying system must be regularly inspected and cleaned to prevent the growth of *Legionella* bacteria and other microbiological growth. Water treatments with chemicals are often required. Mazurkiewicz (1996) reported that between the end of September and the beginning of October, 1996, there was a recent outbreak of Legionnaires' disease in a food market's evaporative condenser, in Farmington, Michigan.
- Designers and operators have less experience with evaporative condensers.
- Heat rejected from the evaporative condenser is more difficult to use in winter heating for energy conservation than that from a water-cooled condenser.

10.9 COOLING TOWERS

A cooling tower is a device in which recirculating condenser water from a condenser or cooling coils is evaporatively cooled by contact with atmospheric air. Most cooling towers used in refrigerating plants for commercial buildings or industrial applications for air conditioning are mechanical draft cooling towers. A mechanical draft cooling tower uses fan(s) to extract atmospheric air. It consists of a fan to extract intake air, a heat-transfer medium (or fill), a water basin, a water distribution system, and an outer casing, as shown in Fig. 10.13.

According to the location of the fan corresponding to the fill and to the flow arrangements of air and water, currently widely used mechanical draft cooling towers for HVAC&R can be classified into the following categories:

- Counterflow induced-draft
- Crossflow induced-draft
- Counterflow forced-draft

Counterflow Induced-Draft Cooling Towers

In a counterflow induced-draft cooling tower, as shown in Fig. 10.13*a*, the fan is located downstream from the fill at the air exit. Atmospheric air is drawn by the fan through the intake louver or, more simply, an opening covered by wire mesh. Water from the condenser or recirculating water from the coil, or a mixture of the two, is evenly sprayed or distributed over the fill and falls down into the water basin.

Air is extracted across the fill and comes in direct contact with the water film. Because of the evaporation of a small portion of the condenser water, usually about 1 percent of the water flow, the temperature of the water gradually decreases as it falls down through the fill countercurrent to the extracted air. Evaporated water vapor is absorbed by the airstream. Large water droplets entrained in the airstream are collected by the drift eliminators. Finally, the airstream and drift are discharged at the top exit. Drift, or carryover, is the minute water droplets entrained in the airstream discharged out of the tower. The evaporatively cooled water falls into the water basin and flows to the condenser.

In a counterflow induced-draft cooling tower, the driest air contacts the coldest water. Such a counterflow arrangement shows a better tower performance than a crossflow arrangement. In addition, air is drawn through the fill more evenly by the induced-draft fan and is discharged at a higher velocity from the top fan outlet. Both higher exhaust air velocity and even velocity distribution reduce the possibility of exhaust air recirculation. Compared with the crossflow induced-draft cooling tower, the vertical height from the installation level to the inlet of the water-spraying nozzles in a counterflow tower is greater and, therefore, requires a higher pump head.

Crossflow Induced-Draft Cooling Towers

In a crossflow induced-draft cooling tower, as shown in Fig. 10.13*b*, the fan is also located downstream from the fill at the top exit. The fill is installed at the same level as the air intake. Air enters the tower from the side louvers and moves horizontally through the fill and the drift eliminator. Air is then turned upward and finally discharged at the top exit. Water sprays from the nozzles, falls across the fill, and forms a crossflow arrangement with the airstream.

The crossflow induced-draft cooling tower has a greater air intake area. Because of the crossflow arrangement, the tower can be considerably lower than the counterflow tower. However, the risk of recirculation of tower exhaust air increases.

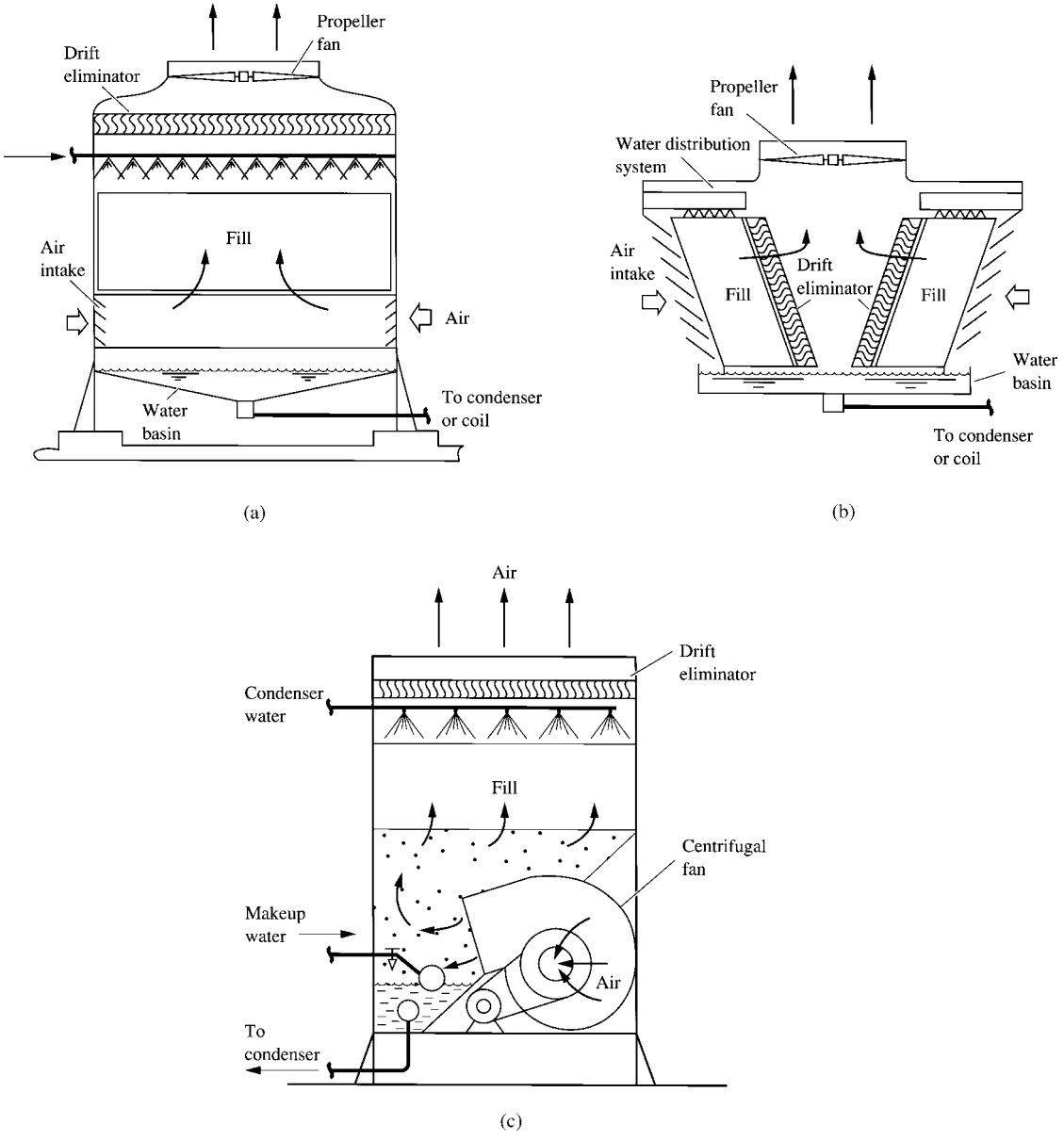


FIGURE 10.13 Cooling towers: (*a*) counterflow induced-draft; (*b*) crossflow induced-draft; (*c*) counterflow forced-draft.

Counterflow Forced-Draft Cooling Towers

In a counterflow forced-draft cooling tower, as shown in Fig. 10.13*c*, the fan is positioned at the bottom air intake, i.e., on the upstream side of the fill. Condenser water sprays over the fill from the top and falls down to the water basin. Air is forced across the fill and comes in direct contact with

the water. Because of the evaporation of the water, its temperature gradually decreases as it flows down along the fill in a counterflow arrangement with air. In the air stream, large water droplets are intercepted near the exit by the eliminator. Finally, the airstream containing drift is discharged at the top opening.

Because the fan is located near the ground level, the vibration of the counterflow forced-draft tower is small compared with that of the induced-draft tower. Also, if the centrifugal fan blows toward the water surface, there is a better evaporative cooling effect over the water basin. However, the disadvantages of this type of cooling tower include the uneven distribution of air flowing through the fill, which is caused by the forced-draft fan. In addition, the high intake velocity may recapture a portion of the warm and humid exhaust air. Counterflow forced-draft cooling towers are often used in small and medium-size installations.

The preceding types of cooling towers, which use the evaporation of water to cool the condenser water, are sometimes called wet towers. There is also a kind of cooling tower called a dry tower. It is essentially a dry cooler, a finned coil and induced fan combination that cools the condenser water flowing inside the tubes.

10.10 THERMAL ANALYSIS OF COOLING TOWERS

Technological Terms

The following terms are commonly used when referring to the performance of a cooling tower:

Approach. Temperature difference between the temperature of the condenser water leaving the tower and the wet-bulb temperature of the air entering the tower.

Blowdown. Water discharged to the drain periodically to avoid buildup of dissolved solids.

Fill. The structure that forms the heat-transfer surface within the tower. Water from the condenser or coil is distributed along the flow passages of the fill down to the water basin.

Makeup. Water added to the circulating water to compensate for the loss of water to evaporation, drift, and blowdown.

Range. Temperature difference between the temperature of condenser water entering the tower T_{te} and the temperature leaving the cooling tower T_{tl}, both in °F (°C).

Tower Coefficient

Baker and Shryock (1961) developed the following theory for the thermal analysis of the cooling tower. Because the makeup water in a typical cooling tower is usually about 2 percent of the circulating condenser water, if the heat energy difference between the makeup water and the blowdown and the drift losses are ignored, and if the enthalpy increase of the water from the addition of liquid water is ignored, then the energy balance between condenser water and air can be calculated as

$$\dot{m}_w c_{pw}\, dT_w = \dot{m}_a\, dh_a \tag{10.22}$$

where $\dot{m}_w, \dot{m}_a$ = mass flow rate of water and air, lb/h (kg/s)
c_{pw} = specific heat of water, Btu/lb · °F (kJ/kg · °C)
T_w = temperature of condenser water, °F (°C)
h_a = enthalpy of air, Btu/lb (kJ/kg)

If the thermal resistance of the saturated air film that separates the condenser water and the airstream is ignored, the combined heat and mass transfer from the air-water interface (the saturated

air film that surrounds the condenser water droplets) to the bulk airstream can be evaluated as

$$\dot{m}_a dh_a = K_m(h_s - h_a)\,dA \tag{10.23}$$

where K_m = mass-transfer coefficient, lb/h · ft^2 (kg/s · m^2)
h_s = enthalpy of saturated air film, Btu/lb (kJ/kg)
A = surface area at air-water interface, ft^2 (m^2)

In Eq. (10.23), the change of enthalpy, or total heat of air, consists of changes in sensible heat and latent heat. Consider a cooling tower with a fill volume V and a contact surface area $A = aV$, ft^2 (m^2). Here a is the surface area of fill per unit volume. Also let $K = K_m/c_{pw}$. Then combining Eqs. (10.22) and (10.23), we get

$$\frac{KaV}{\dot{m}_w} = \int_{T_{w1}}^{T_{w2}} \frac{dT_w}{h_s - h_a} \tag{10.24}$$

The integrated value of Eq. (10.24) is generally known as the tower coefficient, or the number of transfer units NTU, of the cooling tower.

Heat- and Mass-Transfer Process for Counterflow Cooling Tower

Figure 10.14 shows the heat- and mass-transfer process between condenser water and air in a counterflow cooling tower with a water-air ratio $\dot{m}_w/\dot{m}_a = 1.2$. In Fig. 10.14*a*, water is cooled from the entering temperature $T_{we} = 85$ °F (29.4°C) to the leaving temperature $T_{wl} = 75$°F (23.9°C) at the exit. The temperature of the saturated air film corresponding to the condenser water temperature drops along the saturation curve and can be represented by the section *we-wl*.

Air enters the counterflow tower at a dry-bulb $T_{ae} = 80$°F (26.7°C) and a wet-bulb $T'_{ae} = 68$°F (20°C). Because the temperature of the entering air is higher than the temperature of leaving water, that is, $T_{ae} > T_{wl}$, first air is evaporatively cooled until the air temperature approaches the water temperature. After that, air is humidified and heated because the water temperature is higher. Air is essentially saturated at the top exit of the tower. The humidifying and heating process of air is illustrated by the curve *ae-al* shown in Fig. 10.14*a*. The driving potential of heat and mass transfer at the condenser water inlet is indicated by the difference in enthalpy between the saturated air film and the air, or $h_{s,\,we} - h_{al}$. Above the water basin after the fill, it is denoted by $h_{s,\,wl} - h_{ae}$.

For humidification with evaporative cooling and heating, the sensible heat transfer from condenser water to air is only about 5 percent of the total heat transfer.

In Fig. 10.14*b*, for the same counterflow cooling tower, air enters at $T_{ae} = 72$°F (22.2°C) and at $T'_{ae} = 70$°F (21.1°C); water may be cooled from 86.5 to 76.5°F (30.3 to 24.7°C). Because $T_{wl} > T_{ae}$, air is humidified and heated along a curve nearly parallel to the saturation curve. The driving potential is still the difference in enthalpy between a saturated air film and air. Air is also approximately saturated at the top exit and approaches point *al* on the saturation curve. Under this circumstance, the sensible heat transfer is about 23 percent of the total heat transfer. This value indicates that even on rainy days, when air enters the tower in a near-saturated condition, there is still latent and sensible heat transfer between water and air. The driving potential is mainly due to the higher enthalpy of the saturated air film surrounding the condenser water droplets, which are at a higher temperature than the contacted air.

Tower Capacity, Size, and Coefficient

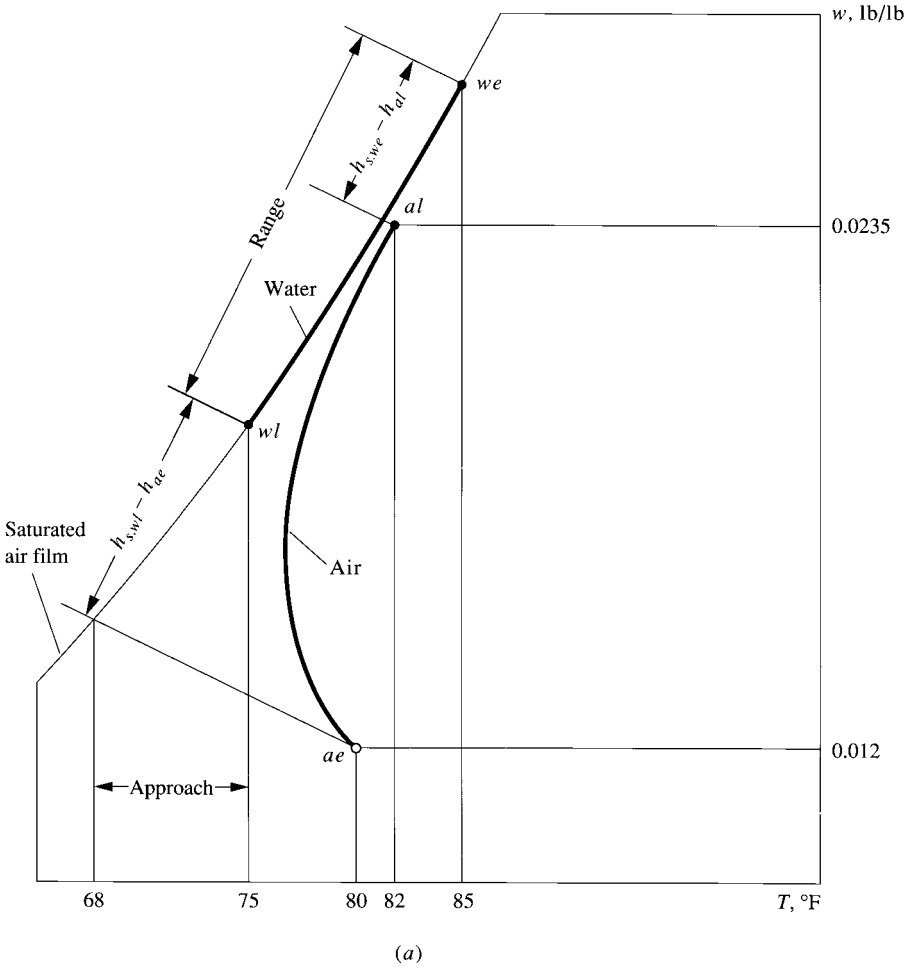
Equation (10.23) can also be expressed as

$$\dot{m}_w c_{pw}(T_{we} - T_{wl}) = K_m aV(h_s - h_a) \tag{10.25}$$

In Eq. (10.25), the total heat removed from the condenser water $\dot{m}_w c_{pw}(T_{we} - T_{wl})$ represents the cooling tower capacity in Btu/h (W) if $\dot{m}_w$ is in lb/h (kg/s). Tower capacity must meet the required total heat rejection at the water-cooled condenser.

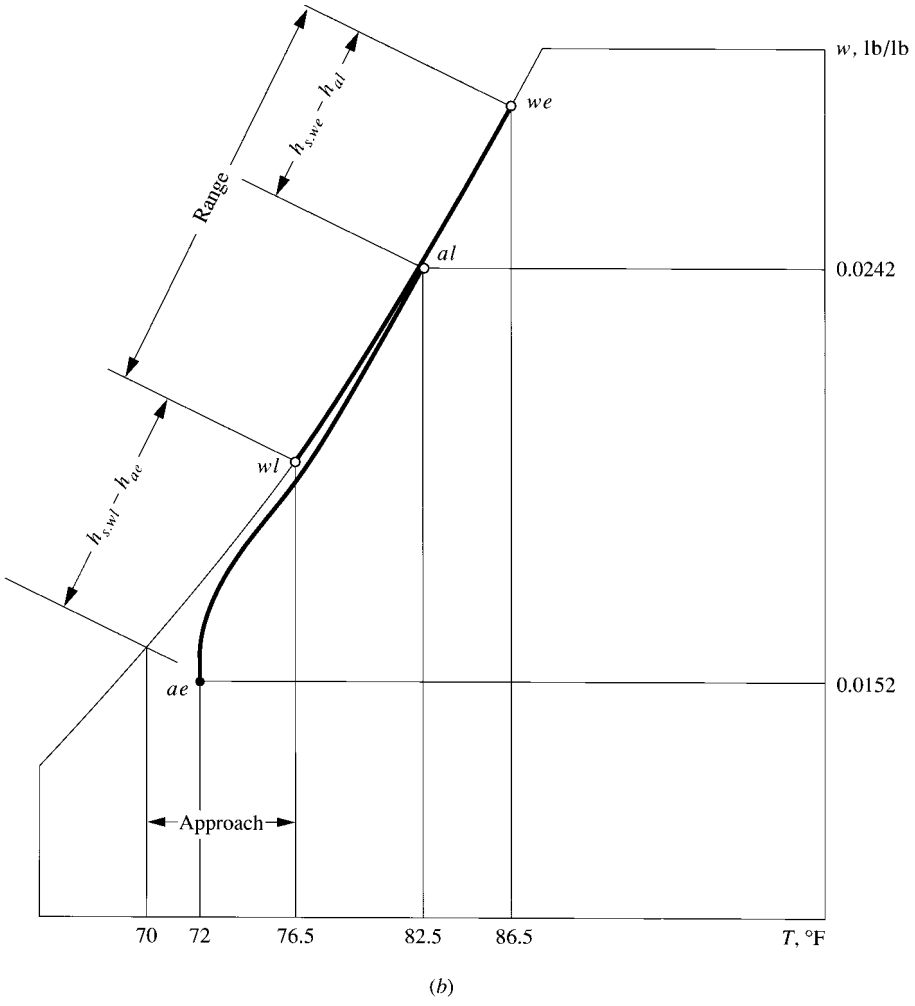
Tower size is indicated mainly by the volume of the fill V, which includes both the cross-sectional area and the depth of the fill. Strictly speaking, heat and mass transfer also occurs in the space between the fill and the water basin as well as between the fill and the water spraying. For simplicity, these can be considered as being included in the volume of fill.

The tower coefficient $KaV/\dot{m}_w$, lb·°F/Btu (kg·°C/J), actually indicates the heat-transfer unit or size of the fill. It is the primary factor that influences the effectiveness of the cooling tower. An increase of tower capacity may be attributable to a larger tower size or a higher tower coefficient or both. For a fixed water circulation rate, an increase in tower coefficient may be caused by a larger tower size or a better fill configuration.



(*a*)

FIGURE 10.14 Heat- and mass-transfer process for a counterflow cooling tower: (*a*) entering air at 80°F dry-bulb and 68°F wet-bulb temperature.



(*b*)

FIGURE 10.14 (*Continued*) (*b*) entering air at 72°F dry-bulb and 70°F wet-bulb temperature.

By using the numerical integration method, the tower coefficient for a counterflow cooling tower can be calculated as

$$\frac{KaV}{\dot{m}_w} = \frac{T_{w2} - T_{w1}}{h_s - h_a} \tag{10.26}$$

For crossflow cooling towers, the variation in air and water temperature is a function not only of the depth of the fill but also of the transverse position of the fill. A theoretical thermal analysis for crossflow towers is far more complicated than that for counterflow towers.

10.11 COOLING TOWER PERFORMANCE

Factors That Affect Cooling Tower Performance

The criteria for selecting a cooling tower for a water-cooled condenser are the effective removal of total heat rejected at the condenser and the minimization of the sum of power consumption in compressors, condenser fans, and condenser water pumps. Proper selection of tower range, water-air ratio, approach, fill configuration, and water distribution system directly affects the performance of a cooling tower.

A cooling tower used for an air conditioning system is rated on the following conditions:

- Unit of heat rejection at condenser: 1 condenser ton = 15,000 Btu/h (4.395 kW)
- Water circulation rate: 3 gpm per condenser ton (0.014 L/s per kW heat rejected)
- Entering condenser water temperature: 95°F (35°C)
- Leaving condenser water temperature: 85°F (29.4°C)
- Outdoor wet-bulb temperature: T_o': 78°F (25.6°C)
- Range: 10°F (5.6°C)
- Approach: 7°F (3.9°C)

Range and Water Circulating Rate

Many cooling towers are custom-made; i.e., the manufacturer varies the fill configuration, water circulation rate, and airflow rate to meet the operating characteristics required by the customer. The range $T_{we} - T_{wl}$, °F (°C), and the corresponding water circulating rate $\dot{m}_w$, lb/min or gpm, are the primary parameters that should be specified during design.

The range depends on the heat rejected from a water-cooled condenser Q_{rej}, Btu/h (W), and the circulation rate of condenser water $\dot{m}_w$, lb/h (kg/s), or $\dot{V}_{\text{gal}}$, gpm. Their relationship can be expressed as

$$T_{te} - T_{tl} = \frac{Q_{\text{rej}}}{c_{pw}\,\dot{m}_w} = \frac{Q_{\text{rej}}}{500\dot{V}_{\text{gal}}} \tag{10.27}$$

The greater the water circulation rate, the smaller the range. In contrast, a smaller $\dot{m}_w$ results in a greater range. During the selection of a cooling tower at the design stage, Q_{rej}, T_o', and T_{tl} are usually known and fixed. Under such circumstances, a smaller range $T_{te} - T_{tl}$ and a greater water circulating rate $\dot{m}_w$ mean the following:

- A lower condensing pressure and temperature
- A greater pumping energy consumption because of the greater $\dot{m}_w$
- A greater tower size because of smaller $h_s - h_a$
- A higher airflow rate for a fixed water-air ratio

A greater range and a smaller $\dot{m}_w$ result in a higher condensing pressure, a lower pumping energy, a smaller tower size, and a lower airflow rate.

The range is usually determined at design conditions, i.e., at summer outdoor design wet-bulb temperature T_o' and a design Q_{rej}. At a lower T_o' and part-load operation, both the range and T_{tl} will be lower. In recent years, there have been projects that used a range greater than 10°F (5.6°C) for air conditioning systems, such as 12°F (6.7°C) or even 15°F (8.3°C). A life-cycle cost analysis is recommended to determine the optimum range at specific conditions, especially for cooling towers whose annual operating time exceeds 3000 h.

Tower Coefficient and Water-Air Ratio

Cooling tower performance is often determined from field tests and presented in the form of an empirical correlation as

$$\frac{KaV}{\dot{m}_w} = C\left(\frac{\dot{m}_w}{\dot{m}_a}\right)^n z^{-m} \tag{10.28}$$

where C = constant and z = depth of the fill, ft. In Eq. (10.28), the average value of exponent n varies from -0.4 to -0.65. According to Webb (1984*a*, 1984*b*), the exponent m varies between 0.7 and 1. From the correlation in Eq. (10.28), Bernier (1994) showed that if all the other conditions remain the same and only the water-air ratio $\dot{m}_w/\dot{m}_a$ varies from 1.17 to 0.46, then the tower coefficient $KaV/\dot{m}_w$ increases 58 percent.

The face velocity of air that is calculated based on the cross-sectional area of the cooling tower perpendicular to the airflow is usually 300 to 400 fpm (1.5 to 2 m/s) with a total pressure drop across a counterflow induced-draft cooling tower of about 1 in. WC (250 Pa). This gives a mass flow density of air between 1250 and 1700 lb/h · ft^2 (6100 and 8296 kg/h · m^2).

When the mass flow density of water drops to 500 lb/h · ft^2 (2440 kg/h · m^2), the water cannot be spread sufficiently over the fill. On the other hand, when the mass flow density of water approaches 3000 lb/h · ft^2 (14,640 kg/h · m^2), the fill becomes flooding. For air conditioning systems, the mass flow density of water usually varies between 1000 and 2000 lb/h · ft^2 (4880 and 9760 kg/h · m^2), and the water-air ratio may vary between 0.7 and 1.5. For a water circulation rate of 3 gpm (0.189 L/s) per condenser ton, and $\dot{m}_w/\dot{m}_a = 1$, the airflow rate is about 360 cfm (170 L/s) per condenser ton.

Approach

Approach determines the temperature of condenser water leaving the tower T_{tl} corresponding to a local outdoor design wet-bulb T'_o. If the range and Q_{rej} are fixed values, a closer approach always means a lower T_{tl} and a lower condensing pressure and temperature.

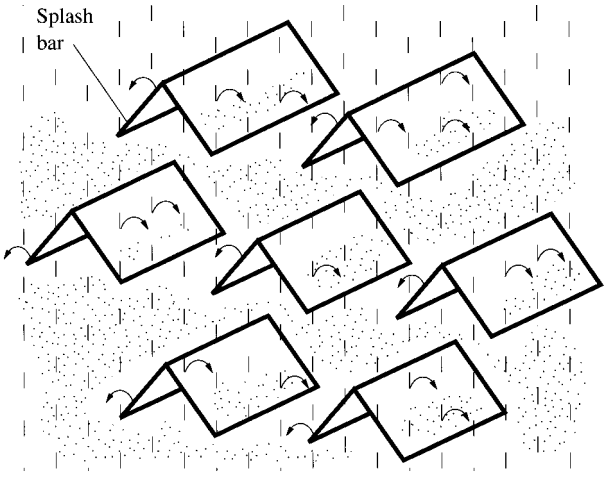
Approach is closely related to the tower coefficient and the size of the tower. A closer approach always means a larger tower, and a greater approach results in a smaller tower. For cooling towers used for air conditioning systems, approach usually varies from 5 to 12°F (2.8 to 6.7°C). When a cooling tower is connected with a precooling coil forming a water-side economizer (which is covered in detail in Sec. 21.2), a closer approach, such as 5°F (2.8°C), means a greater percentage of operating hours will be available to replace refrigeration and thereby reduce energy costs.

If other conditions remain the same, when the approach of a typical cooling tower drops from 10 to 5°F (5.6 to 2.8°C), the tower size may increase around 65 percent. An approach of 5°F (2.8°C) requires a very large tower, which is not economical and is therefore not recommended.

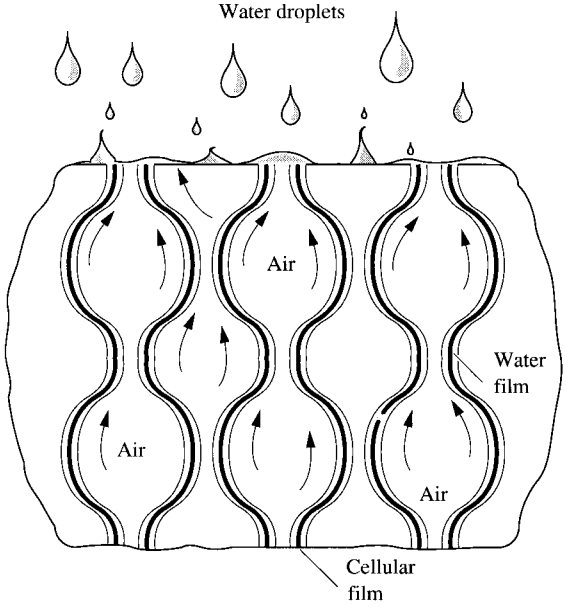
Outdoor Wet-Bulb Temperature

Because heat and mass transfer at the fill is based on the difference in enthalpy between the saturated film and the air, the wet-bulb temperature of the outdoor air is the primary parameter that affects the performance of the cooling tower. It is best to use either a 0.4 percent or 1 percent summer outdoor design wet-bulb temperature to guarantee tower performance during most of the operating period in summer, depending on the requirements of the project.

If two different refrigeration systems at different locations have the same range and approach, the one with a higher design outdoor wet-bulb temperature will have a smaller tower size than the system with a lower T'_o. This is because the enthalpy of the saturated air film is higher, so $h_s - h_a$ is greater in the system with a higher design T'_o. However, T_{tl} is lower with a lower T'_o.



(a)



(b)

FIGURE 10.15 Two types of fill for cooling towers: (*a*) splash bars; (*b*) PVC cellular film.

Fill Configuration and Water Distribution

Fill configuration has a direct effect on the tower coefficient. An efficient fill has a greater surface area per unit volume a, which means more contact surfaces, a longer contact time, and intimate contact between air and water. Currently, two types of fill are widely used in cooling towers for air conditioning systems: splash bars, as shown in Fig. 10.15*a*, and the more recently developed

cellular film, shown in Fig. 10.15*b*. Splash bars have the advantage of not being easily plugged by dirt contained in the water; the splashing action can also redistribute the water at various levels. However, field practice has shown that when wooden splash bars are replaced by polyvinyl chloride (PVC) cellular film fill, the temperature of condenser water leaving the tower drops 2°F (1.1°C) compared to the value before retrofit. Contaminated water must be strained or treated before it is cooled in a PVC cellular film fill.

An even water distribution system is very important to tower performance, especially for counterflow towers. A recently developed large-orifice, square plastic nozzle is essentially nonclogging and can provide an even pattern of spraying over the fill. It also has a far greater spraying capacity than small-orifice conical nozzles, and so its maintenance and cleaning are simple.

Construction Materials

Hot-dip galvanized steel construction provides adequate corrosion protection for most applications at a low cost. PVC and fiberglass-reinforced polyester are often used for fills, drift eliminators, inlet louvers, inlet screens, and the water basin. It is recommended to use stainless steel for the suction strainer for the cooling tower.

10.12 PART-LOAD OPERATION AND OPTIMUM CONTROL

Part-Load Operation

If the water circulating rate $\dot{m}_w$, the water-air ratio $\dot{m}_w/\dot{m}_a$, and the outdoor wet-bulb temperature T'_o remain the same, the performance of a counterflow cooling tower shows the following results when it is operated at part-load (a load ratio less than 1):

- A smaller range due to lower $Q_{\text{rej},\,p}$
- A closer approach because of a fixed tower coefficient and heat-transfer surface and a lower $Q_{\text{rej},\,p}$
- A lower T_{te} and T_{tl}

The load ratio R_{load} is defined as

$$R_{\text{load}} = \frac{Q_{\text{rej},\,p}}{Q_{\text{rej},\,f}} \tag{10.29}$$

where $Q_{\text{rej},\,p}$, $Q_{\text{rej},\,f}$ = heat rejection at condenser at part-load and full-load operation, respectively, Btu/h (W). If $\dot{m}_w$, $\dot{m}_w/\dot{m}_a$, and Q_{rej} remain the same, only T'_o falls to a lower temperature than the design condition. Then we note the following:

- A smaller range
- A smaller $h_s - h_a$ because of lower enthalpy of saturated air film at lower T'_o; also a lower T_{te} and T_{tl}
- A greater approach because of a smaller $h_s - h_a$

If $\dot{m}_w$ and $\dot{m}_w/\dot{m}_a$ remain the same, both T'_o and $Q_{\text{rej},\,p}$ drop at part-load operation. In addition to a smaller range and closer approach, there is a further drop of T_{te} and T_{tl}.

Optimum Control

During part-load operation, the capacity of a cooling tower should be controlled for effective operation of the refrigeration system at minimum energy consumption. In a typical cooling tower, the

electric power input to the tower fans is approximately 10 percent of the power input to the centrifugal compressors; and the power input to the circulating pumps for condenser water is about 2 to 5 percent of the power input to the centrifugal compressors.

One earlier strategy for capacity control of cooling towers during part-load operation is to maintain a fixed temperature T_{tl} for condenser water leaving the tower. This strategy does not minimize the sum of power input to the compressors, tower fans, and circulating pumps when both T_{tl} and the condensing temperature can be lowered. When T'_o falls, a lower condenser water temperature leaving the tower T_{tl} and entering the water-cooled condenser is always energy-saving, except when low T_{tl} causes a low condensing pressure which cannot provide effective operation of the refrigeration system refrigerant flow from the condenser and evaporator and the required evaporation (refrigeration effect) in the evaporator.

In *ASHRAE Journal* (no. 3, 1997), "The HVAC&R 'Evolution' Continues," it was reported that a centrifugal chiller operated with an entering condenser water temperature of 55°F (12.8°C), a minimum entering condenser water temperature. A better strategy of capacity control of cooling towers for water-cooled condensers in a centrifugal chiller is to lower T_{tl} until T_{tl} falls to a limit, the minimum entering condenser water temperature $T_{\text{con, }m}$. If T_{tl} falls below this limit, the refrigeration system cannot be effectively operated.

Most tower fans in current use are two- or three-speed fans. The key points for a nearly optimum control of cooling towers, according to Braun and Diderrich (1990), are as follows:

- Use sequencing of tower fans to maintain a possibly lowered $T_{tl} \geq T_{\text{con, min}}$ during part-load operation to minimize the sum of power input to the compressors and tower fans.
- Heat rejection $Q_{\text{rej, }p}$ determines the sequencing of tower fans, and $Q_{\text{rej, }p}$, is measured by the product of the range and the water circulation rate.
- Establish a simple relationship between the tower capacity and the sequencing of the tower fan.
- Use an open-loop control (see Chap. 5) for a stable operation.
- When an increase of tower capacity is required, the speed of tower fan operated at the lowest speed should be raised first. Similarly, during a decrease of tower capacity, the highest tower fan speed should be the first one to be reduced.
- The speed of the tower fan should not be reduced in sequence to maintain a T_{tl} lower than $T_{\text{con, min}}$.

Example 10.2. A counterflow induced-draft cooling tower is required to cool the condenser water from 95 to 85°F (35 to 29.4°C) at a design outdoor wet-bulb temperature of 78°F (25.6°C). If the water-air ratio of this tower is 1.2, do the following:

1. Calculate the tower coefficient at this design wet-bulb temperature.

2. If the outdoor wet-bulb temperature, water circulation rate, and airflow rate remain the same as in the design conditions, calculate the condenser water entering and leaving the tower if the rate of heat rejection at the water-cooled condenser has fallen to 50 percent of its design value.

Take the specific heat for water as $c_{pw} = 1$ Btu/lb · °F (4187 J/kg · °C).

Solution. To find the enthalpy of saturated air film h_s, the enthalpy of ambient air h_a, and the driving potentials $h_s - h_a$, do these calculations:

- Divide the temperature range of condenser water into 1°F (0.56°C) divisions such as 85, 86, . . ., 93, 94, 95°F. From App. Table B.1, the enthalpy of the saturated air film that surrounds the water droplets h_s at various temperatures can be determined.
- Find the enthalpies of the air that contacts with the condenser water at various temperatures h_a by means of Eq. (10.22), starting with incoming air at a wet-bulb $T'_o = 78$°F (25.6°C) in contact with the coldest condenser water at temperature 85°F (29.4°C).

Based on Eq. (10.26), the tower coefficient or fill surface area required to cool the condenser water from 86 to 85°F and raise the air enthalpy from that at $T'_o = 78$°F to $h_{a78} + 1.2$ is

$(T_{w2} - T_{w1})/(h_s - h_a)$, or their average, $1/2[1/(h_{s86} - h_{a78}) + 1/(h_{s85} - h_{a78})]$. The tower coefficient of the cooling tower that cools the condenser water from 95 to 85°F is the summation or integration of the tower coefficient of 95 to 94°F, . . ., 86 to 85°F.

From App. Table B.1, at a wet-bulb temperature of 78°F, the enthalpy of saturated air is 41.59 Btu/lb. Because the enthalpy of the saturated air film at a water temperature 85°F is 49.45 Btu/lb, the enthalpy difference between the saturated air film at 85°F and air at 78°F wet-bulb temperature is $h_s - h_a = 49.45 - 41.59 = 7.86$ Btu/lb, and $1/(h_s - h_a) = 0.1272$ lb/Btu (0.0609 kg/kJ).

From Eq. (10.22), for each degree of increase in condenser water temperature, the corresponding increase in the enthalpy of air is

$$dh_a = \frac{\dot{m}_w}{\dot{m}_a} c_{pw} dT_w = (1.2 \times 1)\, dT_w$$

Then at a condenser water temperature of 86°F, the enthalpy of saturated air film is 50.68 Btu/lb, and the enthalpy of the corresponding air in contact with this condenser water is

$$h_a = 41.59 + 1.2 = 42.79 \text{ Btu/lb}$$

The enthalpy difference between the saturated air film and air at $T_w = 86$°F is then

$$h_s - h_a = 50.68 - 42.79 = 7.89 \text{ Btu/lb}$$

and

$$\frac{1}{h_s - h_a} = \frac{1}{7.89} = 0.1267 \text{ lb/Btu (0.0606 kg/kJ)}$$

TABLE 10.2 Numerical Integration of a Counterflow Cooling Tower for Example 10.2

Cooling water temperature, °F	Enthalpy of saturated air film h_s, Btu/lb	Air enthalpy h_a, Btu/lb	$h_s - h_a$, Btu/lb	$\frac{1}{h_s - h_a}$, lb/Btu	Average $\frac{\Delta T_w}{h_s - h_a}$,	$\Sigma\left(\frac{\Delta T_w}{h_s - h_a}\right)$, lb/Btu · °F
Design conditions						
85	49.45	41.59	7.86	0.1272		
					0.1270	0.1270
86	50.68	42.79	7.89	0.1267		
					0.1262	0.2532
87	51.95	43.99	7.96	0.1256		
					0.1249	0.3781
88	53.25	45.19	8.06	0.1241		
					0.1231	0.5012
89	54.58	46.39	8.19	0.1221		
					0.1209	0.6221
90	55.95	47.59	8.36	0.1196		
					0.1182	0.7403
91	57.36	48.79	8.57	0.1167		
					0.1151	0.8554
92	58.79	49.99	8.80	0.1136		
					0.1118	0.9672
93	60.27	51.19	9.08	0.1101		
					0.1082	1.0754
94	61.79	52.39	9.40	0.1064		
					0.1045	1.1799
95	63.34	53.59	9.75	0.1026		
Part-load operation (50 percent design load)						
82	45.91	41.59	4.32	0.2315		
					0.2328	0.2328
83	47.06	42.79	4.27	0.2341		
					0.2350	0.4678
84	48.23	43.99	4.24	0.2358		
					0.2402	0.7080
85	49.45	45.19	4.26	0.2347		
					0.2338	0.9418
86	50.68	46.39	4.29	0.2331		
					0.2312	1.1730
87	51.95	47.89	4.36	0.2294		

The average value of $T_w/(h_s - h_a)$ between water temperature 85 and 86°F is 1(0.1272 + 0.1267)/2 = 0.1270 lb · °F/Btu (0.0338 kg · °C/kJ). Similarly, values of h_a, $h_s - h_a$, and $T_w/(h_s - h_a)$ at other water temperatures can be calculated and are listed in Table 10.2.

Using the numerical integration method, we find the tower coefficient is

$$\frac{KaV}{\dot{m}_w} = \Sigma \frac{\Delta T_w}{h_s - h_a} = 0.1270 + 0.1262 + 0.1249 + 0.1231 + 0.1209 + 0.1182 + 0.1151 + 0.1118 + 0.1082 + 0.1045 = 1.1799 \text{ lb} \cdot °\text{F/Btu } (0.3137 \text{ kg} \cdot °\text{C/kJ})$$

At 50 percent part-load operation, the tower coefficient $KaV/\dot{m}_w$, water circulation rate T'_o, and the water-air ratio remain the same. Heat rejection at the condenser $Q_{\text{rej},p} = 0.5\ Q_{\text{rej},f}$ because the range has been dropped approximately from 10°F to 0.5 × 10°F = 5°F. By trial and error, we find that when the condenser water is cooled from 87 to 82°F (30.6 to 27.8°C), the calculated tower coefficient is 1.1730 lb · °F/Btu (0.3119 kg · °C/kJ), which is approximately the same as the value at the design condition, or 1.1799 lb · °F/Btu (0.3137 kg · °C/kJ). The results are also listed in Table 10.2.

10.13 OPERATING CONSIDERATIONS FOR COOLING TOWERS

Recirculation and Interference

Most cooling towers are installed outdoors. Location and layout of the cooling towers should be carefully chosen so that recirculation, interference, and insufficient airflow are minimized.

The recapture of a portion of warm and humid exhaust air at the intake by the same cooling tower is called *recirculation*. Recirculation causes an increase in entering wet-bulb temperature and, therefore, degrades the tower performance. Recirculation is mainly caused by insufficient exhaust air velocity, or insufficient vertical distance between the intake and the exit. When a wind encounters a cooling tower, a pressure lower than atmospheric is formed on the leeward side of the tower. Thus, the exhaust air is extracted downward toward the air intake. If the exhaust air velocity is not high enough, recirculation occurs.

If the exhaust air of a cooling tower or warm air from other heat sources located upstream is extracted by another cooling tower downstream, such a phenomenon is called *interference*. Interference can be minimized by considering the summer prevailing wind direction and by properly planning the layout of the cooling towers.

When the space between the intakes of two cooling towers or the space between the wall and the intake is insufficient, or if a barrier is used to shield the view or noise, airflow may be limited, causing an airflow rate lower than the required rate. Insufficient airflow impairs the effectiveness of heat and mass transfer and results in a greater approach and range and, therefore, a higher condensing pressure.

Freeze Protection and Fogging

When a cooling tower is operated at locations where the outdoor temperature may drop below 32°F (0°C), freeze protection should be provided to avoid the formation of ice on components of the cooling towers (such as intake louvers) which may seriously restrict the airflow.

An electric immersion heater is usually installed in the water basin for freeze protection. In idle towers, the fan should not be run and water flow should be completely shut off. Condenser water may bypass the fill and be distributed directly over the inside surface of the intake louvers through a special piping arrangement. This action eliminates the accumulation of ice on the surfaces of the louvers.

When warmer and essentially saturated air that is exhausted from the cooling tower is mixed with the cooler ambient air in cold winter, the mixture becomes oversaturated, and the excess water

vapor condenses as fog, which is often called *fogging* or *plume*. Fogging limits visibility and may form ice on nearby highways. Fogging should be minimized by locating the cooling towers so that visible plumes and their influence on the nearby environment are not objectionable.

Blowdown

To avoid the concentration of impurities contained in water beyond a certain limit, a small percentage of water in a condenser water system is often purposely drained off or discarded. Such a treatment is called *blowdown* or *bleed-off*. The amount of blowdown is usually 0.8 to 1.0 percent of the total water circulating flow. Because drift in a cooling tower varies from 0.1 to 0.2 percent, if the evaporation loss in a typical location is 1 percent, the amount of makeup water is about 0.9 + 0.1 + 1.0 = 2 percent. The practice of blowdown or bleed-off is widely used in condenser water systems.

If simple blowdown is inadequate to control scale formation, chemicals can be added to inhibit corrosion and to limit microbiological growth. Water treatment to reduce scale formation and corrosion, and to control microbiological growth, is discussed in Chap. 7.

Legionnaires' Disease

In summer of 1976, American Legion conventioneers in Philadelphia suffered an epidemic in which 34 of the victims died. Later at the end of the same year, scientists at the Centers for Disease Control in Atlanta discovered the bacterium that caused Legionnaires' disease and called it *Legionella*.

According to Millar et al. (1997), there are two types of disease due to *Legionella*: the pneumonia experienced by the victim with 105°F (40.6°C) fever which can lead to respiratory failure and death; and the less severe, less frequent Pontiac fever whose patient recovers after a few days of treatment. The estimate of annual cases of Legionnaires' disease in the United States may be 10,000 to 100,000 cases, and the death rate may range from 500 to 2000.

Legionnaires' disease is always aquired from the enviroment. The HVAC&R sources generally associated with the outbreak of Legionnaires' disease are aerosol producers and are associated with reservoirs and water-spraying systems such as cooling towers, evaporative condensers, humidifiers, and direct evaporative coolers. A water source is contaminated with *Legionella*, this water is aerosolized into mist, and a victim inhales the contaminated mist—these are the necessary steps to produce an outbreak of Legionnaires' disease. Most transmission of Legionnaires' disease occurs indoors.

Legionella is a living organism. Its amplification, or growth, is preventable. Regular maintenance of reservoirs and water-spraying systems, ensuring cleanliness, proper pH value and temperature of water, and adequate level of residual biocides, is an effective measure to prevent *Legionella* growth.

The location of a cooling tower or an evaporative condenser should be carefully arranged so that the drift from the cooling tower or evaporative condenser is not extracted into the outdoor air intakes of the air-handling units or packaged units.

In addition to the prevention of *Legionella* growth in the source, the current approach of legionellosis control includes the identification of outbreaks, the investigation of potential sources, biological tests, and decontamination of the source.

Maintenance of Cooling Towers

Maintenance of cooling towers includes the following:

- Inspection of the tower fan, condenser water pump, fill, water basin, and the drift carryover to ensure satisfactory operation

- Periodic blowdown, cleaning of wetted surfaces, and effective water treatments to prevent formation of scale and dirt as well as the growth of microorganisms
- Periodic lubrication of moving parts

The manufacturer's maintenance schedule should be implemented. Otherwise, a maintenance document must be established.

If noise from the tower fan exceeds the limits specified by local codes, sound attenuators at the fan exit and at the intake louver should be installed. Details of sound control are discussed in Chap. 19.

10.14 HEAT REJECTION SYSTEMS

Types of Heat Rejection System

A heat rejection system is a system that extracts condensing heat from refrigerant in the condenser of a refrigeration system, uses some of or all the condensing heat for winter heating or other purposes, if possible, or transfers it to a heat sink, usually atmospheric air or surface water. A heat rejection system consists of a condenser, heat exchangers, fan(s) or pump, piping work, and ductwork if necessary. Four types of heat rejection systems are commonly used in vapor compression refrigeration systems, as shown in Fig. 10.16:

1. Water-cooled condensers using lake, river, ground, or seawater directly. A water-filtering device is often required to prevent dirt and solid particles from fouling the condenser tubes. Water treatment may be required depending on the quality of the cooling water.
2. Water-cooled condensers using recirculation water from the cooling towers. This type of rejection system is most widely used in large refrigeration plants for commercial buildings. Condenser water leaves the water-cooled condenser and enters the top of the cooling tower. It is sprayed over the fills and cooled to a lower temperature by means of evaporation. The condenser water then passes through a strainer and is chemically treated. Finally, it is extracted by a water pump and discharged to the water-cooled condenser again.
3. Air-cooled condensers which are widely used in small and medium-size refrigeration plants or in locations where water is scarce.
4. Evaporative condensers, which extract condensing heat from the coil directly by means of the evaporative cooling effect.

Comparison between Various Systems

Systems should be compared at summer design conditions at a specific location with the following outdoor air and water temperatures:

- Outdoor dry-bulb temperature: 95°F (35°C)
- Outdoor wet-bulb temperature: 78°F (25.6°C)
- Lake, river, or seawater temperature: 78°F (25.6°C)

The evaporating temperature and refrigeration capacity are the same for all the heat rejection systems. During comparison, factors such as condensing temperature, cooling air and water flow rate, maintenance, and initial and energy costs should be considered. The initial cost is compared on the basis of means mechanical cost data for a self-contained rooftop unit of about 32 tons (113 kW) of cooling capacity. The energy cost is compared for the whole refrigeration plant. In addition to the factors mentioned above, designers should recognize that water is becoming scarcer in many

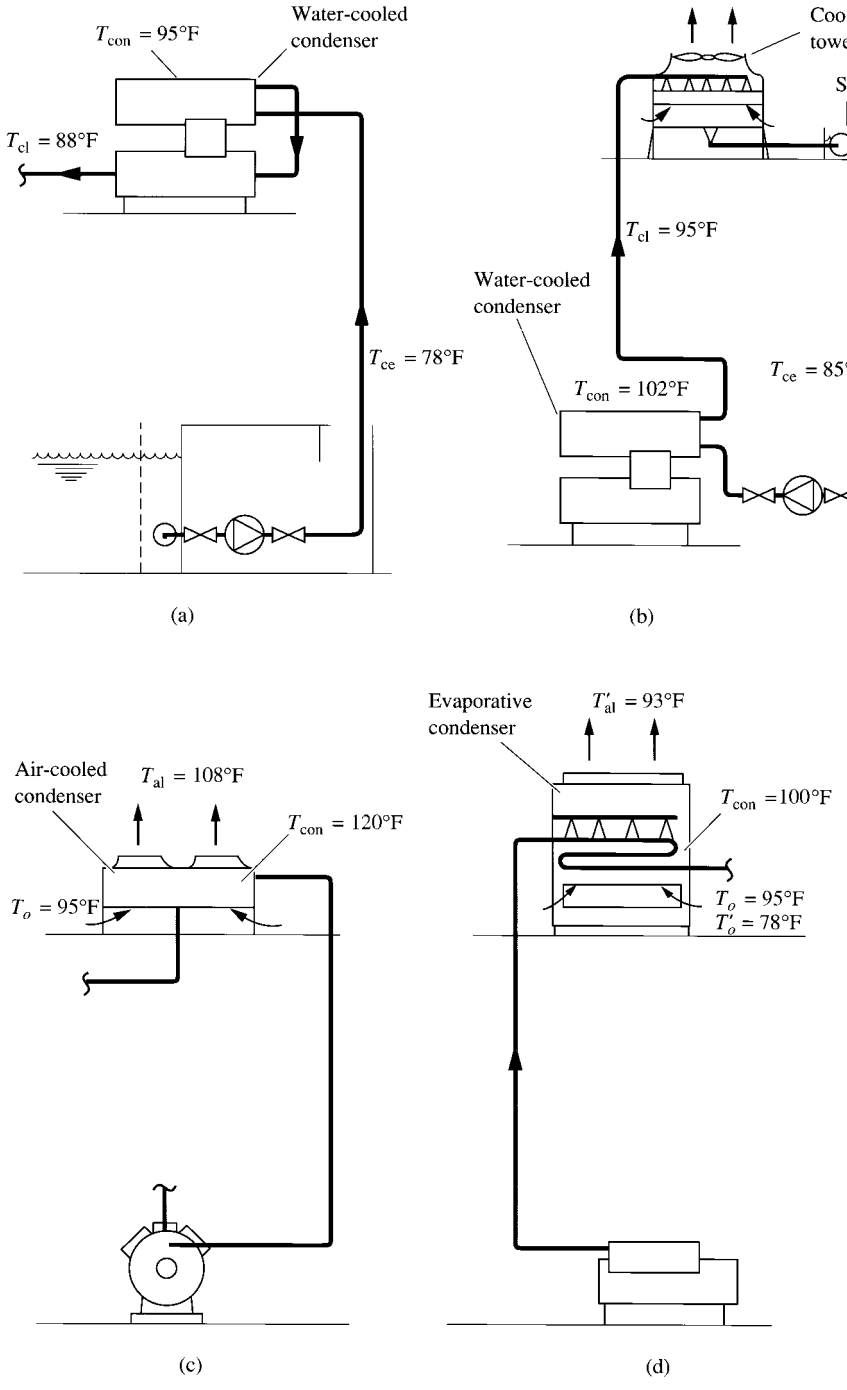


FIGURE 10.16 Heat rejection systems: (*a*) water-cooled condenser; (*b*) water-cooled condenser and cooling tower; (*c*) air-cooled condenser; (*d*) evaporative condenser.

TABLE 10.3 Comparison of Various Heat Rejection Systems at Summer Design Conditions

	Water-cooled condenser	Water-cooled condenser and cooling tower	Air-cooled condenser	Evaporative condenser
Condensing temperature, °F	95	102	120	100
Cooling air volume flow rate, cfm/ton*			600–1200	Smaller than air-cooled condenser
Cooling water, gpm/ton*	3	3		1.6–2
Makeup water, gal/(h·ton)*		2.4		2.4
Maintenance			Periodic cleaning of coil when outdoor air is not clean	Monthly inspection and cleaning of coil
Initial cost (refrigeration plant)	Depending on initial cost on water intake	Lower than air-cooled condenser	Higher	Higher
Energy consumption (refrigeration plant)	Lower	Lower than air-cooled condenser	Higher at design load	Lower
Condensing heat used for winter heating or equipment itself used for evaporative cooling	Easier	Best applied	Between water-cooled and evaporative condenser	Difficult
Application	Large	Medium and large	Medium and small size, or where water is scarce	Medium and large

Comparison is based on outdoor 95°F dry-bulb, 78°F wet-bulb; lake, river, or seawater temperature = 78°F.
*Ton refrigeration capacity at the evaporator.

locations in the United States. Whether the released condensing heat can be easily used for winter heating or the equipment itself can be used for energy conservation is also a factor that should be considered. For example, a cooling tower can be used as part of a water economizer to replace part of the refrigeration. The results of this comparison are listed in Table 10.3 for reference.

For small refrigeration systems, air-cooled condensers are still widely used. For large refrigeration plants where condensing heat is not used for winter heating, the evaporative condenser should be compared with a water-cooled condenser and cooling tower combination through detailed analysis.

ASHRAE/IESNA Standard 90.1-1999

ASHRAE/IESNA Standard 90.1-1999 defines the cooling tower performance as the maximum flow rating of the tower divide by the fan nameplate rated motor power, in gpm/hp (L/s per kW). For cooling towers, the rating conditions are: 95°F (35°C) entering water, 85°F (29.4°C) leaving water, and 75°F (23.9°C) wet bulb of the outdoor air. Standard 90.1-1999 mandates the performance required for the total system heat rejection capacity at rated conditions for cooling towers as of October 29, 2001:

$$\text{Propeller or axial fan cooling towers} \geq 38.2 \text{ gpm/hp } (3.23 \text{ L/s per kW})$$

$$\text{Centrifugal fan cooling towers} \geq 20.0 \text{ gpm/hp } (1.69 \text{ L/s per kW})$$

ASHRAE/IESNA Standard 90.1-1999 defines the air-cooled condenser performance as the heat rejected from the refrigerant divided by the fan nameplate rated motor power, in Btu/h·hp (COP).

The rated conditions of air-cooled condensers using HCFC-22 as test fluid are: 125°F (51.7°C) condensing temperature, 190°F (87.8°C) entering gas temperature, 15°F (8.3°C) subcooling, and 95°F (35°C) entering dry bulb (cooling outdoor air). Standard 90.1-1999 mandates the total system heat rejection capacity at rated conditions for air-cooled condensers as of October 29, 2001:

$$\text{Total system heat rejection capacity} \geq 176{,}000 \text{ Btu/h} \cdot \text{hp (19.7 COP or 0.18 kW/ton)}$$

ASHRAE/IESNA Standard 90.1-1999 also specifies the fan speed control including condenser fans and cooling tower fans for heat rejection systems. Each fan powered by a motor of 7.5 hp (5.6 kW) or larger can operate the fan at two-thirds of full speed or less. The fan can automatically change the fan speed to control the leaving fluid temperature or condensing temperature/pressure of the heat rejection system except for condenser fans serving multiple refrigerant circuits, or where installation located in climates with greater than 7200 CDD50, or up to one-third of the fans on a condenser or tower with multiple fans where the lead fans comply with the requirement of fan speed control.

10.15 REFRIGERANT FLOW CONTROL DEVICES

Refrigerant control devices, or refrigerant flow control devices, include expansion valves, electric expansion valves, float valves, capillary tubes, and multiple orifices. Multiple orifices are discussed in Chap. 13.

In a refrigeration system, the expansion valve is an adjustable throttling device through which the refrigerant at condensing pressure is throttled to evaporating pressure or interstage pressure. At the same time, the expansion valve regulates its opening to feed the required amount of refrigerant to the evaporator to meet the refrigeration load at the evaporator.

Thermostatic Expansion Valves

Thermostatic expansion valves (TXVs) are the most widely used throttling or expansion devices in direct-expansion refrigeration systems.

Operating Characteristics. A thermostatic expansion valve regulates its refrigerant flow rate to the evaporator according to the degree of superheat of vapor refrigerant leaving the evaporator. Figure 10.17 shows a thermostatic expansion valve connected to a DX coil, the evaporator. A thermostatic expansion valve consists of a valve body, a valve spring, a diaphragm, and a sensing bulb at the outlet of the evaporator. The sensing bulb is connected to the upper part of the diaphragm by means of a capillary tube. The outlet of the thermostatic expansion valve is then connected to various refrigerant circuits of the DX coil through the nozzles and tubes of a refrigerant distributor.

When the liquid refrigerant passes through the small opening around the valve pin, its pressure is reduced to the evaporating pressure. Liquid refrigerant flows through the refrigerant distributor and tubes, and vaporizes gradually as it flows inside the copper tubes. At position x, as shown in Fig. 10.17, all the liquid has been vaporized. By the time the vapor refrigerant reaches the outlet o of the evaporator, it is superheated to a few degrees higher than its saturated temperature.

If the coil's load of this DX coil increases, more refrigerant vaporizes. Consequently, point x tends to move toward the inlet of the evaporator. This causes an increase in the degree of superheat at the outlet o as well as an increase in the temperature of the sensing bulb. Because the sensing bulb may be partially filled with the same type of liquid refrigerant as that in the evaporator, the higher temperature exerts a higher saturated pressure on the top of the diaphragm. This lowers the valve pin and widens the valve opening. More liquid refrigerant is allowed to enter the evaporator to match the increase in refrigeration load.

If the coil's load drops, the degree of superheat of the vapor refrigerant at the outlet of the evaporator becomes smaller, resulting in a narrower valve opening. The liquid refrigerant feed to the

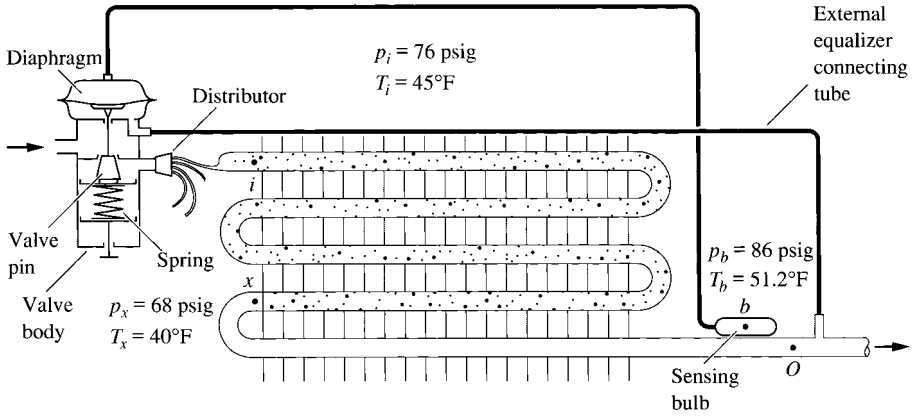


FIGURE 10.17 Thermostatic expansion valve with external equalizer.

evaporator drops accordingly. The degree of superheat of the vapor refrigerant at the outlet can be adjusted by varying the tension of the spring in the thermostatic expansion valve.

Capacity-Superheat Curve. Figure 10.18 shows the capacity-superheat curve of a thermostatic expansion valve. In Fig. 10.18, ΔT_{ss} represents the static superheat when the valve is closed at no-load condition. Static superheat ΔT_{ss} does not have sufficient pressure to open the valve from its closed position. Static superheat usually has a value of 4 to 8°F (2.2 to 4.4°C).

The term ΔT_o indicates the opening superheat, i.e., the superheat required to open the expansion valve from a closed position to a fully open position corresponding to the full refrigeration load or design coil load. Opening superheat usually has a value between 6 and 8°F (3.3 and 4.4°C). The term ΔT_{op} indicates the operating superheat at a specific refrigeration load Q_{op}, Btu/h (W). The term ΔT_{rl} is the sum of ΔT_{ss} and ΔT_o and represents the superheat setting of the thermostatic expansion valve at full-load operation. The recommended ΔT_{rl} is between 10 and 20°F (5.6 and 11.1°C). Superheat in excess of 20°F (11.1°C) means that too much of the coil's surface is producing superheat, which reduces the coil capacity. Too little superheat is also risky because it may cause the liquid refrigerant to flood the compressor. The term T_{nom} indicates nominal superheat. For safety, T_{nom} is generally equal to $1.1T_{rl}$ to $1.4T_{rl}$.

External Equalizer. A thermostatic expansion valve with an external equalizer has a connecting tube that links the bottom diaphragm chamber to the evaporator outlet *o*, as shown in Fig. 10.17, so that the pressure under the diaphragm is the same as that at outlet *o*. Because currently used DX coils have a considerable pressure drop at the refrigerant distributor and tubes before entering the coil, thermostatic expansion valves with external equalizer tubes are often used in DX coils.

Consider a DX refrigeration system using HCFC-22 as the refrigerant. The evaporating temperature at the inlet of the evaporator is 45°F (7.2°C), and the saturated pressure is 76 psig (524 kPa·g). If the spring pressure at full load p_s is 18 psig (124 kPa·g) and the pressure drop of the refrigerant flowing through the evaporator Δp_{ev} is 8 psi (55 kPa), in a thermostatic expansion valve with an external equalizer, the saturated pressure of the refrigerant at point *x*, denoted by p_x, is therefore

$$p_x = p_i - \Delta p_{\text{ev}} = 76 - 8 = 68 \text{ psig } (469 \text{ kPa} \cdot \text{g}) \tag{10.30}$$

The corresponding saturated temperature at point *x* is $T_x = 40$°F (4.4°C). In a thermostatic expansion valve with an external equalizer, the pressure balance at the diaphragm is

$$p_b = p_x + p_s = 68 + 18 = 86 \text{ psig } (593 \text{ kPa} \cdot \text{g}) \tag{10.31}$$

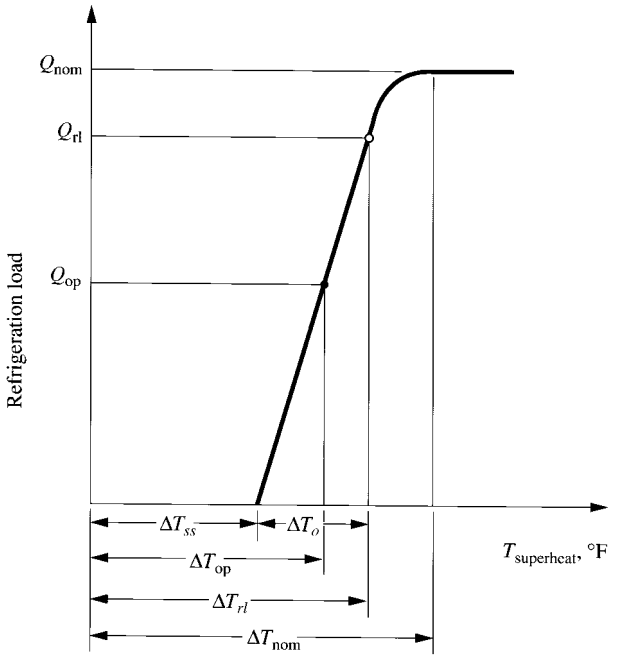


FIGURE 10.18 Capacity-superheat curve of a thermostatic expansion valve.

The corresponding saturated temperature $T_b = 51.2°F$ (10.7°C). Because the temperature of the vapor refrigerant at the outlet T_o can be considered equal to the bulb temperature T_b, the degree of superheat is calculated as

$$\Delta T_{rl} = T_o - T_x = T_b - T_x = 51.2 - 40 = 11.2°F\ (6.2°C) \tag{10.32}$$

Liquid Charge in the Sensing Bulb

Straight and Limited Liquid Charge. When the sensing bulb is charged with an amount of liquid with the same properties as the refrigerant in the evaporator, such a charge is known as the *straight charge*. If there is only a limited amounted of liquid charge, it is known as the *limited liquid charge*. All the liquid vaporizes at a predetermined temperature, as shown in Fig. 10.19*a*. Any increase above this temperature results in only a small increase in pressure on the diaphragm because of the superheated vapor.

A sensing bulb filled with limited liquid charge limits the maximum operating temperature in the evaporator and, therefore, prevents the compressor motor from overloading.

Cross-charge. The temperature versus pressure difference $\Delta T - \Delta p$ of a sensing bulb with straight charge is shown in Fig. 10.19*b*. At a different evaporating temperature T_{ev}, $\Delta T_{bx} = T_b - T_x$ varies accordingly. If the liquid charge in the sensing bulb is different from the refrigerant used in the evaporator, it is known as cross-charge. The primary purpose of cross-charge is to provide a nearly constant degree of superheat at various evaporating temperatures, as shown in Fig. 10.19*c*. For a constant value of $p_b - p_x$, the degree of superheat at various evaporating temperatures is

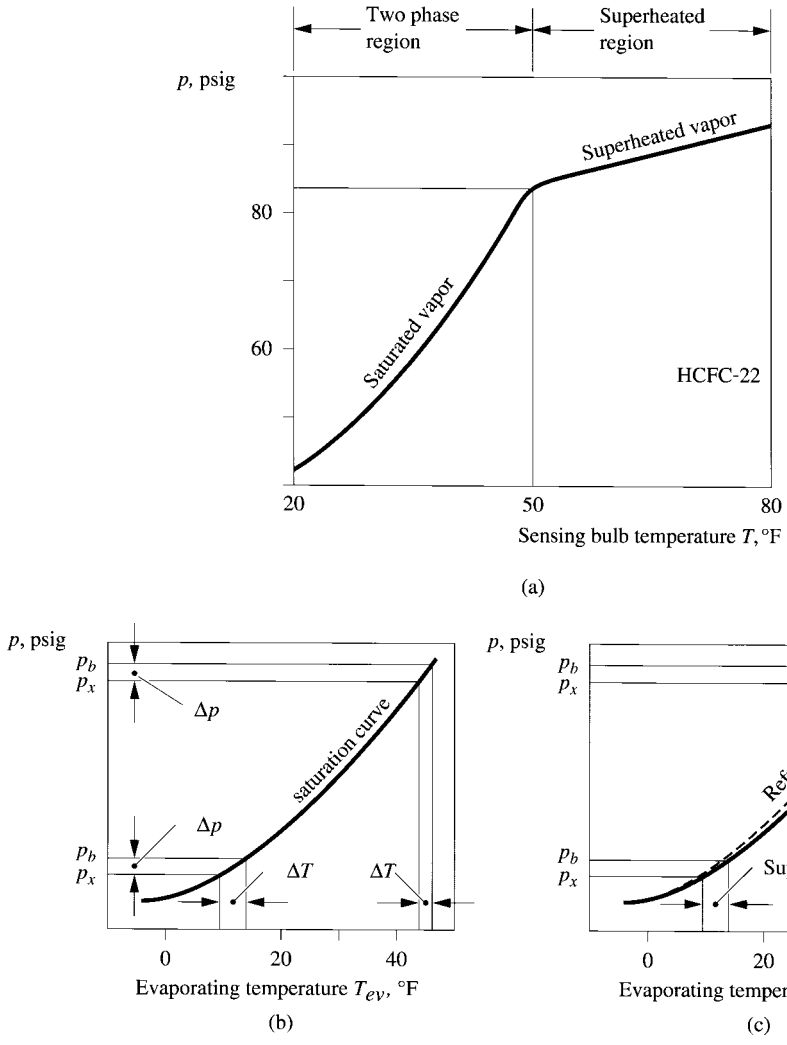


FIGURE 10.19 Thermostatic characteristics of sensing bulb: (*a*) limited liquid charge; (*b*) straight charge; (*c*) cross-charge.

approximately equal. This prevents the refrigerant from flooding back to the compressor at higher evaporating temperatures.

Hunting of Thermostatic Expansion Valve. Hunting is the quick cycling of maximum and minimum output of a controlled device. For the thermostatic expansion valve, hunting results from the over- or underfeeding of refrigerant to the evaporator. It causes temperature and pressure fluctuations and may reduce the capacity of the refrigeration system.

The time lag of refrigerant flow through the evaporator to the sensing bulb may cause a continuous overshooting of the valve position. The overfed refrigerant may form unevaporated slugs. These

slugs may chill the sensing bulb and suddenly reduce the refrigerant feed. The overshooting and sudden reduction cause hunting.

Hunting of the thermostatic expansion valve is determined by the following factors:

- *Valve size*. An oversized valve aggravates hunting.
- *Degree of superheat*. The smaller the degree of superheat, the greater the chance of hunting.
- *Sensing bulb charge*. Cross-charge tends to prevent hunting.
- *Location of sensing bulb*. A properly selected sensing bulb location often minimizes hunting. The sensing bulb should be located at the side of the horizontal section of the suction line immediately after the evaporator outlet, so that it does not sense the oil temperature or liquid refrigerant at the bottom of the tube.

Electric Expansion Valves

Electric, or more truly electronic or microprocessor-controlled, expansion valves can provide more sophisticated, effective, and energy-efficient refrigerant flow controls than thermostatic expansion valves. Currently, three types of electric expansion valves have become widely available: step motor valves, pulse-width-modulated valves, and analog valves.

Step Motor Valve. The step motor is an electronically controlled multiphase motor capable of moving continuously in forward or reverse direction, as shown in Fig. 10.20*a*. It can also be discretely positioned in increments of a fractional revolution. Step motors accept a series of step signals (such as 200/s). The rotational movement is transformed to linear movement required to open and close an expansion valve by means of a gearbox and a threaded driveshaft. The gearing is used to increase the torque. This allows the use of a small motor without overstressing it. One of the available step motor valves needs only 5.7 W of power. Dolin (1997) reported that a step motor valve may have 1532 steps of resolution in 0.125 in. (3.2 mm) of travel. Each step can change the flow rate of HCFC-22 by 0.002 lb/min or 3 Btu/h (0.001 kg/min or 0.88 W).

Pulse-Width-Modulated Valves. These are solenoid valves operated at on/off mode, i.e., either fully opened or fully closed, as shown in Fig. 10.20*b*. The resolution (the act, or capability of rendering the distinguishable parts of an object) of flow rate of refrigerant feed is indicated by the amount of time, the pulse width, or the duration of valve opening. For instance, a pulse-width-modulated valve may be pulsed every 5 s. If only a flow rate of 40 percent is required, the valve will hold open for 2 s and closed for 3 s.

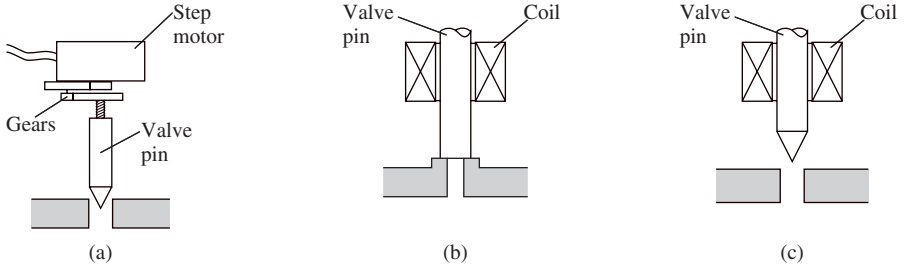


FIGURE 10.20 Electric expansion valves: (*a*) step motor valve; (*b*) pulse-width-modulated valve; (*c*) analog valve.

Liquid refrigerant flows at a velocity of about 100 fpm (0.5 m/s) in the liquid line. A suddenly closing action often causes severe vibration and water hammer. A dampening by forcing out the liquid refrigerant above or below the valve's plunger through a small bleed passage is an effective means of slowing the opening and closing.

Analog Valves. Instead of fully opening or closing, an analog valve can vary the strength of the coil's magnetic field, so that the valve pin (or plunger) can stay at various intermediate positions, as shown in Fig. 10.20*c*. Because the drive circuitry is more complicated than the digital output required for a pulse-width-modulated or a step motor valve, the performance, such as resolution of a analog valve may suffer compared with step motor and pulse-width-modulated valves.

Advantages of Electric Expansion Valves. Compared to the thermostatic expansion valves, the advantages of electric expansion valves are that they

- Provide a more precise temperature control
- Provide consistent superheat control under fluctuating head pressure
- Are able to be operated at low head pressure during lower ambient air temperature
- Are more energy-efficient

Of the three types of electric expansion valves (step motor, pulse-width-modulated, and analog valves), the performance of step motor valves has proved superior to the two others.

Float Valves

Float valves control the refrigerant feed to the evaporator to maintain a specific liquid level. The two types of float valves used in refrigerant control are high-side float valves and low-side float valves. A high-side float valve is located at the high pressure side of the refrigeration system, between the compressor discharge outlet and the inlet of the thermostatic expansion valve. High-side float valves control the flow rate of the refrigeration feed to the evaporator indirectly by maintaining a constant liquid level in the high-pressure float chamber.

Figure 10.21*a* shows a high-side float valve.The valve pin is linked with a float ball in such a way that any rise in the liquid level in the float chamber opens the valve and allows the liquid refrigerant to pass through the valve to the evaporator. A bypass tube provides a passage for noncondensable gas to escape from the float chamber to the evaporator.

The high-side float valve may be located above the condenser. In a sealed refrigerant circuit, the difference in refrigerant pressure before and after the high-side float valve forces the liquid refrigerant through the valve and into the evaporator.

Because the high-pressure float chamber only holds a small and fixed amount of liquid refrigerant, most of the liquid refrigerant in a reciprocating refrigeration system is stored in the evaporator. Therefore, control of refrigerant charge is essential. Excessive refrigerant charge may cause floodback. Insufficient refrigerant charge may lower system capacity. A high-side float valve can be used in direct-expansion or flooded evaporators.

The low-side float valve is located at the low-pressure side of the refrigeration system. The low-side float valve controls the liquid refrigerant feed to maintain a constant liquid level in the evaporator. Figure 10.21*b* shows a low-side float valve. As the liquid level in the evaporator drops, the float ball moves downward and opens the float valve wider so that more liquid refrigerant is fed to the evaporator. In small refrigeration systems, the float chamber is often placed directly inside the evaporator or in an accumulator, instead of a separate low-side float chamber. Operation of the low-side float valve may be continuous or intermittent. Control of the refrigerant charge is not as important for a low-side float valve as for a high-side float valve.

In evaporators with high evaporating rates or limited vapor passages, foaming refrigerant often raises the refrigerant level during operation. Therefore, the float ball should be placed in an

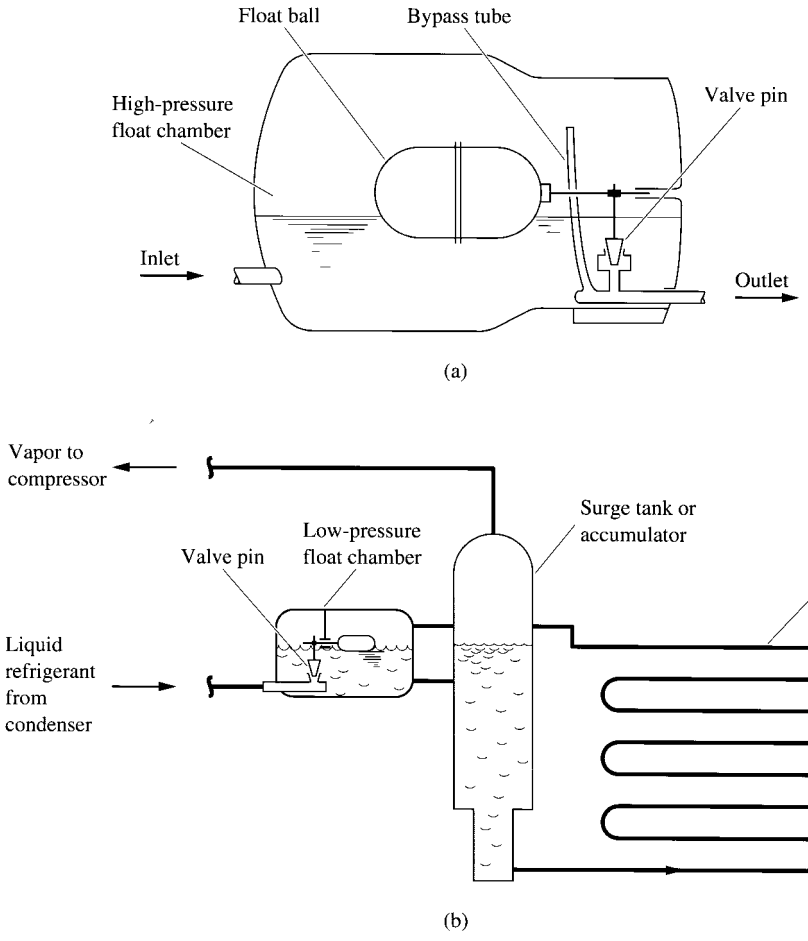


FIGURE 10.21 Float valves: (*a*) high-side float valve; (*b*) low-side float valve.

appropriate position to determine liquid levels accurately. Low-side float valves are usually used in flooded coils or evaporators.

Capillary Tube

In most room air conditioners and many heat pumps, a capillary tube is used as the throttling device. A capillary tube, also sometimes called a restrictor tube, is a fixed length of small-diameter tubing lying between the outlet of the condenser and the inlet of the evaporator. It reduces the pressure of the refrigerant from the high-pressure side to the low-pressure side of the system and controls the flow of refrigerant to the evaporator. The inside diameter of the capillary tube D_{in} is between 0.05 and 0.06 in. (1.27 and 1.52 mm) with a length L from 1 in. (25 mm) to several feet. For halocarbon refrigerants, it is usually made of copper. In recent years, there has been a tendency to use short capillary tubes having L/D_{in} between 3 and 20 in heat pumps. These short capillary tubes are often made of brass or copper alloy. A strainer is generally located before the

capillary tube to prevent clogging of the tube caused by impurities and foreign matter contained in the refrigerant.

In Aaron and Domanski (1990), tests and experiments for short capillary tubes showed that when the downstream pressure was lower than the saturated pressure of the entering refrigerant, the fluid inside the short tube had a metastable inner core of liquid with a surrounding two-phase annular ring. The flow showed a very weak dependence upon the downstream pressure and could be considered a nonideal chocked flow. The diameter of the short capillary tubes was the parameter that most strongly affected the mass flow rate of the refrigerant. A greater differential between the upstream and downstream pressure also had a significant effect on the mass flow rate. Chamfering the inlet of the short capillary tube showed only a minor influence on the rate of refrigerant flow. By using a short capillary tube as a throttling device, one can operate the cooling mode even at an outdoor temperature of 35°F (1.7°C) if necessary.

A capillary tube has a certain capability to balance the refrigerant flow when the refrigerant load changes. At a given condensing pressure, if the evaporating pressure is lower than normal due to a lower refrigeration load, the greater pressure difference causes a higher mass flow rate of refrigerant through the capillary tube. Because the refrigerant flow handled by the compressor is lower due to the low suction pressure, a decrease in the condensing pressure, and liquid level, and seal occurs in the condenser. Some vapor then passes through the capillary tube into the evaporator. All these reduce the mass flow rate of refrigerant flowing through the capillary tube and form a new balance.

When the evaporating pressure becomes higher because of a greater refrigeration load, more liquid refrigerant accumulates in the condenser. Meanwhile, the liquid charge in the evaporator is insufficient. The effects tend to reduce the heat-transfer area in the condenser and the wetted surface area in the evaporator. Consequently, the condensing pressure is raised, and the evaporating pressure is lowered. A greater high- and low-side pressure difference results in a higher refrigerant flow and forms a new balance.

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CHAPTER 11

REFRIGERATION SYSTEMS: RECIPROCATING, ROTARY, SCROLL, AND SCREW

11.1 RECIPROCATING REFRIGERATION SYSTEMS 11.2
- Types of Reciprocating Refrigeration Systems 11.2
- Real Cycle of a Single-Stage Reciprocating Refrigeration System 11.4

11.2 RECIPROCATING COMPRESSORS 11.5

11.3 RECIPROCATING REFRIGERATION SYSTEM COMPONENTS AND ACCESSORIES 11.5
- Compressor Components and Lubrication 11.5
- Accessories 11.8
- Service Valves 11.11

11.4 REFRIGERANT PIPING 11.12
- Copper Tubing 11.12
- Piping Design for Reciprocating Refrigeration Systems 11.13
- Size of Copper Tube, Refrigeration Load, and Pressure Drop 11.14
- Sizing Procedure 11.14
- Suction Line 11.15
- Discharge Line (Hot-Gas Line) 11.20
- Liquid Line 11.21

11.5 CAPACITY AND SAFETY CONTROLS 11.24
- Capacity Control 11.24
- Safety Controls 11.26

11.6 PERFORMANCE OF RECIPROCATING COMPRESSOR, EVAPORATOR, AND CONDENSER 11.29
- Performance of Reciprocating Compressor 11.29
- Performance of Evaporator 11.32
- Performance of Condenser 11.33

11.7 SYSTEM BALANCE 11.34
- Balance of Capacities of Selected Components 11.35

11.8 OPERATING CHARACTERISTICS OF AIR-COOLED, DIRECT-EXPANSION RECIPROCATING REFRIGERATION SYSTEM 11.36
- Operating Balance at Design Condition 11.36
- Pressure Characteristics 11.37
- Part-Load Operation Using a Cylinder Unloader 11.38
- Pump-Down Control 11.39
- Main Problems in Direct-Expansion Reciprocating Refrigeration Systems 11.40
- Minimum Performance and Equipment Selection 11.42

11.9 ROTARY REFRIGERATION SYSTEMS 11.42
- Rotary Compressor 11.42
- Main Components 11.43
- System Performance 11.43

11.10 SCROLL REFRIGERATION SYSTEMS 11.43
- Types of Scroll Refrigeration Systems 11.43
- Scroll Compressors 11.44
- Radial and Axial Compliance 11.44
- Compressor Performance 11.46
- Concentration Shift and Temperature Glide 11.46
- Circulating Concentration Shift and Its Effects on Performance 11.46
- Heat Exchanger Flow Configuration 11.47
- Capacity Control and Part-Load Performance 11.47
- System Characteristics 11.48
- Scroll Chillers 11.48

11.11 SCREW REFRIGERATION SYSTEMS 11.50
- Types of Screw Refrigeration System 11.50
- Screw Compressors 11.50
- Performance of Twin-Screw Compressor 11.52
- Oil Cooling 11.53
- Capacity Control 11.53
- Variable Volume Ratio 11.54
- Economizer 11.54
- Controls 11.54
- Minimum Performance 11.54
- Air-Cooled Screw Chillers 11.55

REFERENCES 11.56

11.1 RECIPROCATING REFRIGERATION SYSTEMS

Types of Reciprocating Refrigeration Systems

As discussed in Sec. 9.18, currently used reciprocating, or more truly, reciprocating vapor compression, refrigeration systems can be classified primarily into the following systems:

Air-Cooled, Reciprocating DX Cooler. Figure 11.1 shows an air-cooled, reciprocating DX cooler. In a DX system, a direct-expansion coil is used as the evaporator, and HCFC-22 and HFC-134a are the primary refrigerants used in new and retrofit projects. The air-cooled condenser is often combined with the reciprocating compressor(s) to form a unit called the *condensing unit*. The condensing unit is generally located outdoors. Either the DX coil is installed in a rooftop packaged unit consisting of fan(s), filters, condensing unit, and controls, mounted on the roof of the building; or it is a system component in a split or indoor packaged unit located indoors. In supermarkets and many industrial applications, DX coils are also installed in refrigerated display cases and food processing and food storage facilities. Air is forced through the DX coil by a fan for better heat transfer. DX systems are most widely used in reciprocating and scroll refrigerating systems.

Air-Cooled Reciprocating Chiller. This kind of reciprocating refrigeration system consists of a direct-expansion shell-and-tube liquid cooler as evaporator (as described in Sec. 10.4), an air-cooled condenser, single or multiple reciprocating compressor(s), accessories, and controls. The entire package is often called an air-cooled reciprocating chiller, and it is shown in Fig. 11.2. As in DX systems, HCFC-22 and HFC-134a are the primary refrigerants used in new and retrofit projects. In the shell-and-tube liquid cooler, the refrigerant evaporates inside the metal tubes, and

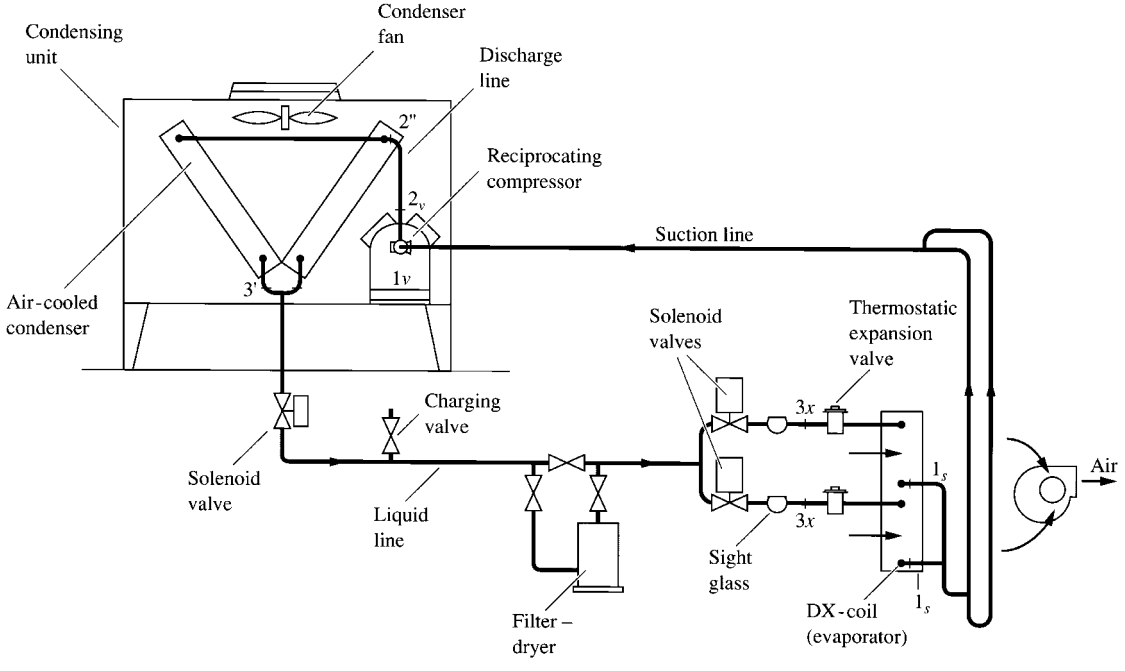


FIGURE 11.1 An air-cooled reciprocating DX cooler.

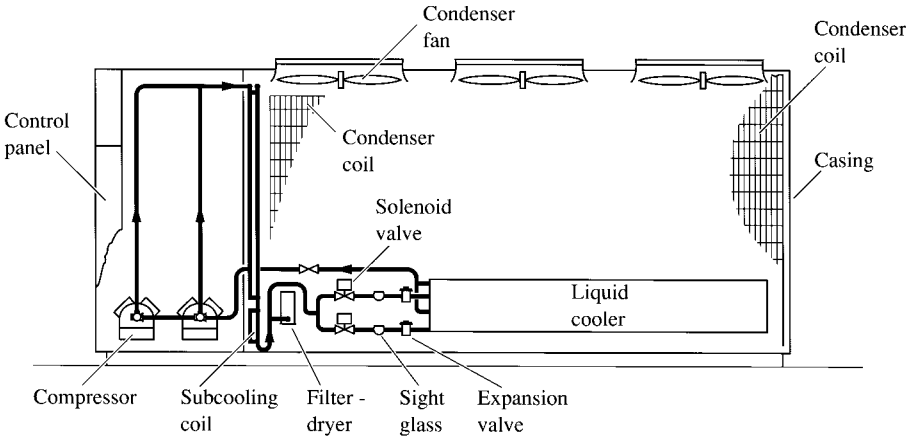


FIGURE 11.2 Air-cooled reciprocating chiller.

the chilled water fills the shell. All components and accessories of an air-cooled reciprocating chiller are enclosed in a casing made of steel sheets coated with corrosion-resistant paint.

In large air-cooled reciprocating chillers, air-cooled condenser coils are installed on both sides of the unit, and the subcooling coil is at the bottom, connected to the condensing coil through a liquid accumulator. The air-cooled reciprocating chiller is usually located on the rooftop of the building.

According to ARI Standard 550/590-98, air-cooled reciprocating chillers are rated the same as centrifugal chillers discussed in Sec. 13.2. In locations where water is scarce and expensive, air-cooled reciprocating chillers are often used in central hydronic air conditioning systems or for process cooling purposes in industrial applications with a cooling capacity less than 250 tons (879 kW).

Liquid Overfeed Reciprocating Refrigeration Systems. Liquid overfeed reciprocating refrigeration systems use a liquid overfeed cooler as an evaporator, as described in Sec. 10.4. Figure 11.3 shows a liquid overfeed reciprocating system. Ammonia R-717 and HCFC-22 are often used as the

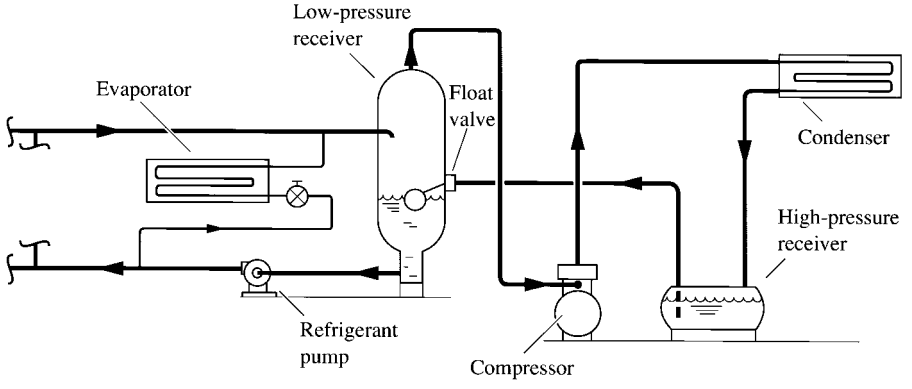


FIGURE 11.3 Liquid overfeed reciprocating refrigeration system.

refrigerants. An additional refrigerant pump is required to circulate the refrigerant to each evaporator at a flow rate several times greater than the actual evaporating rate. After the expansion valve, the refrigerant is throttled to the evaporating pressure. Although liquid refrigerant directly expands into vapor inside the evaporator's tubes and accumulates in the low-pressure receiver, it is not a dry-expansion evaporator. In a liquid overfeed system, a water-cooled condenser is often used.

Liquid overfeed refrigeration systems have the advantages of a higher heat-transfer coefficient of the wetted inner surface and simpler refrigerant flow control. However, they are large and heavy and are generally used for industrial refrigeration, food storage, and sometimes in ice storage systems.

Multistage Reciprocating Refrigeration Systems. As described in Chap. 9, a multistage reciprocating system could be a compound system or a cascade system. Multistage reciprocating refrigeration systems are used for industrial applications to provide refrigeration at low temperature, such as below $-10°$F.

Real Cycle of a Single-Stage Reciprocating Refrigeration System

In a reciprocating refrigeration system, the real refrigeration cycle differs from the ideal cycle because of the pressure drop of the refrigerant when it flows through the pipelines, valves, compressor passages, evaporator, and condenser. Also, the compression and expansion processes are not isentropic.

Figure 11.4 shows the real cycle of a single-stage reciprocating vapor compression refrigeration system on a *p-h* diagram. In a single-stage reciprocating refrigeration system, the pipeline from the outlet of the evaporator to the inlet of the reciprocating compressor is called the suction line. The suction line is shown by 1_s1_v in Figs. 11.1 and 11.4. The pipeline from the outlet of the reciprocating compressor to the inlet of the air-cooled condenser is called the discharge line, or hot-gas line. This is shown as $2_v2''$ in both figures. The pipeline from the outlet of the condenser to the inlet of the thermostatic expansion valve is called the *liquid line*. This is shown as $3'3_x$ in both figures.

In Fig. 11.4, the thick lines represent the real refrigeration cycle, and the thin lines the ideal cycle. Various line segments indicate the following refrigeration processes:

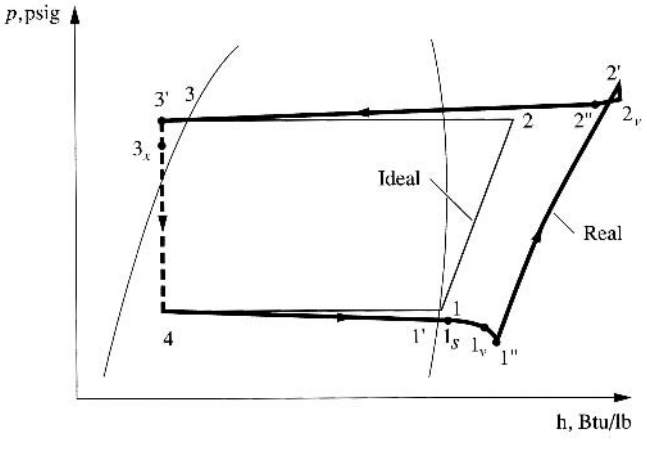


FIGURE 11.4 Real reciprocating vapor compression refrigeration cycle on *p-h* diagram.

$1\ 1_s$	superheat in evaporator
$1_s 1_v$	pressure drop in suction line
$1_v 1''$	pressure drop from suction valve and passage to cylinder
$1''2'$	compression process in cylinder
$2'2_v$	pressure drop from discharge valve and passage
$2_v 2''$	pressure drop in discharge line
$2''3'$	condensing and subcooling processes in condenser
$3'3_x$	pressure drop in liquid line
$3_x 4$	throttling process in expansion valve
$4\ 1_s$	evaporating process in evaporator

11.2 RECIPROCATING COMPRESSORS

In a reciprocating compressor, shown in Fig. 11.5, a single-acting piston in a cylinder is driven by a crankshaft via a connecting rod. At the top of the cylinder are a suction valve and a discharge valve. There are usually two, three, four, or six cylinders in a reciprocating compressor.

Vapor refrigerant is drawn through the suction valve into the cylinder until the piston reaches its lowest position. As the piston is forced upward by the crankshaft, it compresses the vapor refrigerant to a pressure slightly higher than the discharge pressure. Hot gas opens the discharge valve and discharges from the cylinder. The gaseous refrigerant in a reciprocating compressor is compressed by the change of internal volume of the compression chamber caused by the reciprocating motion of the piston in the cylinder.

The cooling capacity of a reciprocating compressor ranges from a fraction of a ton to 200 tons (1 to 3.5 kW to 700 kW). Refrigerant HCFC-22, HFC-134a, HFC-404A, HFC-407A, and HFC-407C in comfort and process air conditioning and R-717 (ammonia) in industrial applications are the currently used refrigerants in reciprocating compressors because of their zero ozone depletion factors, except HCFC-22 which will be restricted from the year 2004 due to its ozone depletion.

The maximum compression ratio p_{dis}/p_{suc} of a single-stage reciprocating compressor is about 7. Volumetric efficiency of a typical reciprocating compressor η_v decreases from 0.92 to 0.65 when p_{dis}/p_{suc} increases from 1 to 6, and the isentropic or compressor efficiency η_{isen} decreases from 0.83 to 0.75 when p_{dis}/p_{suc} increases from 4 to 6. Methods of capacity control during part-load operation include on/off, cylinder unloader, and hot-gas bypass controls.

Reciprocating compressor design is now in its mature stage. There is little room for significant improvement. Reciprocating compressors are still widely used in small and medium-size refrigeration systems. However, they will be gradually replaced by rotary, scroll, and screw compressors.

11.3 RECIPROCATING REFRIGERATION SYSTEM COMPONENTS AND ACCESSORIES

Compressor Components and Lubrication

The housing of a semihermetic reciprocating compressor, shown in Fig. 11.5, is the enclosure that contains the cylinder, pistons, crankshaft, main bearing, oil sump, crankcase, and hermetic motor, including both the rotor and the stator. The crankcase is the bottom part of the housing of the reciprocating compressor in which the crankshaft is housed.

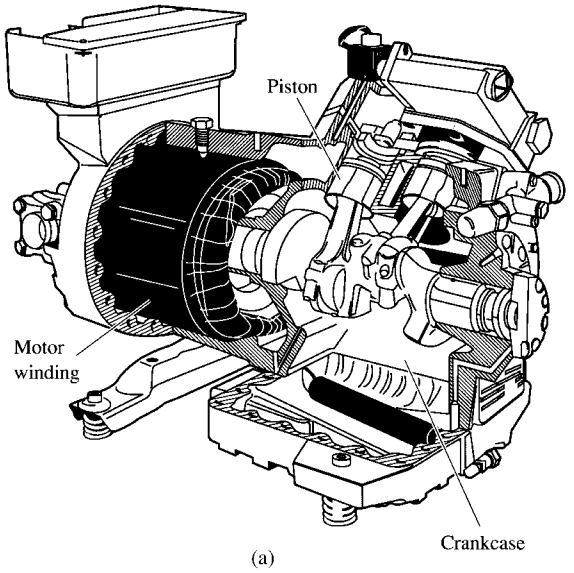


FIGURE 11.5 Schematic reciprocating compressor: (*a*) sectional view; (*b*) intake stroke; (*c*) compression and discharge stroke. (*Source: Trane Reciprocating Refrigeration and Refrigeration Compressors. Reprinted with permission.*)

Cylinder Block and Piston. In most compressors, the cylinder block is integrated with the crankcase and forms a single casting. The crankcase and the cylinder block are usually made of high-grade cast iron. In medium and large reciprocating compressors, premachined cylinder liners or sleeves are often inserted in the crankcase, and they can be replaced when worn. It costs far less to replace cylinder liners or sleeves than to replace the cylinder block.

The pistons are generally made of aluminum, aluminum alloy, or cast iron. For aluminum pistons, piston rings are often used to prevent leakage of gas refrigerant from the cylinder and to prevent oil from entering the cylinder. The upper piston rings are called *compression rings*, and the lower rings are called *oil rings*. For cast-iron pistons, a running clearance of 0.0004 in./in. (cm/cm) cylinder diameter provides an effective seal without oil rings.

Suction and Discharge Valves. The suction valve controls the vapor refrigerant entering the cylinder, and the discharge valve controls the hot gas discharging from the cylinder. Suction and discharge valves are usually made of high carbon-alloy steel or stainless steel. Spring-action ring valves are most extensively used in medium and large compressors. Ring valves are usually heat-treated to the resilience of spring steel and must be precisely ground to a perfectly flat surface. A deflection of 0.001 in. (0.025 mm) may cause leakage.

In addition, reed valves, either free-floating or clamped at one end, are used. A reed valve opens under discharged pressure and is closed by spring action. Reed valves are usually limited to small valve ports. A sufficient valve opening area must be provided to reduce pressure loss. A valve may open up to 0.01 in. (0.25 mm) during suction or discharge. The maximum velocity of HCFC-22 flowing through valves may reach 150 ft/s (45 m/s).

Oil Lubrication. Oil lubrication is necessary to form a fluid film separating the moving surfaces to protect them from wear and corrosion. Oil is also used as a coolant to carry heat away and cools the refrigerant. Oil provides an oil seal between the piston and cylinder and between the valve and valve plates. In refrigeration systems, mineral and synthetic oils are used for lubrication. HCFC-22 is partly oil-miscible so that the refrigerant-oil mixture can return to the compressor. HFCs and HFC blends are not miscible with mineral oil. Polyolester-based synthetic oil should be used for lubrication. Oil must be free of solids such as wax, must be chemically stable, and must have a suitable viscosity. In reciprocating compressors using HCFC-22 and halogenated refrigerants, a Saybolt seconds universal (SSU) viscosity between 150 and 300 is recommended by *ASHRAE Handbook 1994, Refrigeration Systems and Applications*. Kinematic viscosity of 1 ft^2/s (92,900 mm^2/s) is approximately equal to 4.3×10^5 SSU.

In medium and large reciprocating compressors, forced-feed lubrication through a positive displacement vane or gear oil pump is usually used. Lubricant is fed to the main bearing surfaces and crankshaft through oil passages. An oil mist formed by the crankshaft rotation lubricates the cylinder wall and piston pins. The oil pump is often mounted on one end of the crankshaft. An oil strainer submerged directly in the oil sump is connected to the suction intake of the oil pump. Because the vapor refrigerant in the crankcase is always at the suction pressure, the pressure lift of the oil pump is equal to the difference of the discharge pressure at the oil pump and the suction pressure.

In small reciprocating compressors, splash lubrication may be used. Splash lubrication uses the rotation of the crankshaft and the connecting rod to splash oil onto the bearing surfaces, cylinder walls, and other moving parts. Often, oil passages, oil rings, and other devices are used to increase oil splashing. An adequate oil level in the crankcase is necessary for satisfactory operation. Splash lubrication is simple. Its disadvantage is that bearing clearance must be larger than normal, which increases noise during operation.

For HCFC-22 and other halocarbon refrigerants, lubrication of suction and discharge valves is achieved because the refrigerant is miscible with the lubrication oil. Refrigerant piping design must accommodate the return of oil from evaporators and condensers. Oil should be separated from the suction vapor at the compressor and returned to the crankcase.

Crankcase Heater. When the reciprocating compressor is installed outdoors, refrigerants may migrate to the compressor from the indoor evaporator during the shutdown period of the refrigeration

system. This is mainly because the temperature and refrigerant pressure in the indoor evaporator are higher than in the outdoor compressor. If a considerable amount of refrigerant accumulates in the crankcase, it will dilute the lubrication oil. When the compressor starts to operate again, the foaming oil-refrigerant mixture may form a slug, resulting in a serious loss of oil from the crankcase. Slug entering the compression chamber may also damage the valves. For medium and large reciprocating compressors, a built-in crankcase heater is often installed to keep the oil temperature in the crankcase 15 to 25°F (8.3 to 13.9°C) higher than that in the rest of the system and thus prevent refrigerant migration.

Accessories

Liquid Receiver. If not all the refrigerant in the system can be condensed and stored in the condenser during the shutdown period, a high-pressure-side liquid receiver is needed to provide auxiliary refrigerant storage space, as shown in Fig. 11.6*a*. Consequently, if a shell-and-tube water-cooled condenser is equipped in a refrigeration system, a liquid receiver is not necessary. When an evaporative condenser is used, a liquid receiver is often installed to supplement its small storage capacity.

There are two types of receivers: through-type and surge-type receivers. In a through-type receiver, the liquid flows from the condenser to the receiver, and the pressure in the receiver is always lower than that at the condenser outlet. For a surge-type receiver, its purpose is to allow liquid to flow directly to the expansion valve and remain subcooled. Refer to *ASHRAE Handbook 1998*, *Refrigeration*, for details.

The liquid receiver is usually a cylindrical tank made of steel sheets. The top of the tank is connected to the liquid line from the condenser. It often has a liquid discharge line connected to the evaporator by a shutoff valve. A vent pipe combined with a fusible plug is mounted on the receiver to prevent the liquid receiver from bursting when exposed to high temperatures (as in a fire). In most installations, it is common to charge the refrigerant in the receiver 80 percent full during the shutdown period, and the remaining 20 percent is used to allow for liquid expansion.

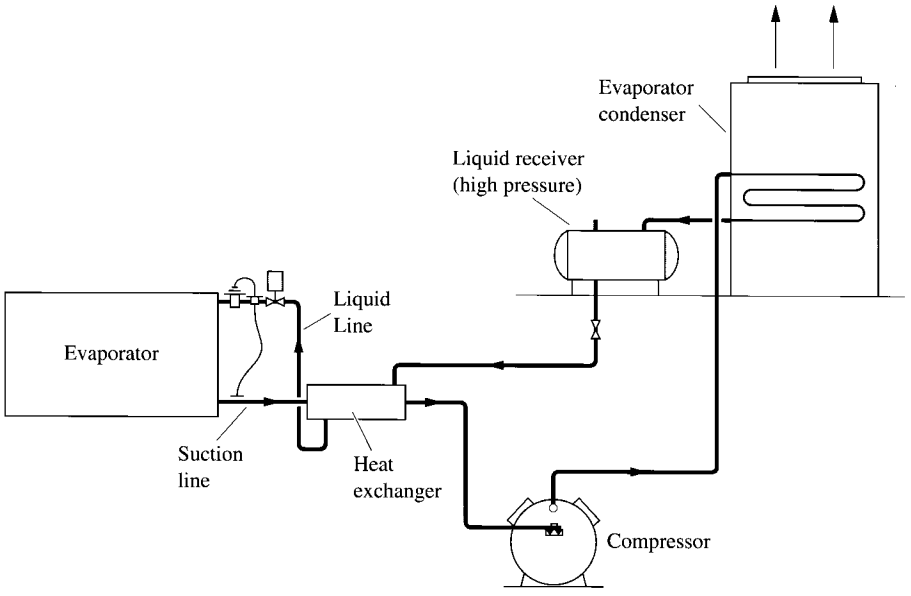
To provide space for refrigerant storage during shutdown and for liquid expansion, saturated pressure exists at the interface of the vapor and liquid refrigerant inside a through-type liquid receiver. No subcooling occurs at the outlet of the through-type liquid receiver unless a subcooling coil is provided after the receiver or a surge-type receiver is used. Because the subcooling liquid is not exposed to gaseous refrigerant in a surge-type liquid receiver, the liquid remains subcooled after flowing through it.

Liquid-Suction Heat Exchanger. A liquid-suction heat exchanger is usually mounted across the suction and liquid lines when the hot-gas discharge temperature of the refrigerant (such as HFC-134a) does not damage the lubrication oil or discharge valves. For ammonia and usually HCFC-22, the superheat of the suction vapor in the heat exchanger may raise the discharge temperature of the hot gas from the compressor to an unacceptable level, so liquid-suction heat exchangers are not used for these refrigerants.

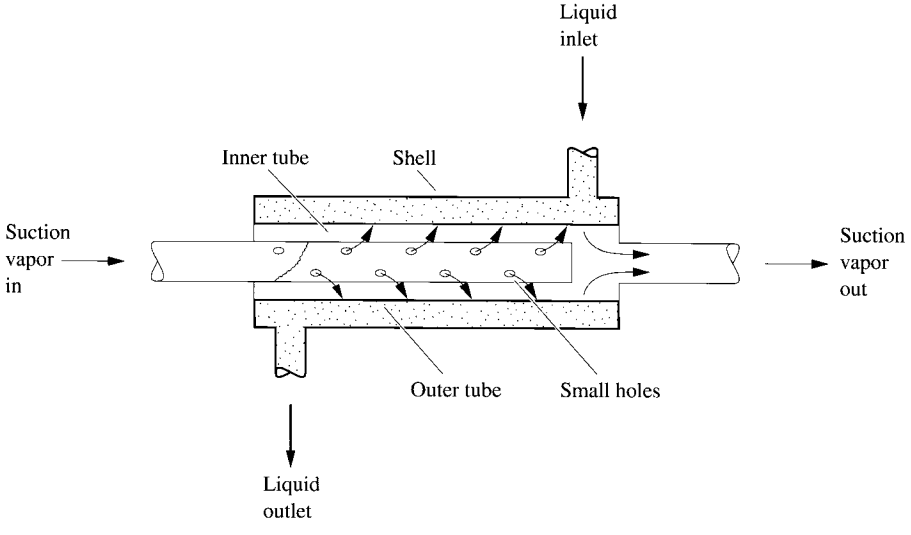
The purposes of a liquid-suction heat exchanger are as follows:

- To subcool the liquid refrigerant and increase the refrigerant effect
- To boil liquid refrigerant in the suction line and prevent liquid refrigerant from entering the reciprocating compressor
- To reduce the flash gas in the liquid line and ensure maximum capacity for the thermostatic expansion valve

Liquid-suction heat exchangers can be constructed as shell-and-coil, shell-and-tube, parallel combined liquid and suction lines, or tube-in-tube heat exchangers. In a typical shell-and-tube heat exchanger, as shown in Fig. 11.6, the hot liquid runs through the shell. The cold suction gas flows



(a)



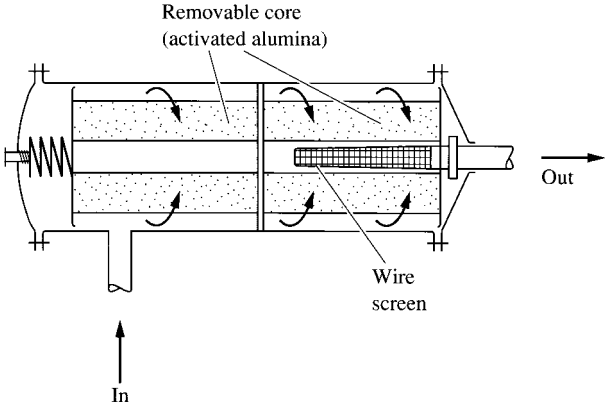
(b)

FIGURE 11.6 Liquid suction heat exchanger: (*a*) schematic diagram; (*b*) heat exchanger.

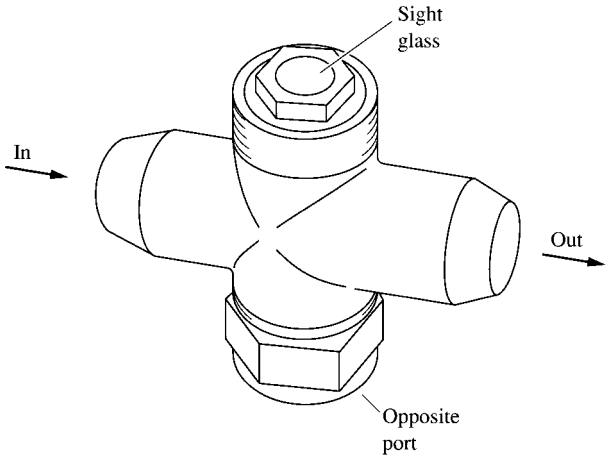
into the inner tube and penetrates the perforations at high velocity. Liquid refrigerant is subcooled between 7 and 15°F (3.9 and 8.3°C) in the heat exchanger where the heat is absorbed by the cold vapor. The pressure drop in the suction vapor should not exceed 2 psi when it flows through the heat exchanger.

Filter Dryer and Strainer. Moisture may freeze in the expansion valve, especially in a low-temperature system, and impair expansion valve operation. Foreign matter in the refrigeration system may also damage the expansion valve, control valves, and compressor. Moisture and foreign matter must be removed from the refrigeration system by means of a filter dryer and strainer.

Figure 11.7*a* shows a filter dryer. It contains a molded, porous core made of material with high moisture affinity, such as activated alumina, and acid-neutralizing agents to remove moisture and



(a)



(b)

FIGURE 11.7 (*a*) Filter dryer and (*b*) sight glass.

foreign matter from the refrigerant. The core should be replaced after a certain period of operation. The filter dryer is usually installed at the liquid line immediately before the expansion valve and shutoff valve.

The strainer contains a fine-mesh screen to remove copper fillings, dirt, and other debris that may be in the system at start-up. The construction of a strainer is similar to that of a filter dryer. Instead of a porous core, there is a fine-mesh wire screen. The strainer is often installed in the suction line just before the compressor.

Sight Glass. A sight glass is a small glass port used to observe the condition of refrigerant flow in the liquid line. Sight glasses are available with one or two ports. A double-port sight glass, as shown in Fig. 11.7*b*, is recommended because light enters the opposite port and makes observations easier. The sight glass is located just before the expansion valve. Bubbles seen through the sight glass indicate the presence of the flash gas instead of liquid refrigerant. The presence of flash gas always indicates that the evaporator's capacity is reduced because of a shortage of refrigerant or insufficient subcooling.

A moisture-detecting chemical is built into the sight chamber behind the sight glass. If excessive moisture is present in HCFC-22, the chemical turns pink. If the moisture is below the safe limit, the color is green.

In addition to these accessories, there are pressure gauges to indicate suction pressure, discharge pressure, and oil pressure. A liquid level indicator with gauge glasses is used to observe the liquid level in the receiver. It is often installed on the receiver or on the liquid line. A hot gas muffler may be used to smooth the pulsation of hot gas discharged from the compressor. The muffler should be located as close to the compressor as possible.

Most modern reciprocating compressors minimize oil pumping. Except in systems where oil removal is necessary, as in low-temperature systems, or for refrigerant that is not oil-miscible, oil separators are seldom used.

Service Valves

Solenoid Valves. Solenoid valves are widely used in the suction, discharge, and liquid lines, and in places where equipment and components are isolated for maintenance and repair. They are often automatically controlled by the direct digital control (DDC) and other controls.

A solenid valve in refrigerant flow control is shown in Fig. 11.8*a*.When the electromagnetic or solenoid coil is energized, the plunger and the attached valve pin are lifted upward to open the valve port fully. The valve is closed by gravity or spring action. For medium-size valves, a pilot-operated solenoid valve is often used. In this type of valve, the solenoid coil opens a pilot port, which produces a pressure imbalance across the diaphragm, forces it upward, and opens the main port. When the solenoid coil deenergizes, the plunger closes the pilot port. The pressure across the diaphragm is balanced through an equalizer hole.

Manual Shutoff Valves. These valves are used to shut off or fully open the refrigerant flow manually. Figure 11.7*b* shows a manual shutoff angle valve.

Most manual shutoff valves have packing around the stem. They also include back seat construction: The valve seat closes against a second seat to prevent refrigerant leakage through the valve stem and back seat port.

Pressure Relief Valves. In refrigeration systems, refrigerant pressure may rise to levels that could result in an explosion in the event of fire, malfunction, or shutdown. As in water systems, pressure relief valves in refrigeration systems are used to relieve the refrigerant pressure when it exceeds a predetermined limit.

A typical sping-loaded pressure relief valve is shown in Fig. 11.8*c*. It is usually mounted on the liquid line or on the liquid receiver. The pressure required to open the relief valve is adjusted by

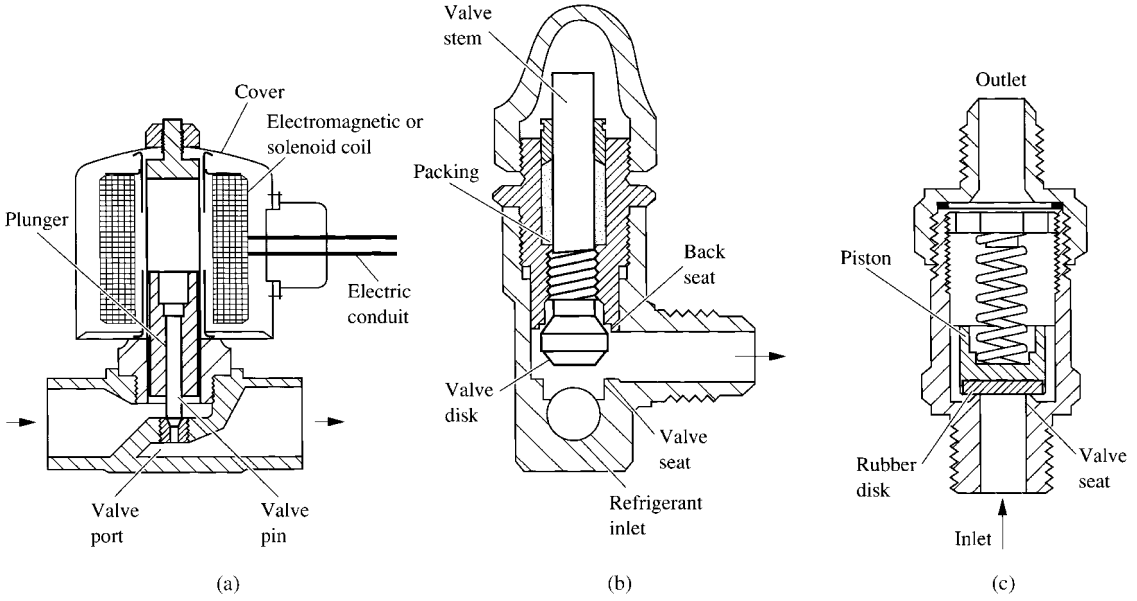


FIGURE 11.8 Service valves: (*a*) solenoid valve; (*b*) manual shutoff valve (angle type); (*c*) pressure relief valve.

the spring. Once it is set, the spring is sealed for protection. When the pressure of the refrigerant exerted on the rubber disk is greater than the spring load, the valve port opens and the refrigerant pressure is relieved through the upper outlet.

Refrigerant Charge Valve. A refrigerant charge valve is a manually operated shutoff valve used only during refrigerant charge. It is often mounted on the liquid line at the outlet of the condenser or on the outlet side of the receiver.

Check valves for refrigerant flow control are similar in construction and operation to the water valves described in Chap. 7.

11.4 REFRIGERANT PIPING

Copper Tubing

Refrigerant pipes transport refrigerant through the compressor, condenser, expansion valve, and evaporator to provide the required refrigeration effect. For halocarbon refrigerants, the refrigerant pipes are usually made of copper. However, if ammonia is used as the refrigerant, the pipes are always made of steel.

As listed in Table 11.1, three types of copper tubing are used for refrigerant piping. Type K is heavy-duty, and type M is lightweight. Type L is the standard copper tubing most widely used in refrigeration systems. Copper tubing installed in refrigeration systems should be entirely free of dirt, scale, and oxides. The open ends of clean new tubes should be capped to keep contaminants out.

TABLE 11.1 Correction Factors for Suction, Discharge, and Liquid Lines for HCFC-22 or T_{suc} Other than 40°F and T_{con} Other than 105°F

	Correction factors for T_{suc} other than 40°F							
T_{suc}, °F	50	40	30	20	10	0	−10	−20
$F_{suc,s}$	1.20	1.00	0.84	0.70	0.58	0.47	0.38	0.30
$F_{dis,s}$	1.01	1.00	0.99	0.97	0.96	0.95	0.93	0.91
	Correction factors for T_{con} other than 105°F							
T_{con}, °F	80	90	100	105	110	120	130	140
$F_{suc,c}$	1.11	1.07	1.03	1.00	0.97	0.90	0.86	0.80
$F_{dis,c}$	0.79	0.88	0.95	1.00	1.04	1.10	1.18	1.26
$F_{liq,c}$	1.12	1.07	1.02	1.00	0.98	0.93	0.88	0.83

Note: $F_{liq,s} = F_{dis,s}$

Piping Design for Reciprocating Refrigeration Systems

Packaged units with factory-assembled and -integrated refrigeration systems have become more popular, and air-cooled reciprocating packaged units or chillers contain a reciprocating compressor, liquid coolers, air-cooled condensers, expansion valves, and controls all in one package. However, many field built-up or split systems need proper refrigeration piping design to provide adequate cooling for air conditioning and industrial applications.

A proper refrigeration piping design must satisfy the following requirements:

- Transport the required amount of refrigerant to the evaporator, compressors, condenser, and throttling devices.
- Provide optimum pressure drop in the refrigeration lines, i.e., the most economical maximum pressure drop in the suction, discharge, and liquid lines. Current practice limits the maximum pressure drop in suction, discharge, and liquid lines corresponding to a change of saturated temperature ΔT_{sat} as follows:

	ΔT_{sat}, °F	HCFC-22 pressure drop, psi (kPa)
Suction line	2	2.91 (20) at T_{suc} = 40°F (4.4°C)
Discharge line	1	3.05 (21) at T_{con} = 105°F (40.6°C)
Liquid line	1	3.05 (21) at T_{con} = 105°F (40.6°C)

- For halocarbon refrigeration systems, oil should be miscible with liquid refrigerant, and oil entrained in the refrigerant should be returned to the crankcase of the compressor to maintain an adequate level for proper lubrication. Refrigerant piping design must accommodate the flow of both refrigerant and oil. Oil does not mix with gaseous refrigerant. Transportation of entrained oil in hot gas or vapor refrigerant is accomplished by ensuring a minimum refrigerant velocity not less than 500 fpm (2.5 m/s) in horizontal refrigerant lines and not less than 1000 fpm (5 m/s) in vertical refrigerant risers. Horizontal refrigerant lines must be pitched in the direction of refrigerant flow.
- Flashing of liquid refrigerant must not occur before the expansion valve. Adequate subcooling should be provided to offset the pressure drop of the liquid line and its accessories.
- Piping design should be configured to prevent liquid slugging, in which liquid refrigerant, oil, or a combination of both floods back to the compressor.
- Maintain a clean and dry refrigeration piping system.

Size of Copper Tube, Refrigeration Load, and Pressure Drop

During the sizing of refrigerant piping, the system refrigeration load Q_{rl}, in tons of refrigeration, is a known value. Because the refrigerant velocity $v = \dot{m}_r/(\rho A_i)$, the refrigerant mass flow rate $\dot{m}_r = Q_{rl}/\Delta h$ and the inner surface area of the copper tubing $A_i = \pi D_i^2/4$. Here ρ represents the density of the refrigerant, and Δh is the enthalpy difference of the refrigerant leaving and entering the evaporator. For a specific size of copper tubing, the relationship between the outside diameter D and inside diameter D_i is a fixed value. From the Darcey-Weisbach equation, the pressure drop in the refrigerant piping can be expressed as

$$D = K\left(\frac{8}{\pi^2 g_c}\frac{fLQ_{rl}^2}{\rho(\Delta h)^2\Delta p}\right)^{1/5} = C_{\text{pipe}}\left[\frac{fLQ_{rl}^2}{\rho(\Delta h)^2\Delta p}\right]^{1/5} \tag{11.1}$$

where g_c = dimensional constant, 32.2 $\text{lb}_\text{m} \cdot \text{ft}/\text{lb}_\text{f} \cdot \text{s}^2$
L = length of piping, ft (m)
Q_{rl} = refrigeration load, Btu/min (W)
Δh = enthalpy difference of refrigerant, Btu/lb (kJ/kg)
Δp = pressure loss, $\text{lb}_\text{f}/\text{ft}^2$ (kPa)

In Eq. (11.1), constant $C_{\text{pipe}} = K\,[8/(\pi^2 g_c)]^{1/5}$. The friction factor of refrigerant f can be calculated by the Colebrook equation and is covered in Chap. 18. If the absolute roughness of the copper tube is taken as 0.000005 ft (0.0015 mm), f has only a minor influence on the outside diameter of the copper tubing D. Constant K takes into account the difference between D and D_i for a specific size of copper tubing.

The size or outside diameter of the copper tubing D is

- Directly proportional to the length of the refrigeration pipe L to the $^1/_5$ power
- Directly proportional to refrigeration load Q_{rl} to the $^2/_5$ power
- Inversely proportional to the density of the suction vapor or hot gas ρ to the $^1/_5$ power, which is affected by suction and condensing temperature, respectively
- Inversely proportional to the enthalpy difference Δh (which is closely related to the suction temperature T_{suc} and condensing temperature T_{con}) to the $^2/_5$ power
- Inversely proportional to the maximum allowable pressure drop Δp to the $^1/_5$ power

Sizing Procedure

Refrigerant piping design and size should be determined as follows:

1. Make an optimum refrigerant piping layout, and measure the length of the piping.
2. Find the correction factor of the refrigeration load from Table 11.1, based on the actual suction and condensing temperatures.
3. Estimate the equivalent length of refrigerant piping including the pipe fittings and accessories L_{eq}, ft (m). It is usually 1.5 to 5 times the measured straight length of the piping, depending on the number of fittings and accessories.
4. Based on the corrected refrigeration load and L_{eq}, determine the tentative diameter of the copper tubing from refrigeration load versus equivalent length Q_{rl}-L_{eq} charts (see Figs. 11.10, 11.11, and 11.12). All suction, discharge, and liquid line Q_{rl}-L_{eq} are plotted based on the data given in *ASHRAE Handbook 1998*, *Refrigeration*.

5. Find the equivalent length of the pipe fittings and accessories listed in Tables 11.2 and 11.3, based on the tentative diameter of the copper tubing. Recalculate the equivalent length of the refrigeration piping including the pipe fittings and accessories.
6. Check the size of the copper tubing from the Q_{rl}-L_{eq} charts based on the calculated L_{eq}. If it is equal to the tentative value, the tentative diameter is the required diameter. If not, adjust the tentative diameter.
7. Determine whether the size of the refrigeration pipe meets the oil entrainment requirement when the refrigeration load is reduced to its minimum load at part-load operation. If it does not meet the requirement, add a double riser.

Suction Line

The suction line is the most critical refrigerant line in the piping design. Vapor refrigerant and entrained oil in the evaporator should return to the compressor free of liquid slugs.

Oil Trap and Piping Pitch. In the suction line, an oil trap should be installed at the bottom of a single riser, as shown in Fig. 11.9*a*. The sensing bulb of the thermostatic expansion valve is often

TABLE 11.2 Pressure Drop of Valves Expressed in Equivalent Length of Straight Pipe, in Feet, for Refrigeration Systems *(Screwed, Welded, Flanged, and Flared Connections) Losses are for all valves in fully open position.*

Nominal pipe or tube size, in.	Globe*	60°-*Y*	45°-*Y*	Angle*	Gate†	Swing check‡	Lift check
3/8	17	8	6	6	0.6	5	
1/2	18	9	7	7	0.7	6	Globe and
3/4	22	11	9	9	0.9	8	vertical lift
1	29	15	12	12	1.0	10	same as
1 1/4	38	20	15	15	1.5	14	globe
1 1/2	43	24	18	18	1.8	16	valves§
2	55	30	24	24	2.3	20	
2 1/2	69	35	29	29	2.8	25	
3	84	43	35	35	3.2	30	
3 1/2	100	50	41	41	4.0	35	
4	120	58	47	47	4.5	40	
5	140	71	58	58	6	50	
6	170	88	70	70	7	60	
8	220	115	85	85	9	80	
10	280	145	105	105	12	100	Angle lift
12	320	165	130	130	13	120	same as
14	360	185	155	155	15	135	angle
16	410	210	180	180	17	150	valve
18	460	240	200	200	19	165	
20	520	275	235	235	22	200	
24	610	320	265	265	25	240	

*These losses do not apply to valves with needle point type seats.

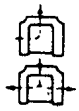
†Regular and short pattern plug cock valves, when fully open, have same loss as gate valve. For valve losses of short pattern plug cocks above 6 in. check manufacturer.

‡Losses also apply to the in-line, ball type check valve.

§For *Y* pattern globe lift check valve with seat approximately equal to the nominal pipe diameter, use values of 60° − *Y* valve for loss.

Source*: *ASHRAE Handbook 1990, Refrigeration Systems and Applications. Reprinted with permission.

TABLE 11.3 Fitting Losses in Equivalent Feet of Pipe
(Screwed, Welded, Flanged, Flared, and Brazed Connections)

Nominal pipe or tube size, in.	Smooth bend elbows						Smooth bend tees			
	90° Std*	90° Long-radius†	90° Street*	45° Std*	45° Street*	180° Std*	Flow-through branch	Straight-through flow: No reduction	Straight-through flow: Reduced 1/4	Straight-through flow: Reduced 1/2
										
3/8	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4
1/2	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
3/4	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
1	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.2	2.6
1 1/4	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
1 1/2	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0
2	5.0	3.3	8.2	2.6	4.5	8.2	10.0	3.3	4.7	5.0
2 1/2	6.0	4.1	10.0	3.2	5.2	10.0	12.0	4.1	5.6	6.0
3	7.5	5.0	12.0	4.0	6.4	12.0	15.0	5.0	7.0	7.5
3 1/2	9.0	5.9	15.0	4.7	7.3	15.0	18.0	5.9	8.0	9.0
4	10.0	6.7	17.0	5.2	8.5	17.0	21.0	6.7	9.0	10.0
5	13.0	8.2	21.0	6.5	11.0	21.0	25.0	8.2	12.0	13.0
6	16.0	10.0	25.0	7.9	13.0	25.0	30.0	10.0	14.0	16.0
8	20.0	13.0	—	10.0	—	33.0	40.0	13.0	18.0	20.0
10	25.0	16.0	—	13.0	—	42.0	50.0	16.0	23.0	25.0
12	30.0	19.0	—	16.0	—	50.0	60.0	19.0	26.0	30.0
14	34.0	23.0	—	18.0	—	55.0	68.0	23.0	30.0	34.0
16	38.0	26.0	—	20.0	—	62.0	78.0	26.0	35.0	38.0
18	42.0	29.0	—	23.0	—	70.0	85.0	29.0	40.0	42.0
20	50.0	33.0	—	26.0	—	81.0	100.0	33.0	44.0	50.0
24	60.0	40.0	—	30.0	—	94.0	115.0	40.0	50.0	60.0

*R/D approximately equal to 1.
†R/D approximately equal to 1.5.
***Source*: *ASHRAE Handbook 1998, Refrigeration Systems and Applications*.** Reprinted with permission.

installed between the coil outlet and the oil trap. This oil trap helps to prevent liquid accumulation under the sensing bulb while the compressor is turned off. The accumulated liquid under the sensing bulb causes abnormal operation of the expansion valve when the compressor is started again. An oil trap consists of two elbows connected with a short stub of pipe.

If the evaporator is located above the compressor, the horizontal section of suction line should be pitched toward the compressor at least 1/2 in. each 10 ft (13 mm each 3 m) of pipe length.

Double Riser. When the compressor is located above the evaporator, if the compressor is equipped with a capacity control or unloading mechanism to reduce the refrigeration load Q_{rl} so that the minimum Q_{rl} in the vertical riser is less than the value listed in Tables 11.4 and 11.5, a double riser is needed instead of a single vertical riser, as shown in Fig. 11.9*b*.

A double riser consists of a small riser, a large riser, and a oil trap that connects the lower ends of the risers. During full-load operation, the vapor velocity in both risers is high enough to entrain oil with the gaseous refrigerant, and there is no oil in the oil trap. However, when the Q_{rl} and vapor

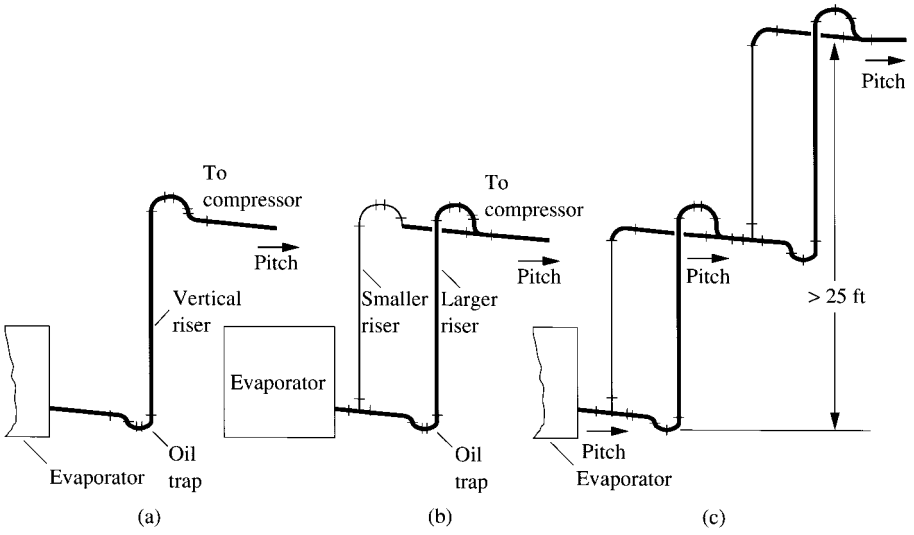


FIGURE 11.9 (*a*) Single riser, (*b*). double riser, and (*c*) additional oil trap.

velocity drop so that oil is no longer entrained in the large riser at part-load operation because of cylinder unloading or on/off control, oil accumulates in the oil trap. This oil blocks the vapor passage to the large riser. The vapor velocity in the small riser therefore increases to a velocity high enough to carry the entrained oil to the compressor.

If the vertical riser has a rise greater than 25 ft (7.5 m), an additional oil trap for each 25-ft (7.5-m) rise is required. Thus, the upper and lower sections are drained separately, and oil leaves the traps in quantities that will not damage the compressor.

Maximum Pressure Drop. In a suction line using HCFC-22 as the refrigerant, a pressure drop equivalent to 2°F (1.1°C) change in saturation temperature between 40 and 50°F (4.4 and 10°C) is about 3 psi (21 kPa). When HFC-134a is used as the refrigerant, a 2°F (1.1°C) equivalent is about 2 psi (14 kPa); and if ammonia R-717 is used as the refrigerant, a 1°F (0.56°C) equivalent is about 1.5 psi (7 kPa).

Suction Line Sizing Chart. Figure 11.10 shows the Q_{rl}-L_{eq} suction line sizing chart. The abscissa of this chart is the refrigeration load Q_{rl} that the evaporated refrigerant in the evaporator can provide at a corresponding mass flow rate, in tons of refrigeration, at 40°F (4.4°C) suction and 105°F (40.6°C) condensing temperature. The ordinate is the equivalent pipe length L_{eq}, ft (m). Table 11.4 indicates the minimum refrigeration load in the suction riser so that the suction vapor velocity v_{suc} inside the copper tubing can still carry the oil.

Suction temperature is the saturation temperature of the refrigerant at the inlet of the compressor. It is usually 2°F (1.1°C) lower than T_x in the evaporator or the DX coil, depending on the pressure drop of the suction line. When the suction temperature $T_{suc} \neq 40$°F (4.4°C) and the condensing temperature $T_{con} \neq 105$°F (40.6°C), the corrected refrigeration load $Q_{rl,c}$, in tons, for sizing the suction line can be found by dividing Q_{rl} by correction factors $F_{suc,s}$ and $F_{suc,c}$, as listed in Table 11.1, that is,

$$Q_{rl,c} = \frac{Q_{rl}}{F_{suc,s}F_{suc,c}} \tag{11.2}$$

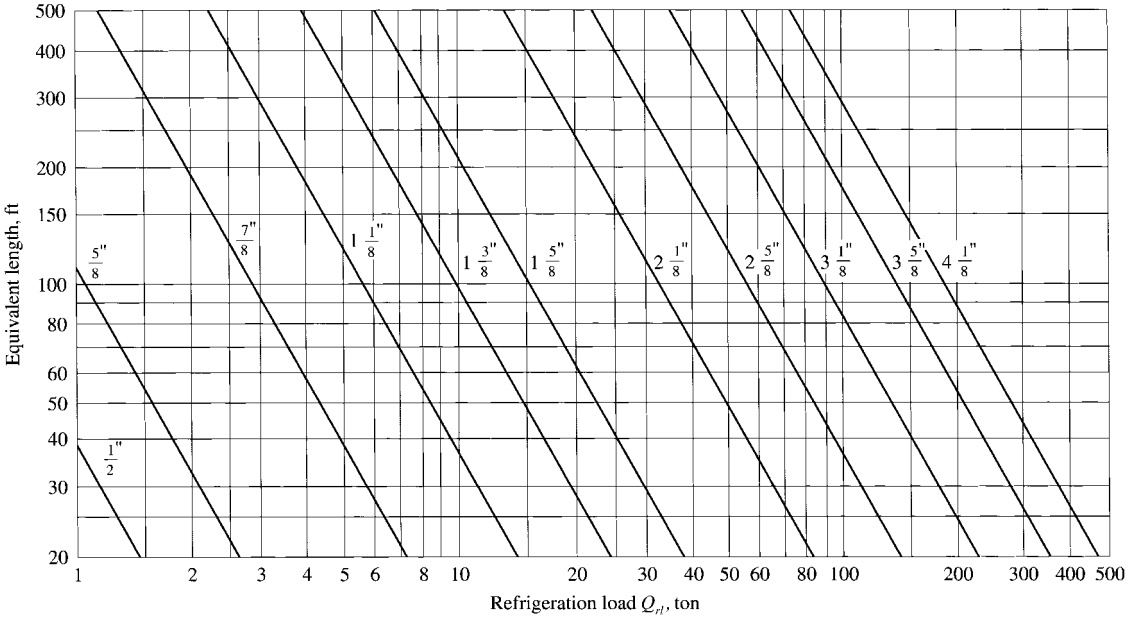


FIGURE 11.10 Suction line sizing Q_{rl}-L_{eq} chart for HCFC-22 at 40°F suction temperature and 105°F condensing temperature (type L copper tube, outside diameter, 2°F change in saturated temperature).

where $F_{suc,s}$ = correction factor for refrigeration load in suction line when $T_{suc} \neq 40$°F (4.4°C)
$F_{suc,c}$ = correction factor for refrigeration load in suction line when $T_{con} \neq 105$°F (40.6°C)

From Eq. (9.24), refrigeration load Q_{rl}, in tons, is given as

$$Q_{rl} = \dot{m}_r(h_{lv} - h_{en}) = \dot{V}_p \rho \eta_v (h_{lv} - h_{en})$$

where ρ = density of suction vapor, lb/ft^3
η_v = volumetric efficiency

Here, $\dot{V}_p$ represents the piston displacement, which is discussed later in this chapter. It is a fixed value for a specific compressor at a certain speed. Volumetric efficiency η_v is affected slightly by the compression ratio p_{dis}/p_{suc}. Therefore, correction factor $F_{suc,s}$ for the suction line is a combined index that takes into account the effect of $\rho(h_{lv}$-$h_{en})$ when $T_{suc} \neq 40$°F (4.4°C). If $T_{suc} <$ 40°F (4.4°C), then $F_{suc,s} < 1$, and $Q_{rl,c} > Q_{rl}$. The correction factor $F_{suc,c}$ for the suction line takes into account the effect of h_{lv} - h_{en} when $T_{con} \neq 105$°F. If $T_{con} < 105$°F, then $F_{suc,c} > 1$ and $Q_{rl,c} < Q_{rl}$.

Suction Line Sizing. Consider the size of the suction line of a reciprocating refrigeration system in a rooftop packaged unit with a refrigeration load of 40 tons (141 kW) at 45°F (7.2°C) suction temperature and 115°F (46.1°C) condensing temperature with a subcooling of 10°F (5.6°C). There are two refrigeration circuits and two compressors, each with a capacity of 20 tons (70 kW). Each compressor can be unloaded to 50 percent of its full-load capacity.

The suction line has a total pipe length, including horizontal lines and vertical riser of 3 ft (0.9 m), of 30 measured ft (9 m). There is also a manual shutoff angle valve and 10 short-radius elbows.

TABLE 11.4 Minimum Refrigeration Load for Oil Entrainment up Suction Risers of Type L Copper Tubing, in Tons

			Pipe OD, in.											
			0.500	0.625	0.750	0.875	1.123	1.375	1.625	2.125	2.625	3.125	3.625	4.125
	Saturated temperature,	Suction gas temperature,	Area, in.^{2}											
Refrigerant	°F	°F	0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
HCFC-22	−40.0	−30.0	0.067	0.119	0.197	0.298	0.580	0.981	1.52	3.03	5.20	8.12	11.8	16.4
		−10.0	0.065	0.117	0.194	0.292	0.570	0.963	1.49	2.97	5.11	7.97	11.6	16.1
		10.0	0.066	0.118	0.195	0.295	0.575	0.972	1.50	3.00	5.15	8.04	11.7	16.3
	−20.0	−10.0	0.087	0.156	0.258	0.389	0.758	1.28	1.98	3.96	6.80	10.6	15.5	21.5
		10.0	0.085	0.153	0.253	0.362	0.744	1.26	1.95	3.88	6.67	10.4	15.2	21.1
		30.0	0.086	0.154	0.254	0.383	0.747	1.26	1.95	3.90	6.69	10.4	15.2	21.1
	0.0	10.0	0.111	0.199	0.328	0.496	0.986	1.63	2.53	5.04	8.66	13.5	19.7	27.4
		30.0	0.108	0.194	0.320	0.484	0.942	1.59	2.46	4.92	8.45	13.2	19.2	26.7
		50.0	0.109	0.195	0.322	0.486	0.946	1.60	2.47	4.94	8.48	13.2	19.3	26.8
	20.0	30.0	0.136	0.244	0.403	0.608	1.18	2.00	3.10	6.18	10.6	16.6	24.2	33.5
		50.0	0.135	0.242	0.399	0.603	1.17	1.99	3.07	6.13	10.5	16.4	24.0	33.3
		70.0	0.135	0.242	0.400	0.605	1.18	1.99	3.08	6.15	10.6	16.5	24.0	33.3
	40.0	50.0	0.167	0.300	0.495	0.748	1.46	2.46	3.81	7.60	13.1	20.4	29.7	41.3
		70.0	0.165	0.296	0.488	0.737	1.44	2.43	3.75	7.49	12.9	20.1	29.3	40.7
		90.0	0.165	0.296	0.488	0.738	1.44	2.43	3.76	7.50	12.9	20.1	29.3	40.7

Note: The capacity in tons is based on 90°F liquid temperature and superheat as indicated by the listed temperatures. For other liquid line temperatures, use correction factors in the table below.

	Liquid temperature, °F								
Refrigerant	50	60	70	80	100	110	120	130	140
HCFC-22	1.17	1.14	1.10	1.06	0.98	0.94	0.89	0.85	0.80

Source: *ASHRAE Handbook 1998, Refrigeration Systems and Applications.*
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1. From Table 11.1, for a suction temperature of 45°F (7.2°C) and a condensing temperature of 115°F (46.1°C), $F_{suc,s}$ is 1.10 and $F_{suc,c} = 0.93$. The corrected refrigeration load Q_{rl} for the suction line is therefore $20/(1.1 \times 0.93) = 19.6$ tons (68.7 kW).
2. Assume an equivalent length of the suction line L_{eq} of 90 ft (27 m). From Fig. 11.10, for a refrigeration load of 19.7 tons and $L_{eq} = 90$ ft (27 m), the next-higher value of the tentative outside diameter of the suction line is $2\frac{1}{8}$ in. (54 mm), a nominal diameter of 2 in. (50 mm).
3. Recalculate the equivalent length, based on a copper tubing outside diameter of $2\frac{1}{8}$ in. (54 mm). The pressure drops of the refrigerant piping fittings and the accessories expressed as equivalent length of pipe are listed in Table 11.2. The calculated total equivalent length L_{eq} is

10 short radius elbows	$10 \times 5 = 50$ ft (15 m)
1 angle valve	$1 \times 24 = 24$ ft (7.2 m)
Measured length of pipe	30 ft (9 m)
Total	104 ft (31.2 m)

4. From the capacity-L_{eq} chart, at $Q_{rl} = 19.7$ tons (68.7 kW) and $L_{eq} = 104$ ft (31.2 m), the diameter of copper tubing is the same as the tentative diameter $2\frac{1}{8}$ in. (54 mm). Consequently, $2\frac{1}{8}$ in. (54 mm) should be the size of the suction line.

5. The minimum capacity of compressor in each refrigeration circuit is 0.5 × 20.7 = 10.4 tons (36.4 kW). From Table 11.4, when the liquid temperature is 115 − 10 = 105°F (40.6°C), the correction factor at 105°F is 0.96. The allowable minimum refrigeration load for a suction riser with entrained oil is 7.9 × 0.96 = 7.6 tons, which is smaller than 10.4 tons. Therefore, a double riser is not needed.

Discharge Line (Hot-Gas Line)

The design and sizing of the discharge line (hot-gas line) are similar to those of the suction line. Oil return and optimum pressure drop are the primary considerations.

In the discharge line, oil does not mix with gaseous refrigerant. It is entrained in the gaseous refrigerant at the discharge outlet of the compressor as oil mist. The horizontal pipe run should be pitched away from the compressor at a pitch not less than $^1/_2$ in. for each 10 ft (13 mm for each 3 m) of pipe run. The minimum refrigeration loads for hot-gas risers to carry entrained oil to the condenser are listed in Table 11.5. There would be an oil trap at the bottom of the vertical riser. When the compressor is unloaded to part-load operation, if the minimum Q_{rl} is less than that listed in Table 11.5, a double riser should be used. An additional oil trap should be added for each 25 ft (7.6 m) of pipe rise.

In the discharge line, the saturated temperature difference corresponding to the maximum pressure drop is 1°F (0.56°C) for halocarbon refrigerants. The maximum pressure drop is about 3 psi (21 kPa) for both HCFC-22 and HFC-134a. For ammonia R-717, a 1°F (0.56°C) saturated temperature difference is equivalent to about 4 psi (27.6 kPa).

The procedure for sizing of the discharge line is similar to that for the suction line. Figure 11.11 shows the discharge line Q_{rl}-L_{eq} chart for HCFC-22 at 40°F (4.4°C) suction temperature and 105°F (40.6°C) condensing temperature with a pressure drop corresponding to a 1°F (0.56°C) change in saturated temperature difference. If the suction temperature T_{suc} is not 40°F (4.4°C) and the

TABLE 11.5 Minimum Refrigeration Load for Oil Entrainment up Hot-Gas Risers of Type L Copper Tubing, in Tons

			Pipe OD, in.											
			0.500	0.625	0.750	0.875	1.123	1.375	1.625	2.125	2.625	3.125	3.625	4.125
	Saturated temperature,	Suction gas temperature,	Area, in.^{2}											
Refrigerant	°F	°F	0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
HCFC-22	80.0	110.0	0.235	0.421	0.695	1.05	2.03	3.46	5.35	10.7	18.3	28.6	41.8	57.9
		140.0	0.223	0.399	0.659	0.996	1.94	3.28	5.07	10.1	17.4	27.1	39.6	54.9
		170.0	0.215	0.385	0.635	0.960	1.87	3.16	4.89	9.76	16.8	26.2	38.2	52.9
	90.0	120.0	0.242	0.433	0.716	1.06	2.11	3.56	5.50	11.0	18.9	29.5	43.0	59.6
		150.0	0.226	0.406	0.671	1.01	1.97	3.34	5.16	10.3	17.7	27.6	40.3	55.9
		180.0	0.216	0.387	0.540	0.956	1.88	3.18	4.92	9.82	16.9	26.3	38.4	53.3
	100.0	130.0	0.247	0.442	0.730	1.10	2.15	3.83	5.62	11.2	19.3	30.1	43.9	60.8
		160.0	0.231	0.414	0.884	1.03	2.01	3.40	5.26	10.5	18.0	28.2	41.1	57.0
		190.0	0.220	0.394	0.650	0.982	1.91	3.24	3.00	9.96	17.2	26.8	39.1	54.2
	110.0	140.0	0.251	0.451	0.744	1.12	2.19	3.70	5.73	11.4	19.6	30.6	44.7	62.0
		170.0	0.235	0.421	0.693	1.05	2.05	3.46	3.35	10.7	18.3	28.6	41.8	57.9
		200.0	0.222	0.399	0.658	0.994	1.94	3.28	5.06	10.1	17.4	27.1	39.5	54.8
	120.0	150.0	0.257	0.460	0.760	1.15	2.24	3.78	5.85	11.7	20.0	31.3	45.7	63.3
		180.0	0.239	0.428	0.707	1.07	2.08	3.51	5.44	10.8	18.6	29.1	42.4	58.9
		210.0	0.225	0.404	0.666	1.01	1.96	3.31	5.12	10.2	17.6	27.4	40.0	55.5

Notes: The capacity in tons is based on a saturated suction temperature of 20°F with 15°F superheat at the indicated saturated condensing temperature with 15°F subcooling. For other saturated suction temperatures with 15°F superheat, use the following correction factors:

Saturated suction temperature, °F	−40	−20	0	40
Correction factor	0.88	0.95	0.96	1.04

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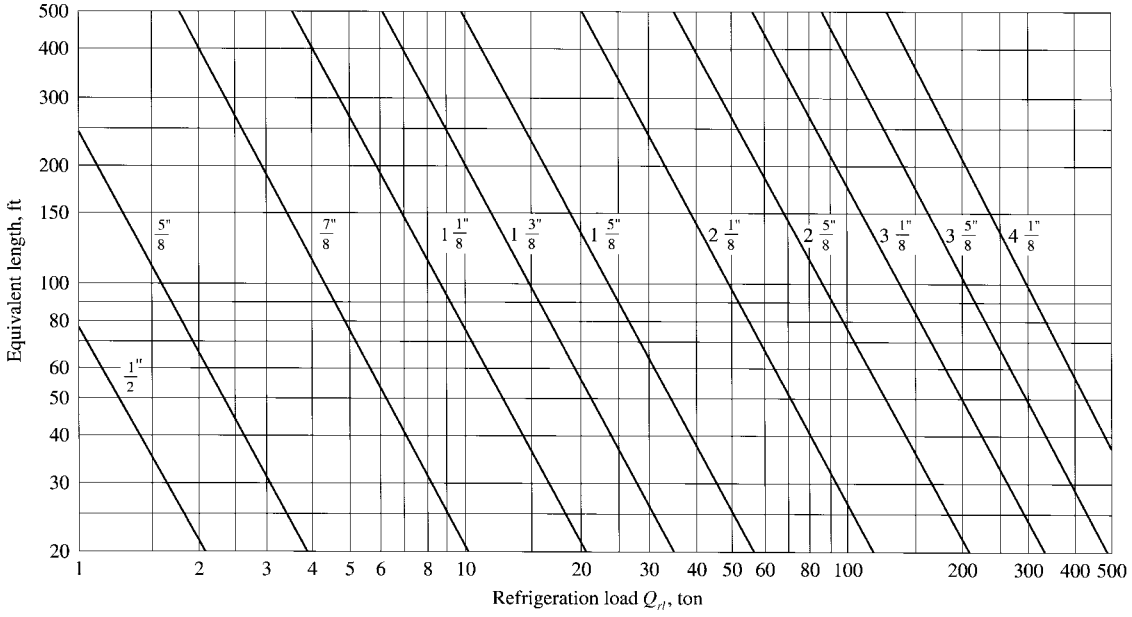


FIGURE 11.11 Discharge line sizing Q_{rl}-L_{eq} chart for HCFC-22 at 40°F suction temperature and 105°F condensing temperature (type L copper tube, outside diameter, 1°F change in saturated temperature).

condensing temperature T_{con} is not 105°F (40.6°C), the corrected refrigeration load is

$$Q_{rl,c} = \frac{Q_{rl}}{F_{\text{dis},s} F_{\text{dis},c}} \tag{11.3}$$

Correction factor $F_{\text{dis},s}$ takes into account the effect of h_{lv}-h_{en} on $\dot{m}_r$ in the discharge line when $T_{\text{suc}} \neq 40$°F (4.4°C). If $T_{\text{suc}} < 40$°F (4.4°C), then $F_{\text{dis},s} < 1$ and $q_{rl,c} > q_{rl}$.

Correction factor $F_{\text{dis},c}$ takes into account the combined effect of the density of the hot gas ρ and enthalpy difference $h_{lv} - h_{\text{en}}$ on $\dot{m}_r$ when $T_{\text{con}} \neq 105$°F (40.6°C). If $T_{\text{con}} < 105$°F (40.6°C), then $F_{\text{dis},c} < 1$ and $Q_{rl,c} > Q_{rl}$. Table 11.1 lists the correction factors $F_{\text{dis},s}$ and $F_{\text{dis},c}$ for HCFC-22 for the discharge line.

Liquid Line

Because oil is partially miscible with liquid refrigerant such as HCFC-22, HFC-134a is miscible with synthetic oil, and oil is separated from ammonia after the compressor, oil return is not a problem in the liquid line. Therefore, the primary considerations in liquid line design and sizing are the optimum pressure drop and the prevention of flashing of liquid refrigerant into vapor before the expansion valve.

Current practice is to limit the maximum pressure drop in the liquid line corresponding to a change of 1°F saturated temperature. For HCFC-22 it is about 3 psi (20.7 kPa), for HFC-134a it is about 2.5 psi (17.2 kPa), and for ammonia R-717 it is about 3.5 psi (24 kPa).

To prevent the flashing of liquid to vapor, liquid refrigerant must be subcooled before it is discharged to the liquid line. Recently designed air-cooled condensers for reciprocating refrigeration systems can provide a subcooling of 10 to 20°F (5.6 to 11.1°C). The liquid refrigerant should be in a subcooled state when it enters the expansion valve.

If a receiver is installed after the condenser, the liquid line between the condenser and receiver should be sized so that liquid flows from the condenser to the receiver and gas flows in the opposite direction. This section of liquid line is usually sized based on a liquid velocity of 100 fpm (0.5 m/s).

In the design and planning of the liquid line layout, the following factors must be considered:

- If a receiver is placed after the subcooling section of the condenser, subcooling will occur at the outlet of the receiver only when a surge-type liquid receiver is used or if an additinal subcooling setup is added after the receiver.
- Static pressure due to the liquid refrigerant column must be taken into account. At 105°F (40.6°C), the density of liquid HCFC-22 is 70.3 lb/ft^3 (1125 kg/m^3). For each foot of liquid column, the increase of static pressure is therefore $\Delta p_s = 70.3/144 = 0.488$ psi (3.4 kPa), or roughly 0.5 psi (3.5 kPa) for each foot of liquid column. At the top of a 10-ft (3-m) riser, the pressure is about $10 \times 0.5 = 5$ psi (35 kPa) less than at the bottom of the riser.

Figure 11.12 is a Q_{rl}-L_{eq} chart for a liquid line with HCFC-22 at 40°F (4.4°C) suction temperature and 105°F (40.6°C) condensing temperature with a pressure drop corresponding to a change of 1°F (0.56°C) saturation temperature.

For suction temperature other than 40°F (4.4°C) and condensing temperature other than 105°F (40.6°C), the corrected refrigeration load is calculated as

$$Q_{rl,c} = \frac{Q_{rl}}{F_{\text{liq},s} F_{\text{liq},c}} \tag{11.4}$$

Correction factor $F_{\text{liq},s}$ takes into account the effect of enthalpy difference h_{lv}-h_{en} on $\dot{m}_r$. Correction factor $F_{\text{liq},c}$ also takes into account the effect of enthalpy difference on $\dot{m}_r$. Its values are listed in Table 11.1. When $T_{\text{con}} > 105$°F (40.6°C), $F_{\text{liq},c} < 1$, and $Q_{rl,c} > Q_{rl}$.

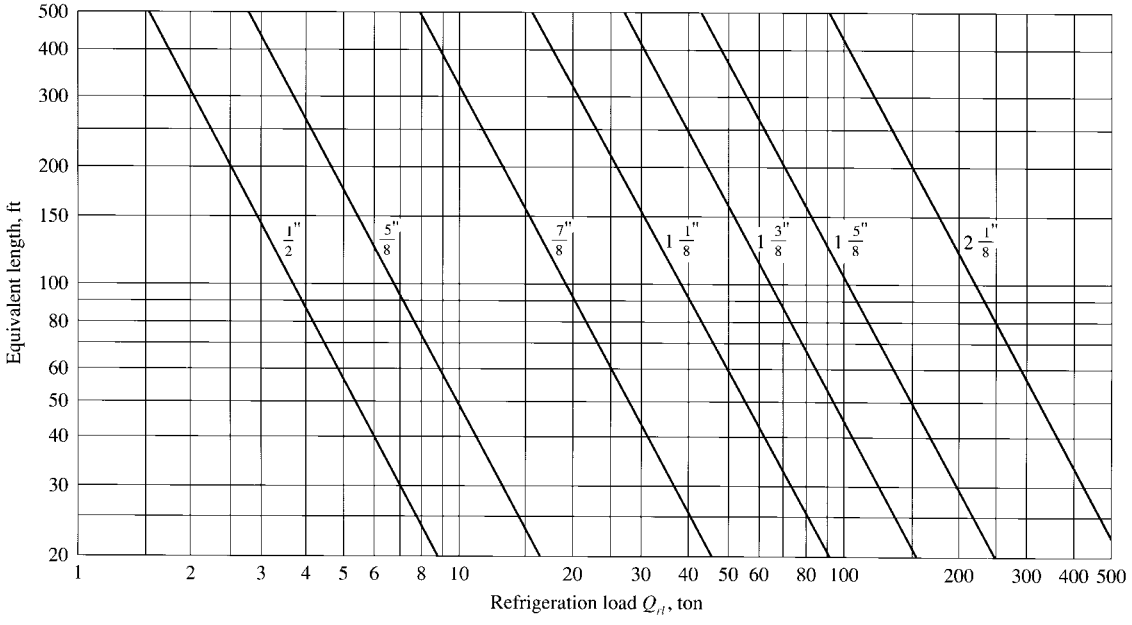


FIGURE 11.12 Liquid line sizing Q_{rl}-L_{eq} chart for HCFC-22 at 40°F suction temperature and 105°F condensing temperature (type L copper tube, outside diameter, 1°F change in saturated temperature).

Consider the reciprocating refrigeration system described in the discussion of suction line sizing. The subcooling at the condenser outlet is 10°F (5.6°C). There are three manual shutoff angle valves, one solenoid valve (using the pressure drop as globe valve), one sight glass of 5-ft (1.5-m) equivalent length, and 1 filter dryer of 150-ft (45-m) equivalent length according to the manufacturer's data in the liquid line. Also a 5-ft (1.5-m) vertical riser is connected to the outlet of the condenser, and the measured length of the liquid line is 40 ft (12 m). Sizing of the liquid line can be accomplished as follows:

1. From Table 11.1, the correction factor for the refrigeration load in the liquid line at a suction temperature of 45°F is $F_{\text{liq},s} = F_{\text{dis},s} = 1.01$. Condensing temperature is 115°F and with a subcooling of 10°F, $F_{\text{liq},c} = 1.0$. The corrected refrigeration load is then

$$Q_{rl,c} = \frac{Q_{rl}}{F_{\text{liq},s} F_{\text{liq},c}} = \frac{20}{1.01 \times 1.0} = 19.8 \text{ tons (69.6 kW)}$$

2. Assume an equivalent length $L_{\text{eq}} = 250$ ft (75 m).

3. From Fig. 11.12, the Q_{rl}-L_{eq} chart for the liquid line, the next-higher tentative diameter of copper tubing for the liquid line at $Q_{rl,c} = 19.8$ tons and $L_{\text{eq}} = 250$ ft is $1\frac{1}{8}$ in. (29 mm).

4. Calculate the equivalent length as follows:

3 angle valves	$3 \times 12 = 36$ ft (10.8 m)
1 solenoid valve	$1 \times 29 = 29$ ft (8.7 m)
1 sight glass	$1 \times 5 = 5$ ft (1.5 m)
1 filter dryer	$1 \times 150 = 150$ ft (45 m)
Measured pipe length	40 ft (12 m)
Total	260 ft (7 m)

5. At $Q_{rl} = 19.8$ tons and $L_{\text{eq}} = 260$ ft, the next-higher diameter of the liquid line is still $1\frac{1}{8}$ in. (29 mm), the same as the tentative diameter. This is the required diameter of the liquid line.

6. For a condensing temperature of 115°F (46.1°C) and a subcooling temperature of 10°F (5.6°C), the saturated pressure at 115 − 10 = 105°F is 210.8 psig (1453 kPag). When the pressure of the refrigerant in the liquid line is greater than 210.8 psig (1453 kPag), flashing will not occur.

The lowest pressure occurs at the inlet of the expansion valve. This pressure can be calculated as follows:

Condensing pressure, corresponding to 115°F	242.7 psig (1673 kPag)
Pressure drop of the liquid line	− 3 psi (− 20.6 kPa)
Pressure drop from riser	$5 \times 0.5 = -2.5$ psi (− 17.2 kPa)
Liquid pressure at inlet of expansion valve	237.2 psig (1636 kPag)

The pressure at the inlet of the expansion valve is greater than 210.8 psig (1453 kPag), so liquid refrigerant will not be flashed in the liquid line.

Parallel Connections. When two or three compressors are connected in parallel, their suction lines should be arranged so that the suction vapor is evenly distributed to the compressors. A gas equalizer should be connected to the crankcase of each compressor to equalize suction pressure, and an oil equalizer should be used to equalize the oil level in all compressors.

When two or three condensers are connected in parallel, hot gas should be evenly distributed to the condensers. If a liquid receiver is used, all condensers should be connected to a single receiver with an equalizer between the hot-gas supply and the top of the receiver. Refer to ASHRAE Handbooks and manufacturers' manuals for details.

11.5 CAPACITY AND SAFETY CONTROLS

Capacity Control

Refrigeration systems for comfort air conditioning or industrial applications need capacity control to meet refrigeration load variations during operation. In vapor compression refrigeration systems, control of the refrigerant flowing through the compressor is widely used as the primary refrigeration capacity control. For reciprocating refrigeration systems using DX coils as evaporators, row or intertwined face control of DX coil capacity is often employed in conjunction with compressor capacity control for better performance. In medium-size and large systems, two separate refrigerant circuits are often used so that refrigerant flow in one can be cut off when the refrigeration load drops below 50 percent of design load.

There are four basic methods of controlling the capacity of reciprocating refrigeration systems by reducing the refrigerant flow rate during part-load operation.

On/Off Control. On/off or start/stop control is widely used to control the refrigerant flow and the capacity of the reciprocating compressors in residential air conditioners. For reciprocating refrigeration systems equipped with multiple compressors, as used in supermarkets, on/off control is often the suitable choice. On/off control is simple and inexpensive, but it produces a greater variation in the space air temperature and relative humidity maintained by the refrigeration system than other types of control do. In addition, cyclic operation may lead to a start/stop cyclic loss for the evaporator and condenser, which are located indoors and outdoors separately. Cyclic loss is discussed in the next chapter. On/off control also results in increased wear and tear on compressor components and shortens service life.

Cylinder Unloader. For multiple-cylinder reciprocating compressors, a cylinder unloader is often used for capacity control. Cylinder unloaders reduce the capacity of compressors by bypassing the compressed gas to the suction chamber, blocking the suction or discharge valve, or closing the suction valve late or early. Recently developed cylinder unloaders include a solenoid valve that unloads the cylinder(s) in response to any one of the following:

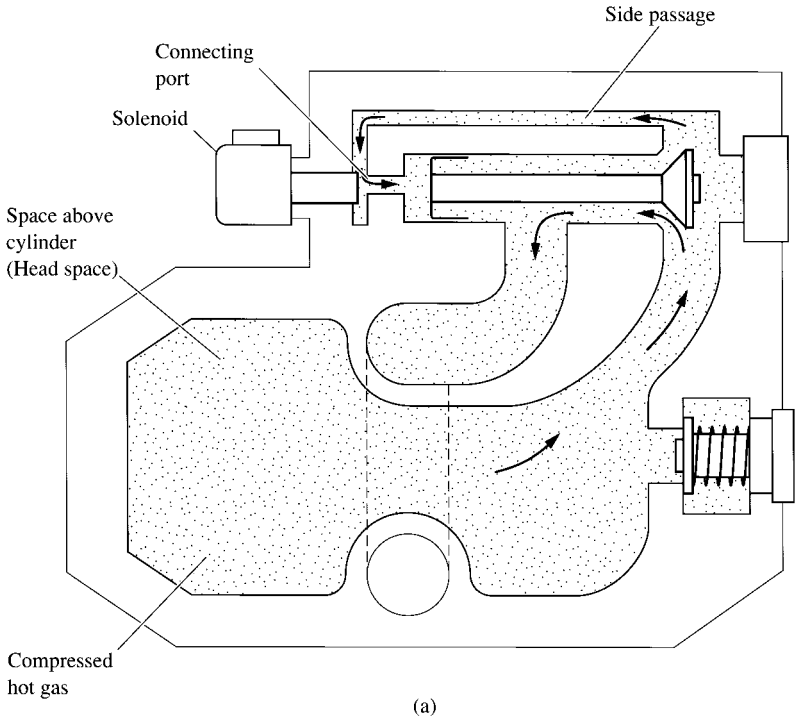
- Variation of discharge air temperature in a packaged unit
- Change in return chilled water temperature in a water chiller
- Changes in suction pressure in the evaporator

Figure 11.13 shows a typical cylinder unloader. When a temperature sensor in a packaged unit detects a drop in the discharge air temperature, the controller energizes the solenoid, draws the plunger away from the connecting port, allows the compressed gas to flow through a side passage, pushes the piston and valve stem of the unloader away from the solenoid, and opens the unloader valve. The compressed gas from the cylinder then flows to the suction side of the compressor. The two cylinders under the same head space are unloaded.

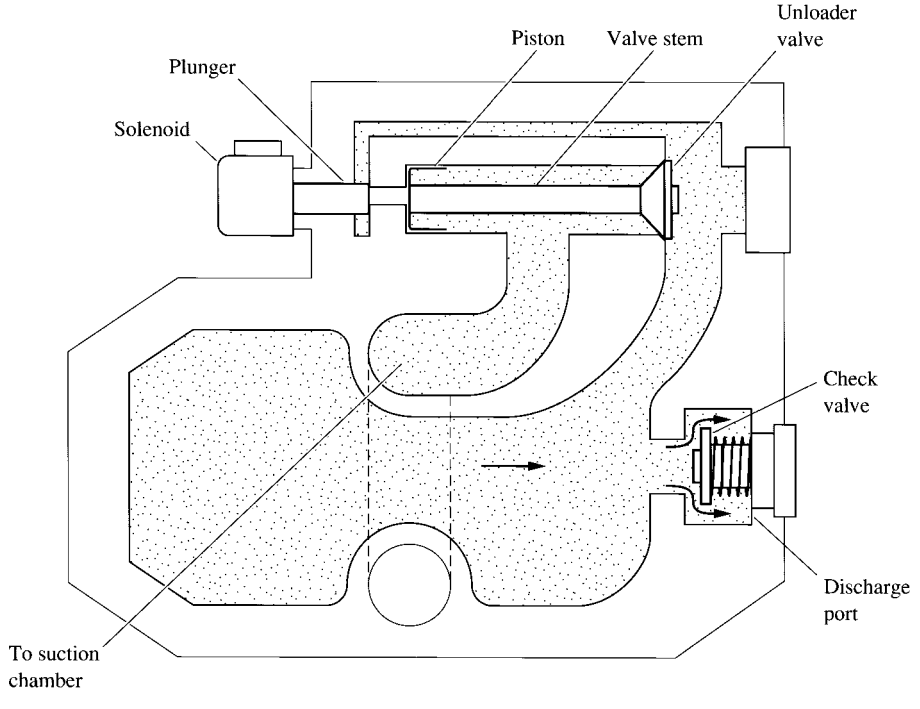
When the discharge air temperature rises, the plunger of the solenoid covers the connecting port. The pressure of the compressed gas exerted on the disk closes the unloader valve. Consequently, the increased pressure in the space above the cylinder opens the check valve and permits the discharge gas to enter the discharge line. The two cylinders are now loaded.

A cylinder unloader is a form of step control. Typically, the capacity of an eight-cylinder reciprocating compressor can be controlled in four steps: 100 percent, 75 percent, 50 percent, and 25 percent.

Speed Modulation. Recent developments in solid-state electronics may allow adjustable-frequency ac invertor-driven variable-speed motors to be used for capacity control in reciprocating compressors. Commercial variable-speed air conditioners with a capacity of few tons of refrigeration were available in Japan in the 1980s.



(a)



(b)

FIGURE 11.13 Cylinder unloader: (*a*) unloaded cylinders; (*b*) loaded cylinders.

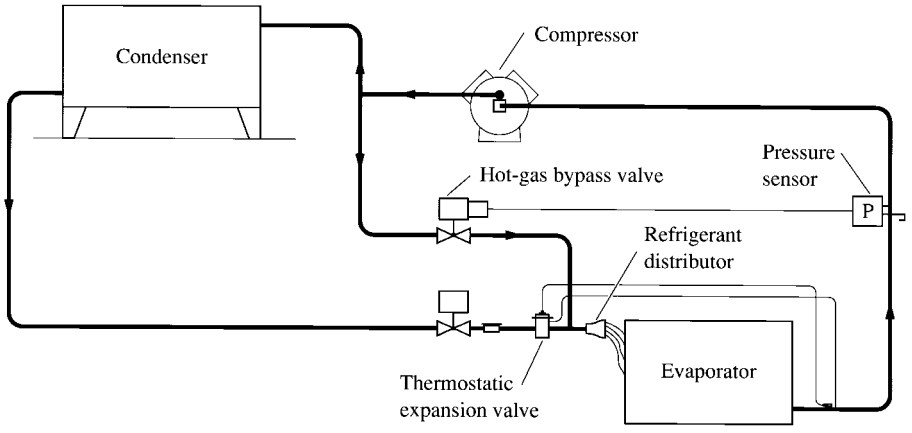


FIGURE 11.14 Hot-gas bypass control for a DX reciprocating system.

Invertor-driven variable-speed reciprocating compressors have the advantage of: lower start/stop or cycling losses, stable temperature control, and better system performance. They also have several drawbacks, such as cooling of the hermetic motor at low refrigerant flow rates, noise at high rotation speeds, and vibration at low speeds. In addition, they require a specially designed expansion valve for low-speed operation because a condensing pressure below a certain limit may affect the normal operation of the expansion valve.

Hot-Gas Bypass Control. In hot-gas bypass control, according to the sensed suction pressure, a certain amount of hot gas that balances the reduction in refrigeration load bypasses the condenser and the expansion valve, and enters the low-pressure side directly. In a DX reciprocating refrigeration system, bypassed hot gas is often introduced between the expansion valve and the liquid refrigerant distributor, as shown in Fig. 11.14.

Although hot-gas bypass control permits modulation of system capacity to a very low percentage, this method does not save energy at reduced refrigeration load. It is often used as a safety device to protect the evaporator or DX coil from frosting or excessive cycling of the compressor during on/off control when the refrigeration load is less than 20 to 25 percent of design capacity, or for industrial applications where strict temperature or humidity control is essential.

ASHRAE/IESNA Standard 90.1-1999 specifies that cooling systems shall not use hot-gas bypass or other evaporator pressure control systems except for (1) systems designed and equipped with multiple steps of unloading or continuous capacity modulation or (2) packaged systems with cooling capacities not grearer than 90,000 Btu/h (26 kW). When the cooling system rated capacity is less than or equal to 240,000 Btu/h (70 kW), the maximum hot-gas capacity is 50 percent of the total capacity. When the cooling system rated capacity is greater than 240,000 Btu/h (70 kW), the maximum hot-gas capacity is 25 percent of total capacity.

Safety Controls

Safety controls are used to shut down the refrigeration system during a malfunction and to prevent damage of system components. In reciprocating refrigeration systems, safety controls include low-pressure, high-pressure, low-temperature, frost, oil pressure failure, and motor overload controls.

Low-Pressure and High-Pressure Controls. The purpose of low-pressure control is to stop the compressor when the suction pressure drops below a preset value or when the refrigerant flow rate is too low to cool the compressor motor. Figure 11.15*a* shows a typical low-pressure control

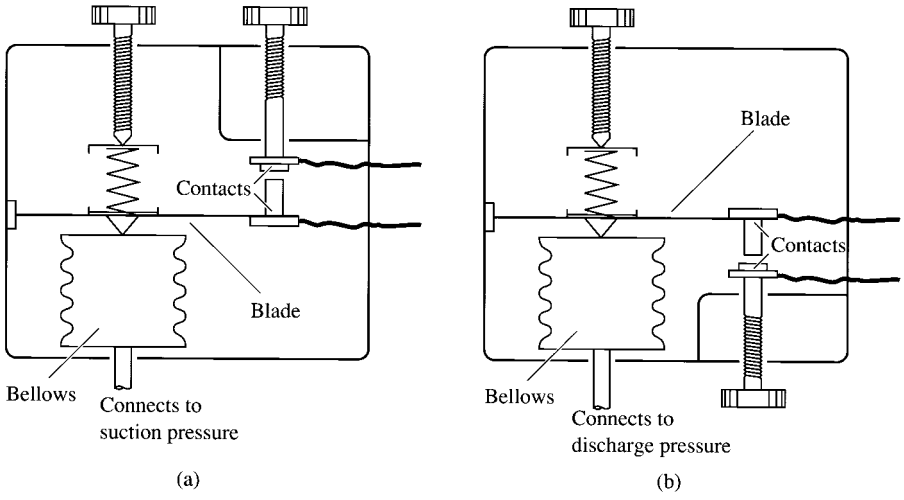


FIGURE 11.15 (*a*) Low pressure control and (*b*) high pressure control.

mechanism. When the suction pressure falls below a certain limit, the spring pushes the blade downward, opens the motor circuit, and stops the compressor. When the suction pressure increases, the bellows expand, thus closing the contact of the motor circuit and restarting the compressor. The two adjusting screws are used to set the cut-out and cut-in pressures. Cut-out pressure is the pressure at which the compressor stops, and cut-in pressure is the pressure at which the compressor starts again.

The purpose of high-pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 11.15*b* shows a typical high-pressure control mechanism. If the discharge pressure reaches a certain limit, the bellows expand so that the blade opens the motor circuit contact and the compressor stops. When the discharge pressure drops to a safe level, the bellows contract and close the contact, and the compressor starts again. As in a low-pressure control, two adjusting screws are used to set the cut-out and cut-in pressures.

In small refrigeration systems, low-pressure and high-pressure controls are often combined to form a dual-pressure control.

Low-Temperature Control. The purpose of a low-temperature control is to prevent the temperature of chilled water in the liquid cooler from falling below a certain limit, to protect the water from freezing. Freezing water damages the liquid cooler. Figure 11.16 shows a typical low-temperature control system. The sensing bulb senses the chilled water leaving temperature. When the temperature falls below a limit, contraction of the bellows opens the motor circuit and stops the compressor. When the chilled water temperature rises above the limit, the expansion of the bellows closes the circuit and restarts the compressor.

Frost Control. The purpose of frost control is to prevent the formation of frost on the outer surface of the DX coil. Usually, a temperature sensor is used to sense the outer surface temperature of the DX coil. When the temperature drops to 32°F (0°C), the controller actuates a relay, which opens the circuit and stops the compressor.

Oil Pressure Failure Control. The purpose of the oil pressure failure control is to stop the compressor when the oil pressure drops below a certain limit and fails to lubricate the main bearings and other components. Figure 11.17 shows a typical oil pressure failure control mechanism. When

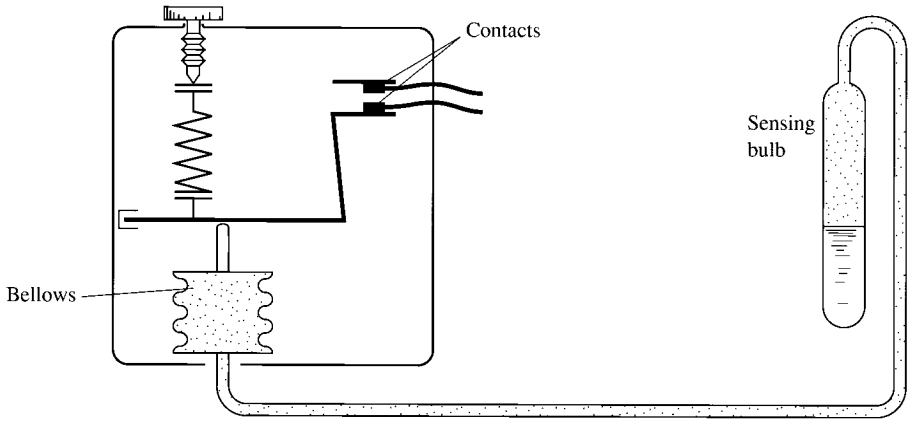


FIGURE 11.16 Low-temperature control.

the compressor starts, points *a* and *b* of the differential pressure switch and points *c* and *d* of the timer switch are in contact with each other. The differential pressure switch is affected by the differential pressure between oil pump discharge pressure and crankcase suction pressure. If the pressure of oil discharged from the oil pump does not reach a predetermined level within a certain time interval, the low oil pressure keeps the differential pressure switch *a* and *b* in a close contact position.

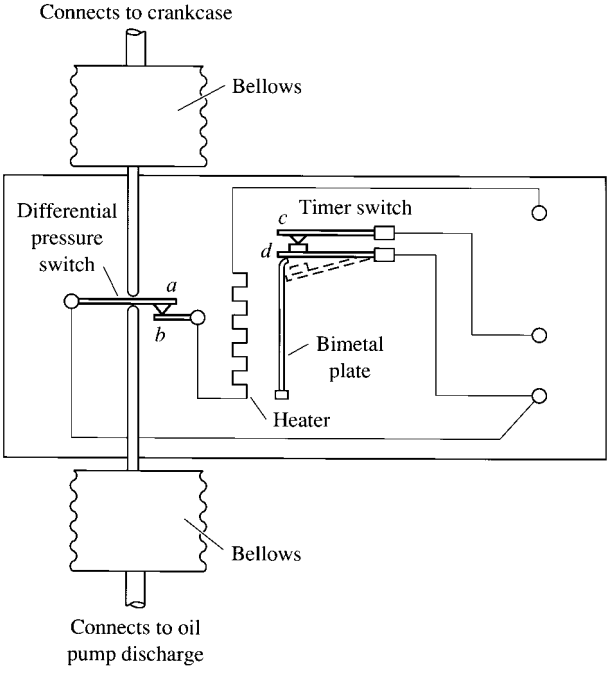


FIGURE 11.17 Oil pressure failure control.

The heater is energized, which causes the bimetal plate to deviate from the vertical position. The bimetal plate no longer keeps the timer switch *d* in contact with *c,* and the compressor stops.

If the oil pressure reaches the preset value within the predetermined time interval, *a* does not make contact with *b*. The breaking in contact of *a* and *b* deenergizes the heater before the bimetal plate starts to bend. The bimetal plate still holds the timer switch *d* in contact with *c*, and the compressor continues to operate.

If the oil pressure drops below the preset value and does not rise again within a certain time interval, the energized heater causes the bimetal plate to bend, opens the contact of the timer switch, and stops the compressor.

Motor Overload Control. The purpose of motor overload control is to protect the motor from dangerous overheating caused by a long period of overloading or failure of the compressor to start. Thermal protectors with bimetal or thermistor sensors are installed inside the motor or mounted on the outer surface of the motor shell. These sensors sense the temperature or both the temperature and electric current supplied to the motor windings. If dangerous overheating and overloading occur, a controller breaks the circuit.

Recently developed microprocessor-operated DDC system controllers combine all safety control mechanisms with other control systems of a reciprocating refrigeration system in one package. The DDC system controller senses, analyzes, and operates the safety control systems in coordination with other systems with far more functions and more sophisticated comprehensive controls.

11.6 PERFORMANCE OF RECIPROCATING COMPRESSOR, EVAPORATOR, AND CONDENSER

Performance of Reciprocating Compressor

There are two important parameters in the performance analysis of a reciprocating compressor: refrigeration capacity Q_{rc}, Btu/h or tons (W or kW) of refrigeration (as described in Sec. 9.10); and power input P_{in}, hp (kW); or energy efficiency ratio EER, Btu/h (W) (as defined in Sec. 9.20). Because Q_{rc} and P_{in} are always affected by suction pressure p_{suc} and discharge pressure p_{con} (both in psig), it is convenient to relate the performance of the reciprocating compressor to suction temperature T_{suc} (in °F) and to the condensing temperature T_{con}. The saturated pressure at T_{ev} is equal to p_{ev}. Suction pressure is approximately equal to the saturated pressure of $T_{ev} - 2$, and discharge pressure is approximately equal to the saturated pressure of $T_{con} + 1$, both in °F (°C).

Consider an accessible hermetic compressor with the following operating and construction characteristics:

Refrigerant	HCFC-22
Number of cylinders	8
Bore of cylinder	2.65 in. (67.3 mm)
Stroke of piston	2.25 in. (57.2 mm)
Revolutions per minute	1750
Suction temperature	40 °F (4.4°C)
Condensing temperature	115°F (46.1°C)
Subcooling	10 °F (5.6°C)
Superheating	10 °F (5.6°C)

Refrigeration Capacity. In reciprocating compressors, the piston only performs compression work during the upward stroke in a complete rotation of the crankshaft. Piston displacement $\dot{V}_p$, cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_p = \frac{\pi D_{cy}^2 L_{st} N n}{4} \tag{11.5}$$

where D_{cy} = diameter of the cylinder, ft (m)
L_{st} = stroke of cylinder, ft (m)
N = number of cylinders
n = rotating speed of compressor, rpm

From Eq. (9.72), the mass flow rate of the refrigerant $\dot{m}_r = \dot{V}_p \eta_v \rho_{suc}$, so from Eq. (9.25), the refrigeration capacity Q_{rc}, Btu/h (W), of a reciprocating compressor can be calculated as

$$Q_{rc} = 60 \dot{V}_p \eta_v \rho_{suc} (h_{1'} - h_4) \tag{11.6}$$

where ρ_{suc} = density of suction vapor, lb/ft^3 (kg/m^3)
$h_{1'}$ = enthalpy of superheated vapor leaving evaporator, Btu/lb (J/kg)
h_4 = enthalpy of subcooled liquid entering evaporator, Btu/lb (J/kg)

In the eight-cylinder reciprocating compressor using HCFC-22 as refrigerant described above, volumetric efficiency η_v depends mainly on compression ratio p_{dis}/p_{suc}. The smaller the p_{dis}/p_{suc} and the lower the reexpansion of clearance volume, cylinder heating, and leakages through valves and pistons, the greater the η_v. Figure 11.18*a* shows the η_v versus p_{dis}/p_{suc} curve for this compressor. For a suction temperature of 40°F (4.4°C), the absolute saturated pressure is 68.5 + 14.7 = 83.2 psia (574 kPa abs.). For a condensing temperature of 115°F (46.1°C), the absolute condensing pressure is 243 + 14.7 = 257.7 psia (1777 kPa abs.). The compression ratio is therefore p_{dis}/p_{suc} = (257.7 + 3)/83.2 = 3.13. From Fig. 11.18*a*, η_v = 0.81.

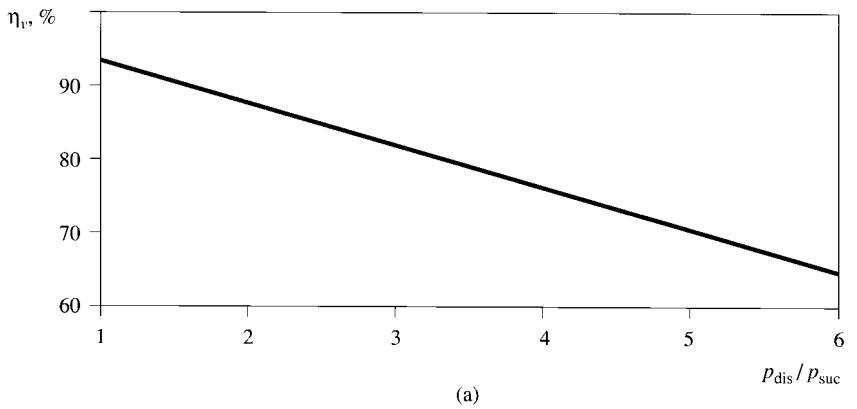
The upper part of Fig. 11.18*a* shows the refrigeration capacity versus suction temperature Q_{rc}-T_{suc} curves. If T_{con} remains constant and T_{suc} increases, the refrigeration capacity rises accordingly for the following reasons:

- For a specific compressor rotating at a specific speed, the piston displacement $\dot{V}_p$ is constant.
- Volumetric efficiency increases as T_{suc} rises.
- Density of the suction vapor refrigerant ρ_{suc} increases as T_{suc} increases because of higher saturated pressure.
- For a constant T_{con} and a fixed amount of subcooling, enthalpy h_4 is also constant. Enthalpy of superheated refrigerant leaving evaporator at a fixed degree of superheat $h_{1'}$ increases nearly linearly between 40 and 80°F (4.4 and 26.7°C) as T_{suc} rises. This also causes an increase of refrigeration effect $Q_{rf} = h_{1'} - h_4$. These factors result in a very slight upward bend in the Q_{rc} - T_{suc} curve.

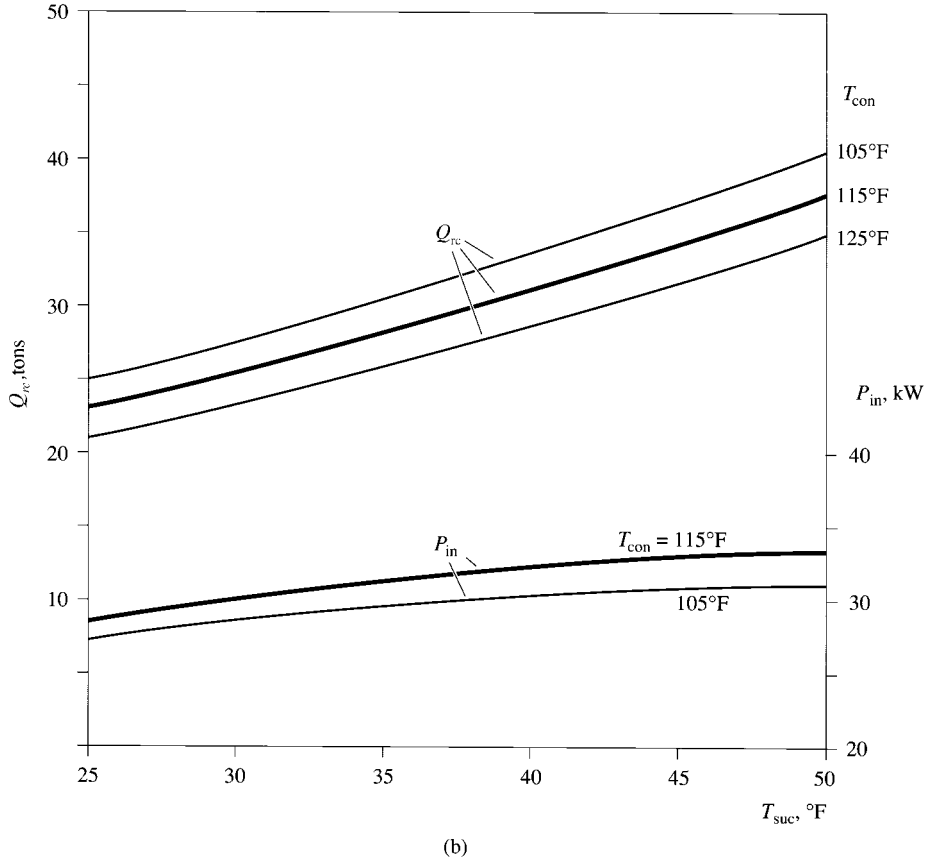
If the suction temperature T_{suc} is at a constant value, an increase in condensing temperature T_{con} results in a decrease in η_v and a decrease in Q_{rf} because of a smaller h_4. Consequently, Q_{rc} decreases as T_{con} rises, and Q_{rc} increases as T_{con} falls, as shown in Fig. 11.18.

Power Input. The refrigeration compressor consumes most of the energy input to a refrigeration system. In a single-stage reciprocating refrigeration system, power input P_{in}, kW, to the compressor can be calculated from Eqs. (9.71) and (9.72):

$$P_{in} = \frac{\dot{V}_p \eta_v \rho_{suc} (h_2 - h_{1'})}{56.85 \eta_{com}} \tag{11.7}$$



(a)



(b)

FIGURE 11.18 Cooling capacity, power input, and volumetric efficiency of a typical reciprocating compressor; (*a*) η_v–p_{dis}/p_{suc} curve; (*b*) Q_{rc}-T_{suc} and P_{in}-T_{suc} curves.

where h_2 = enthalpy of hot gas discharged from compressor if compression process is isentropic, Btu/lb
η_{com} = compressor efficiency, as defined in Sec. 9.20

If T_{con} remains constant and suction temperature T_{suc} increases, both η_v and ρ_{suc} rise and $h_2 - h_{1'}$ drops as T_{suc} increases. Compressor efficiency η_{com} also increases within a rather narrow range (0.75 to 0.80) when T_{suc} changes from 5 to 50°F (2.8 to 10°C), and the η_{com}-T_{suc} curve becomes nearly horizontal as T_{suc} exceeds 35°F (1.7°C). Because the increase in η_v and ρ_{suc} has a greater effect on P_{in} than the drop in h_2 -$h_{1'}$ and the change of η_{com}, the P_{in}-T_{suc} curve is slightly convex, indicating that P_{in} increases as T_{suc} rises, as shown in Fig. 11.18*b*. The P_{in}-T_{suc} curves are significantly flatter than Q_{rc}-T_{suc} curves.

If T_{suc} remains constant and T_{con} increases, η_v falls and $h_2 - h_{1'}$ becomes larger. The effect of the increase in $h_2 - h_{1'}$ is greater than that of the fall of η_v. The result is a higher P_{in}.

Performance of Evaporator

Cooling or refrigeration capacity Q_{rc}, Btu/h (W), is the most important parameter of an evaporator. For both DX coils and flooded liquid cooler evaporators, it is more convenient to use the following equation for performance analysis:

$$Q_{rc} = AU\Delta T_m \tag{11.8}$$

where A = evaporating surface area, ft^2 (m^2)
U = overall heat-transfer coefficient, Btu/h · ft^2 · °F (W/m^2 · °C)
ΔT_m = log-mean temperature difference, °F (°C)

For a DX coil or a flooded liquid cooler, surface area A is a constant and is always based on the total outer surface area A_o including the tubes and fins. For a liquid cooler with direct-expansion refrigerant feed, the surface area is the inner surface area of tubes plus inserts or enhanced surfaces.

Overall heat-transfer coefficient U depends mainly on fluid velocity, heat flux and the configuration at the evaporating surface, a dry or wet surface on the air side, and the construction of the DX coil or liquid cooler. In DX coils, the air velocity of a VAV system changes only when the system load varies. In a direct-expansion liquid cooler, chilled water velocity remains constant during operation.

The log-mean temperature difference ΔT_m is the temperature difference between the air temperature and the temperature of evaporating refrigerant inside the DX coil, or between the temperature of chilled water in the shell and the refrigerant in the liquid cooler. If the air temperature and its enthalpy entering the DX coil or the temperature of chilled water entering the liquid cooler both remain constant, then an increase of suction temperature T_{suc} reduces ΔT_m and a fall of T_{suc} increases ΔT_m. The cooling capacity of a DX coil or liquid cooler is then inversely proportional to T_{suc}.

Medium-size or large reciprocating refrigeration systems usually have two DX coils or two liquid coolers. Figure 11.19 compares the Q_{rc}-T_{suc} curves of two selected dual DX coils and two selected dual direct-expansion liquid coolers. As Q_{rc} is linearly proportional to T_{suc}, the Q_{rc}-T_{suc} curves are straight lines. The selected larger DX coil has a total face area of 2 × 21 ft^2 (3.9 m^2), and the smaller coil has a total face area of 2 × 15 ft^2 (2.8 m^2). The selected larger direct-expansion liquid cooler has a total tube area of 172 ft^2 (16 m^2), and the smaller one has a total tube area of 90 ft^2 (8.4 m^2).

When the entering air is at 80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb temperature, each 1 ft^2 (0.093 m^2) of face area of the DX coils has a total cooling capacity of 18,000 Btu/h (5274 W) at 40°F (4.4°C) suction temperature, 115°F (46.1°C) condensing temperature, and 600 fpm (3 m/s) of air velocity.

Because air is directly cooled in the DX coil, whereas chilled water is cooled before the air in a liquid cooler, the suction and evaporating temperatures in liquid coolers are lower than those in DX coils. Because the ΔT_m of a liquid cooler is far smaller than that of a DX coil, the Q_{rc}-T_{suc} curves of

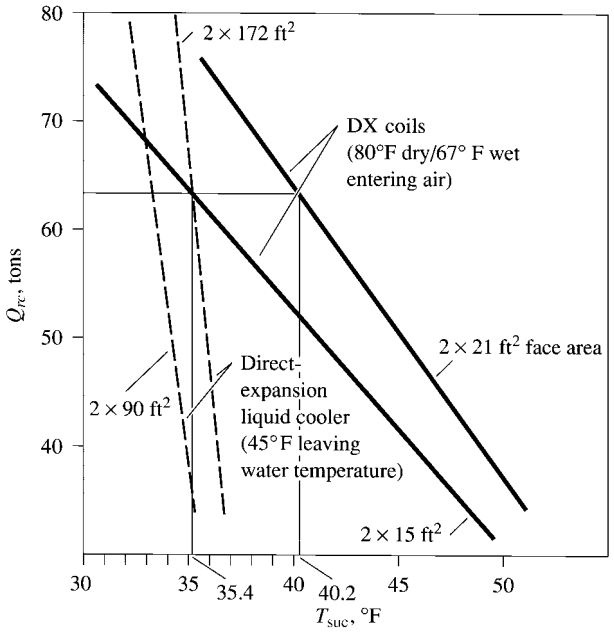


FIGURE 11.19 Performance curves Q_{rc}-T_{suc} for DX coils and direct-expansion liquid coolers.

direct-expansion liquid coolers have a much steeper negative slope than those of DX coils. For two evaporators at the same operating condition and Q_{rc}, the larger evaporator, which has a greater evaporating surface area, has a higher T_{suc} than the smaller one, as shown in Fig. 11.19.

Performance of Condenser

As for an evaporator, the performance of a condenser can be expressed by the following formula:

$$Q_{rej} = AU\Delta T_m \tag{11.9}$$

where Q_{rej} = total heat rejection at condenser, Btu/h (W). As described in Sec. 10.5, $Q_{rej} = F_{rej}\, Q_{rc}$. Here F_{rej} is the heat rejection factor.

In two condensers of the same type with different condensing surface areas, if their operating conditions are the same, the condensing temperature T_{con}, in °F (°C), of the condenser with a larger condensing surface is lower than that of the smaller condenser. For a specific condenser, although the heat flux at the condensing surface and fluid velocity affect the overall heat-transfer coefficient U value, the log-mean temperature difference ΔT_m is still the dominating factor that influences the performance of the condenser.

Figure 11.20 shows the Q_{rej}-T_{con} curves for condensers. Because Q_{rej} is linearly proportional to T_{con}, Q_{rej}-T_{con} curves are straight lines. Air-cooled condensers have a greater ΔT_m than water-cooled condensers, so the slope of the Q_{rej}-T_{con} curves for air-cooled condensers is smaller than that of water-cooled condensers.

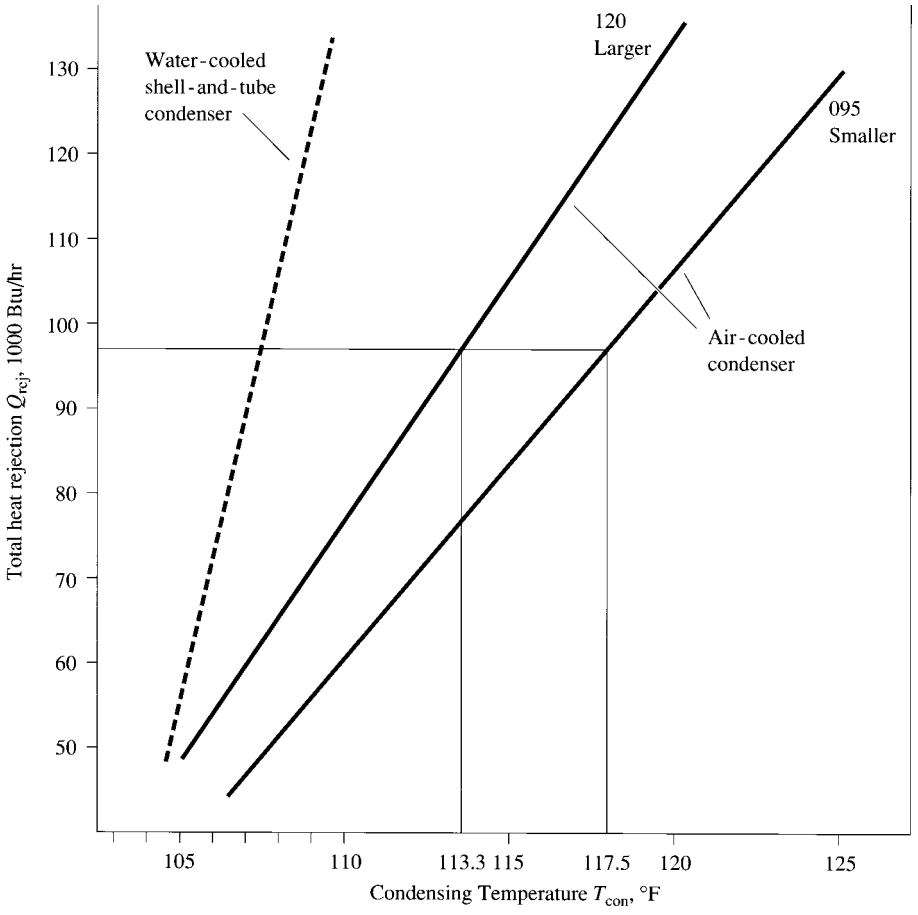


FIGURE 11.20 Performance curves Q_{rej}-T_{con} for condensers.

11.7 SYSTEM BALANCE

Because the refrigeration cycle of a reciprocating refrigerating system is a closed cycle, if the system is operated in a continuous and steady state (i.e., in an equilibrium state), according to the principle of continuity of mass and energy balance, the mass flow rates of refrigerant flowing through the evaporator, compressor, condenser, and expansion or float valve must all be equal. Also the total amount of energy supplied to the refrigeration system must be approximately equal to the total energy rejected from the system.

A continuous and steady state means that the flow is continuous, and the properties of the refrigerant at any point in the refrigeration system do not vary over time. Therefore, during the design of a refrigeration system, the system components selected should have equal or approximately equal mass flow rates of refrigerant at stable operation conditions.

Of the four main system components (compressor, evaporator, condenser, and expansion valve) in a reciprocating refrigeration system, the expansion valve adjusts its mass flow rate of refrigerant according to the degree of superheat of the vapor leaving the evaporator in the DX coil and the DX

liquid cooler. Although the displacement of a selected compressor $\dot{V}_p$ is a constant, the compressor can adjust its mass flow rates within a certain range because of the increase or decrease of the density of suction vapor ρ_{suc} and hot gas ρ_{dis} as T_{ev} and T_{con} vary.

There are two kinds of imbalance problems: imbalance of capacities of selected system components and imbalance caused by system load deviations from design conditions at part-load operation. Part-load operation is discussed later in this chapter.

Balance of Capacities of Selected Components

If the imbalance in required capacities of the selected system components is small, the refrigeration system adjusts its evaporating and condensing temperatures to reach a system balance slightly different from the required design operating parameters. If the capacities of the system components are seriously imbalanced, then either the refrigeration system will operate at parameters very different from the required conditions or the system will fail to operate or even be damaged by unacceptable operating parameters such as heavy overloading, low evaporating temperature, or high pressure cutout.

Consider a built-up rooftop reciprocating packaged system. The design refrigeration load Q_{rc} is 60 tons (211 kW) at a suction temperature T_{suc} = 40°F (4.4°C) and a condensing temperature T_{con} = 115°F (46.1°C). If two compressors are selected, each has a Q_{rc} of 31.6 tons (111 kW). Each compressor has eight cylinders, and the capacity of the unit can be controlled at 100, 75, 50, 25, and 0 percent. The Q_{rc}-T_{suc} curve of a single compressor is shown in Fig. 11.18.

Two DX coils are selected to match with the compressors, each with a face area of 21 ft^2 (1.95 m^2) with two expansion valves and two refrigerant distributors for intertwined face control. Their Q_{rc}-T_{suc} curves are shown in Fig. 11.19.

When these two compressors are connected with two 21-ft^2 (1.95-m^2) DX coils, the operating point of each compressor–DX coil combination must be the intersecting point Y of their Q_{rc}-T_{suc} curves, as shown in Fig. 11.21. At the intersecting point Y representing the design condition, Q_{rc} = 63 tons (222 kW) and T_{suc} = 40.2°F (4.5°C). The energy efficiency ratio EER is 11.8.

If two 15-ft^2 (1.4-m^2) DX coils are selected to form the compressor–DX coil combination, the intersecting point then moves to Y', with a total cooling capacity about 59.5 tons (209 kW) at T_{suc} = 38°F (3.3°C). The drop in T_{suc} necessitates a greater energy input because of the higher

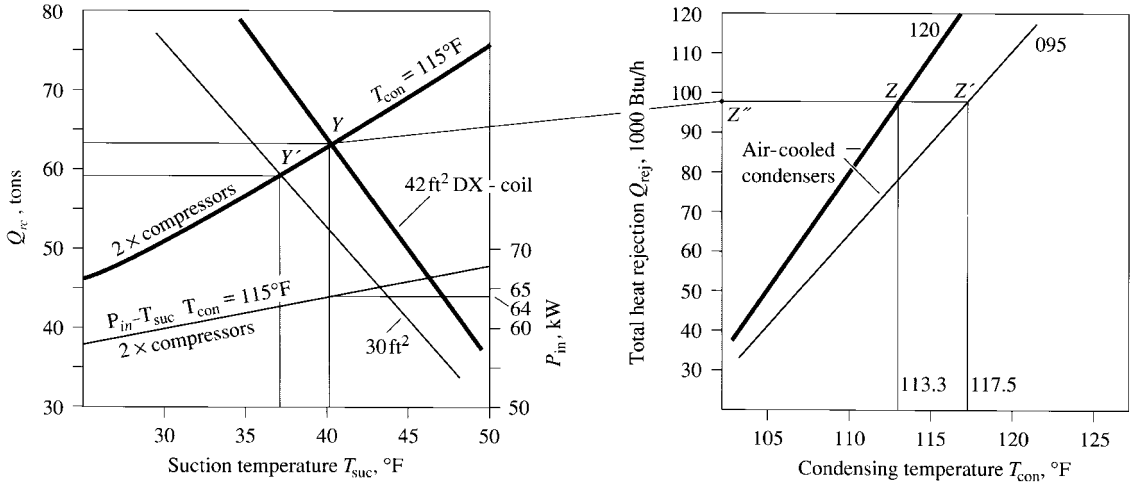


FIGURE 11.21 System balance of selected components for a reciprocating refrigeration system.

compression ratio. The EER drops to 11.5. It is preferable to select two DX coils each with a face area of 21 ft^2 (1.95 m^2) to reduce energy input.

Because the T_{con} of an air-cooled condenser affects its total heat rejection Q_{rej}, often T_{con} is estimated first when an air-cooled condenser is selected to determine Q_{rej} for the compressor–DX coil combination. If T_{con} is estimated at 115°F (46.1°C), from Fig. 11.21, for a power input of 64 kW to two compressors, the required total heat rejection of the selected air-cooled condenser is

$$Q_{rej} = Q_{rc} + 3413P_{in} = 63 \times 12{,}000 + 64 \times 3413 = 974{,}400 \text{ Btu/h (285,500 W)}$$

In Fig. 11.21, draw a line from point Y to point Z'' whose Q_{rej} = 974,000 Btu/h (285,500 W). Draw horizontal line $Z''Z$ intersecting the Q_{rej}-T_{con} curve of condenser size 120 at Z. At point Z, T_{con} = 113.3°F (45.2°C). If a smaller condenser 095 is selected, then the intersecting point is Z' and the corresponding T_{con} is 117.5°F (47.5°C), which necessitates a higher P_{in} to the compressor.

Using a larger condenser, the final selected capacity of this direct-expansion reciprocating refrigeration system may be as follows:

Refrigeration load	64 tons (225 kW)
Suction temperature	40°F (4.4°C)
Condensing temperature	114°F (45.6°C)
Power input to the compressor motor	64 kW
Total heat rejection Q_{rej}	986,000 Btu/h (288,900 W)

For thermostatic expansion valves, the selected capacity must be greater than the cooling capacity of the compressor at a pressure drop across the expansion valve of 75 to 120 psi (517 to 827 kPa) for HCFC-22 at design conditions.

11.8 OPERATING CHARACTERISTICS OF AIR-COOLED, DIRECT-EXPANSION RECIPROCATING REFRIGERATION SYSTEM

Operating Balance at Design Condition

Consider an air-cooled direct-expansion reciprocating refrigeration system in a rooftop packaged system with the same characteristics as described in the last section and shown in Fig. 11.21. If the design refrigeration load from the evaporation of HCFC-22 inside the two DX coils is only 60 tons or 720,000 Btu/h (211 kW) instead of the selected capacity of 64 tons (225 kW), then the volume flow rate of the piston displacement $\dot{V}_p$ to extract suction vapor is greater than that of the vaporized refrigerant in the evaporator. This imbalance lowers the evaporating pressure and temperature in the evaporator and, therefore, increases the specific volume v_{suc} of the suction vapor and the log-mean temperature difference as well as the heat transfer to the evaporator until $\dot{V}_p$ is balanced with the volume flow rate of refrigerant evaporized in the evaporator $\dot{V}_{ev}$, cfm (m/s). A new equilibrium between the compressor and evaporator is reached

$$\dot{m}_{com} = \dot{V}_p \eta_v \rho_{suc} = \dot{m}_{ev} = \frac{\dot{V}_{ev}}{v_{suc}} = \frac{200Q_{rc}}{q_{rf}} \tag{11.10}$$

where $\dot{m}_{ev}$ = mass flow rate of refrigerant evaporized in evaporator, lb/min (kg/min)
Q_{rc} = refrigeration load at evaporator, tons
q_{rf} = refrigeration effect, Btu/lb

The refrigerant discharged from the compressor then enters the air-cooled condenser at mass flow rate $\dot{m}_{com}$. Because the selected condenser can condense a mass flow rate of refrigerant equal to

a Q_{rej} of about 986,400 Btu/h (289,000 W), a mass flow rate of refrigerant greater than $\dot{m}_{com}$ is condensed in the air-cooled condenser and causes a drop in condensing pressure p_{con} and condensing temperature T_{con}. As T_{con} drops, ΔT_m and Q_{rej} both fall accordingly until the condensed mass flow rate of refrigerant equals the mass flow rate of refrigerant discharged from the compressor $\dot{m}_{com}$ and a new balance is again formed in the condenser.

Pressure Characteristics

Figure 11.22 shows the pressure characteristics of a typical air-cooled, direct-expansion reciprocating refrigeration system using DX coils and HCFC-22 as refrigerant. At design conditions, subcooled liquid refrigerant at a temperature of 105°F (40.6°C) and a pressure of 238 psig (1641 kPa · g) is fed to the thermostatic expansion valve. Because the saturated pressure of HCFC-22 at 105°F (40.6°C) is 210 psig (1448 kPa · g), which is smaller than 238 psig (1641 kPa · g), no flashing will occur before the inlet of the expansion valve.

The pressure drop across the expansion valve is 100 psi (690 kPa), and the pressure drop in the refrigerant distributor and tube is 59 psi (407 kPa). A fraction of liquid refrigerant is vaporized in the expansion valve, distributor, and tubes. Refrigerant with a quality of about 0.15 enters the evaporator, the DX coil, at a temperature of 47°F (8.3°C) and a pressure of 79 psig (545 kPa · g).

In the DX coil, all the liquid refrigerant is vaporized. The pressure drop across the refrigerant circuits of the DX coil is 8 psi (55 kPa). At point *x*, the evaporating temperature is 42°F (5.6°C), and the evaporating pressure is 71 psig (490 kPa · g). At the outlet of the evaporator, vapor refrigerant

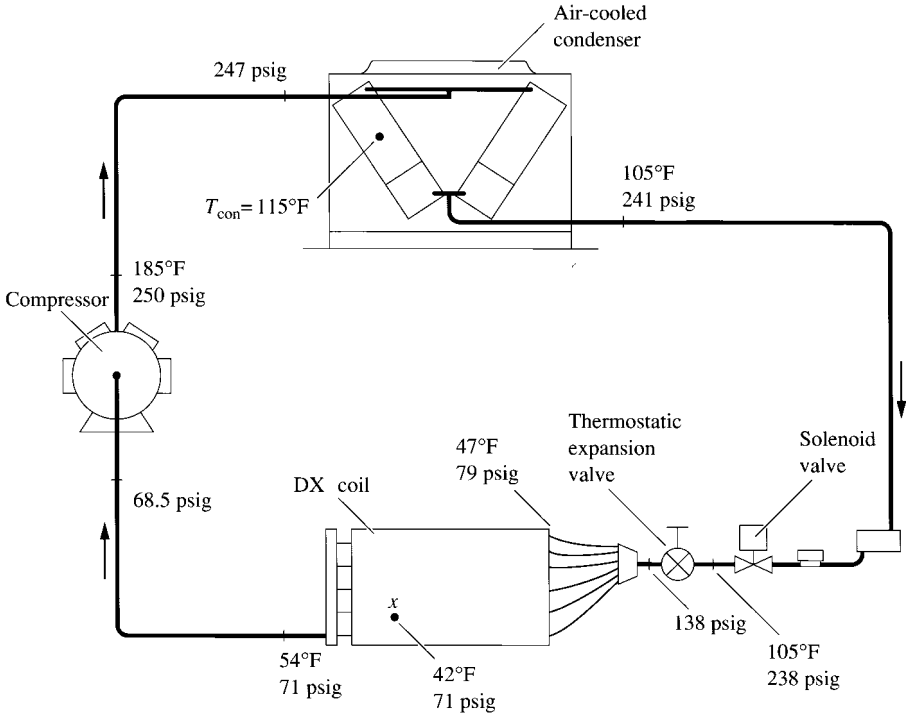


FIGURE 11.22 Pressure characteristics of a typical air-cooled direct-expansion reciprocating refrigeration system using HCFC-22 as a refrigerant.

is superheated to 54°F (12.2°C), with a pressure of 71 psig (490 kPa · g). The degree of superheat is 12°F (6.7°C).

The pressure drop at the suction line is 3 psi (20.7 kPa) corresponding to a 2°F (1.1°C) change in saturated temperature. Suction vapor enters the reciprocating compressor at a suction temperature of 40°F (4.4°C) and a pressure of 68.5 psig (472 kPa · g). In the compressor, the pressure is raised to the hot-gas discharge pressure of 250 psig (1724 kPa · g) with a temperature of about 185°F (85°C).

In the discharge line, the pressure drop is 3 psi (20.7 kPa), corresponding to a 1°F (0.56°C) change in saturated temperature. At the inlet of the air-cooled condenser, the pressure of the hot gas is 247 psig (1703 kPa · g). Hot gas is desuperheated, condensed, and subcooled in the air-cooled condenser at a condensing temperature of 115°F (46.1°C) and a pressure of 243 psig (1676 kPa · g). Liquid refrigerant is discharged from the subcooling coil of the air-cooled condenser at a temperature of 105°F (40.6°C) and a pressure of 241 psig (1662 kPa · g). The degree of subcooling is 10°F (5.6°C).

In the liquid line, the pressure drop is 3 psi (20.7 kPa), corresponding to 1°F (0.56°C) change in saturated temperature. Liquid refrigerant may enter the thermostatic expansion valve at a temperature of 105°F (40.6°C) and a pressure of 238 psig (1641 kPa · g).

Of the total system pressure drop of 250 − 68.5 = 181.5 psi (1248 kPa), a pressure drop of 159 psi (1096 kPa), or 88 percent of the total system drop, occurs in the expansion valve, distributor nozzle, and tubes. During part-load operation, when outdoor temperature is lower, this pressure drop may decrease considerably.

Part-Load Operation Using a Cylinder Unloader

The cylinders of a modern DX refrigeration system in a rooftop packaged unit using a microprocessor-based DDC system controller can be loaded or unloaded successively in four successive steps when discharge temperature T_{dis} rises and falls between 50 and 56°F (10 and 13.3°C) and T_{dis} is maintained at 53°F (11.7°C) by means of PI control mode, as shown in the lower right-hand bottom corner of Fig. 11.23. Cylinders are loaded in successive steps when T_{dis} is raised 1°F (0.56°C) higher than the upper limit of the control band 55°F (2.8°C), and cylinders are unloaded when T_{dis} is 1°F (0.56°C) lower than the lower limit of 51°F (10.6°C). Details of T_{dis} control for a rooftop packaged unit are discussed in Sec. 29.6.

In the morning of a summer day, when the rooftop unit starts with a refrigeration load of $Q_{rc} >$ 60 tons (211 kW) from the cool-down load, all 16 cylinders are operating. The operating point may be at A_o in Fig. 11.23 with a suction temperature T_{suc} = 42°F (5.6°C). As the refrigeration load drops to a value of 58 tons (204 kW), the discharge temperature T_{dis} gradually falls until it is equal to 50°F (10°C). At the same time, less refrigerant is vaporized in the DX coil because of the drop in refrigeration load. As the 16 cylinders of the two compressors have a greater extracted volume than the vapor refrigerant evaporated in the DX coil, the suction pressure p_{suc} and temperature T_{suc} drop to point A' on the 16-cylinder curve in Fig. 11.23. The DDC system controller unloads four cylinders and closes one of the four liquid line solenoid valves of the two evaporators. The operating point immediately shifts to point B on the 12-cylinder curve.

As the cooling capacity of the 12 operating cylinders is less than 58 tons (204 kW) of refrigeration load, more vapor refrigerant is evaporated in the DX coils than can be extracted by the 12 cylinders, and the suction pressure and temperature of the refrigerant at operating point B tend to move upward along the 12-cylinder curve to B'' with a T_{suc} equal to about 44°F (6.7°C). At the same time, because the cooling capacity of 12 cylinders Q_{rc} is smaller than the 58-ton (204-kW) required refrigeration load, T_{dis} rises to 56°F (13.3°C). The DDC system controller again loads four cylinders and opens the shutoff solenoid valve in the evaporator. The operating point suddenly moves to point A^* on the 16-cylinder curve. When 16 cylinders are operating, the cooling capacity provided is greater than 58 tons (204 kW), so point A^* again moves downward along the 16-cylinder curve. At the same time, T_{dis} drops to 50°F (10°C) and unloads four cylinders and opens the solenoid valve again. The refrigeration system cycles between points A^* and A' with 16 cylinders and 100 percent

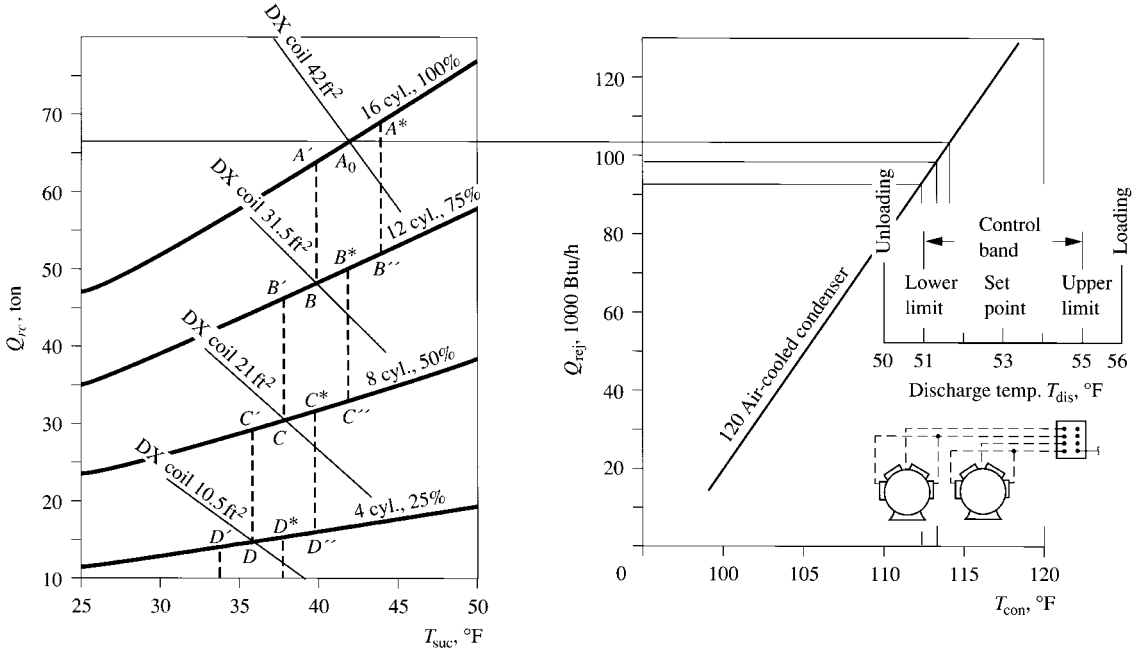


FIGURE 11.23 Part-load capacity control of a DX reciprocating refrigeration system using a cylinder unloader.

of evaporator's face area operating, and B and B'' with 12 cylinders and 75 percent of the face area operating. Operating points A^*, A', B, and B'' form a cycle. Because of the variation of percentages of operating time on 16 cylinders and on 12 cylinders, the refrigeration load provided by the compressor is balanced with the required 58-ton (204-kW) refrigeration load.

If the required refrigeration load drops to only 44 tons (155 kW), the operating point B on the 12-cylinder curve drops to B'. Because of the fall of refrigeration load, T_{dis} again drops to 50°F (10°C) and the DDC system controller then unloads to point C on the 8-cylinder curve and shuts-off one of the two compressors and two of the solenoid valves in the two evaporators. The compressors cycle between B', C, C'', and B^* until the refrigeration capacity provided is balanced with the required refrigeration load of 44 tons (155 kW) by varying the time duration of 12-cylinder operation and 8-cylinder operation.

Therefore, any refrigeration load from design load to a very low part-load can be balanced by unloading the cylinders, shutting off the compressors, and deenergizing the face area of the evaporator. When the required refrigeration load is greater than the refrigeration provided by the compressor and the DX coils, the cylinders are loaded and the DX coils are energized successively in the similar manner.

Pump-Down Control

Pump-down control is an effective means of preventing migration of refrigerant from the evaporator to the crankcase of the reciprocating compressor during the shutdown period. When a compressor starts, migrated refrigerant tends to mix with the oil and forms slugs which may damage the compressor.

When the discharge air temperature T_{dis}, or suction pressure drops below a preset limit and is sensed by a sensor, the DDC system controller deenergizes the liquid line solenoid valve, as shown

in Fig. 11.22, and the valve closes. The compressor pumps all gaseous refrigerant to the condenser, where it condenses to liquid. When the vapor pressure in the evaporator falls below a certain value, the low-pressure control breaks the electric circuit and stops the compressor.

If cooling is again required, then a temperature sensor senses the increase of T_{dis} or a pressure sensor senses the rise of suction pressure over a preset value, and the DDC system controller energizes the liquid line solenoid valve. Liquid refrigerant then flows into the evaporator where it evaporates. Vapor pressure builds up. As the pressure exceeds the cut-in pressure of the low-pressure control, the compressor starts again.

Pump-down control acts as a capacity control by controlling T_{dis} or the suction pressure. It also acts as a safety control to protect the compressor from damage from liquid slugs. Pump-down control is widely used in DX reciprocating refrigeration systems.

When the outdoor air temperature is low, in order to maintain proper operation of the thermostatic expansion valve, low ambient air control should be used to modulate the dampers or cycling, or even shut off the condenser fans, as described in Sec. 10.7.

Main Problems in Direct-Expansion Reciprocating Refrigeration Systems

Liquid Slugging. Liquid slugging is a mixture of by-product of corroded metallic surface with the refrigerant and oil. It is formed under the following conditions:

- Liquid refrigerant floods back from the evaporator to the crankcase of the compressor because of either an insufficient degree of superheat or hunting of the thermostatic expansion valve. Liquid refrigerant dilutes the lubrication oil.
- Liquid refrigerant migrates from the warmer indoor evaporator to the colder outdoor compressor during a shutdown period in split packaged units.
- A considerable amount of oil returns to the crankcase from the oil traps.
- When air and water are presented in any appreciable amount as contaminants in a refrigeration system, the chemical action involving these contaminants (the refrigerant and oil) often occurs, forming corrosive acid and producing sludge.

A liquid slug is incompressible. When a liquid slug enters the compression chamber, it may damage the valves, pistons, and other components.

An efficient filter dryer is essential to remove the moisture and prevents the formation of corrosive acid and sludge. Pump-down control and crankcase heater are effective means of preventing refrigerant migration in split packaged units. A proper degree of superheat is essential to prevent liquid refrigerant floodback to the compressor. Proper sizing of the thermostatic expansion valve and proper location of the sensing bulb prevent hunting of the thermostatic expansion valve.

Oil Return. For oil-miscible halocarbon refrigerants, the system design must include a means of returning oil from the condenser and evaporator to the compressor.

Compressor Short Cycling. Short cycling is on/off control of the compressor or loading or unloading cylinders within too short a time period (less than 3 min). It is mainly a result of close tolerance control or improper design of on/off capacity control. If the compressor is short-cycled, the cycling action may pump oil away from the compressor, causing insufficient oil lubrication. It may also damage system components. In addition, the short cycle does not allow sufficient time to restabilize the system.

Compressor short cycling can be solved by using a time delay for restart, or by overlapping unloading and loading schedules in step capacity control.

Defrosting. In a direct-expansion reciprocating refrigeration system, if the surface temperature of the DX coil is 32°F (0°C) and lower, frost accumulates on the coil surface. Because frost impedes

air passage and reduces the rate of heat transfer of the coil, it must be removed periodically. The process of removing frost from the evaporator is called defrosting.

If air enters the coil at a temperature above 36°F (2.2°C), the ambient air flowing through the DX coil can be used for defrosting. However, in low-temperature refrigeration systems, if the ambient air temperature is also below 32°F (0°C), an electric heating element may be used as a simple and effective way to defrost the coil. Hot-gas defrosting is effective, but must be planned during the design stage to ensure an adequate quantity of hot gas at a pressure high enough to flow through the coil. After defrosting, the high-pressure condensate should be returned to the liquid line, and the pressure of the desuperheated gas should be reduced gradually and released to the suction line.

The defrosting cycle can be controlled by sensing the pressure or temperature difference of air entering and leaving the DX coil over a fixed time interval.

Proper Refrigerant Charge. A proper amount of refrigerant charge is neccessary to ensure a proper DX system performance. Too much refrigerant charge will reduce the condensing area and therefore have a higher condensing pressure. An insufficient charge will decrease the required evaporation in the DX coil and thus have less than the required cooling capacity. A new refrigeration system should be charged with the exact amount of refrigerant specified by the manufacturer. For an operating DX system, a practical and convenient way to ensure a proper refrigerant charge is to check the discharge/suction pressure by means of pressure gauges and the degree of superheat at the outlet of the DX coil through its outer surface temperaure measurement.

If the condensing temperature of an air-cooled DX reciprocating system at an outdoor air temperature of 95°F (35°C) is 125°F (51.7°C), the discharge pressure is about 293 + 4 = 297 psia

TABLE 11.6 Electrically Operated Packaged Units and Condensing Units—Minimum Efficiency Requirements

Equipment type	Size category	Rating condition	Minimum efficiency*	Efficiency as of 10/29/2001*	Test procedure
Packaged units, air cooled	< 65,000 Btu/h†	Split system	10.0 SEER	10.0 SEER	ARI 210/240
		Single package	9.7 SEER	9.7 SEER	
	≥ 65,000 Btu/h and < 135,000 Btu/h	Split system and single package	8.9 EER§	10.3 EER§	
	≥ 135,000 Btu/h and < 240,000 Btu/h	Split system and single package	8.5 EER§	9.7 EER§	ARI 340/360
	≥ 240,000 Btu/h and < 760,000 Btu/h	Split system and single package	8.5 EER§ 7.5 IPLV§	9.5 EER§ 9.7 IPLV§	
	≥ 760,000 Btu/h	Split system and single package	8.2 EER§ 7.5 IPLV§	9.2 EER§ 9.4 IPLV§	
Packaged units water, and evaporatively cooled	< 65,000 Btu/h	Split system and single package	9.3 EER	12.1 EER	ARI 210/240
	≥ 65,000 Btu/h and < 135,000 Btu/h	Split system and single package	10.5 EER§	11.5 EER§	
	≥ 135,000 Btu/h and < 240,000 Btu/h	Split system and single package	9.6 EER§	11.0 EER§	ARI 340/360
	> 240,000 Btu/h	Split system and single package	9.6 EER§ 9.0 IPLV§	11.0 EER§ 10.3 IPLV§	
Condensing units, air cooled	≥ 135,000 Btu/h		9.9 EER 11.0 IPLV	10.1 EER 11.2 IPLV	ARI 365
Condensing units, water, or evaporatively cooled	≥ 135,000 Btu/h		12.9 EER 12.9 IPLV	13.1 EER 13.1 IPLV	

*IPLVs are only applicable to equipment with capacity modulation.
†Single-phase air-cooled air conditioners < 65,000 Btu/h are regulated by NAECA. SEER values are those set by NAECA.
§Deduct 0.2 from the required EERs and IPLVs for units with a heating section other than electric resistance heat.
***Source*:** ASHRAE/IESNA Standard 90.1-1999. Reprinted by permission.

(2048 kPa abs.), and the suction pressure is 91 − 3 = 88 psia (607 kPa abs.) with a suction temperature of 45°F (7.2°C). A 10 to 12°F (5.6 to 6.7°C) superheat at the outlet of the DX coil is considered suitable. Then at an outdoor air temperature of 95°F (35°C), if the discharge/suction pressure of the operating air-cooled DX system is around 282/73 psig (1944/503 kPag) and the superheat is between 10 and 12°F (5.6 to 6.7°C), the refrigerant charge is considered all right.

Minimum Performance and Equipment Selection

ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency requirements of air-cooled, water-cooled, and evaporatively cooled packaged units for direct-expansion reciprocating and scroll refrigeration systems. The requirements are expressed in terms of energy efficiency ratio (EER) and integrated part-load value (IPLV) as listed in Table 11.6. When multiple rating conditions or performance requirements are provided, the equipment shall satisfy all stated requirements unless otherwise exempted. The selected air-cooled packaged unit should meet the minimum performance guidelines at both design and part-load operation.

11.9 ROTARY REFRIGERATION SYSTEMS

Rotary refrigeration systems using rotary compressors are widely adopted in small refrigeration systems such as refrigerators, room air conditioners, and room heat pumps having a cooling capacity from a fraction of a horsepower to about 4 hp (< 0.75 kW to 3.0 kW).

Rotary Compressor

The rolling-piston rotary compressor, shown in Fig. 11.24, is the most widely used rotary compressor in small refrigeration systems today. A rolling piston is mounted on an eccentric shaft, and a

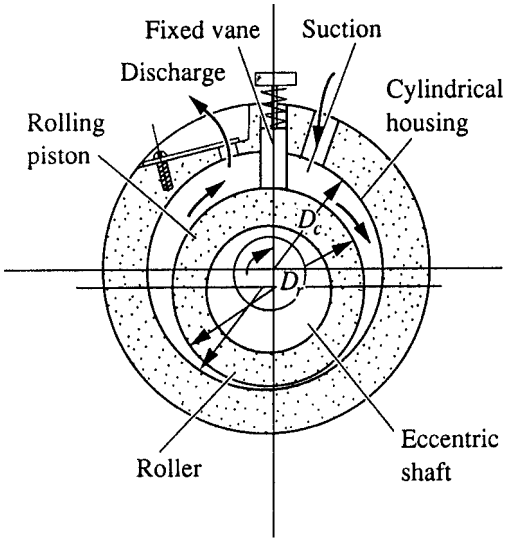


FIGURE 11.24 A rolling-piston rotary compressor.

fixed vane sliding in a slot remains in contact with the roller. As the piston rotates, the vane is in reciprocating motion. Vapor refrigerant enters the compression chamber through the suction inlet and is compressed by the eccentric motion of the roller. When the rolling piston is in contact with the top of the cylindrical housing, the hot gas is squeezed out through the discharge valve. Internal leakage depends on clearance of mating parts, surface speed, surface finish, and oil viscocity.

HCFC-22 is the most widely used refrigerant in rotary compressors. HFC-407C and HFC-410A with ozone depletion potential equal to zero will become the alternative refrigerants to replace HCFC-22 after the year 2020.

A rotary compressor has a small clearance volume and therefore greater volumetric efficiency and smaller reexpansion losses than a reciprocating compressor. The mechanical efficiency η_{mec} of a typical rotary compressor operating at a compression ratio of 3.5 is about 0.87.

Main Components

Similar to a reciprocating refrigeration system, a rotary refrigeration system consists of the following main components in addition to the rotary compressor:

- *Evaporator.* A direct-expansion or DX coil is often used as the evaporator. The DX coil is usually made of tube and corrugated fins to improve the heat transfer. Copper tubes typically of 0.5-in. (13-mm) diameter and aluminum fins with a thickness of 0.006 in. (0.15 mm) are used. An adequate cooling rate per unit coil face area, $Btu/h \cdot ft^2$ (W/m^2) is important during the selection of a DX coil.
- *Condenser.* An air-cooled condenser is also often made of tube and corrugated fins. The diameter of the copper tube is usually between $^1/_4$ and $^3/_4$ in. (6.5 and 19 mm). A subcooler is sometimes used and is located between the condenser and the capillary tube submerging in the condensate. The condensing coil usually has one to three rows of tubes because of the lower air-side pressure drop provided by the propeller fan.
- *Restrictor.* A capillary tube is the most widely used restrictor because of its low cost and reliability.
- *Permanent split-capacitor motor.* This type of motor provides adequate starting torque for rotary compressors and improves the power factor during operation.

System Performance

According to *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*, for an ambient temperature of 95°F (35°C), an evaporating temperature of 35°F (1.7°C), and a condensing temperature of 130°F (54.4°C) with a subcooling temperature of 15°F (8.3°C), the EER of a rotary refrigeration system for cooling is 9.0 (COP = 2.7). Because of its rotary motion, a rotary refrigeration system makes less noise than a reciprocating system.

11.10 SCROLL REFRIGERATION SYSTEMS

Types of Scroll Refrigeration Systems

As in the reciprocating refrigeration systems, the currently used scroll refrigeration systems for air conditioning are mainly for small to medium-size systems (packaged units and heat pumps) with a cooling capacity often between slightly less than 1 ton and 25 tons (3.5 and 90 kW) used in light commercial, industrial, and residential buildings. These include the following:

- Air-cooled, scroll DX cooler
- Air-cooled, scroll heat pumps
- Scroll chillers

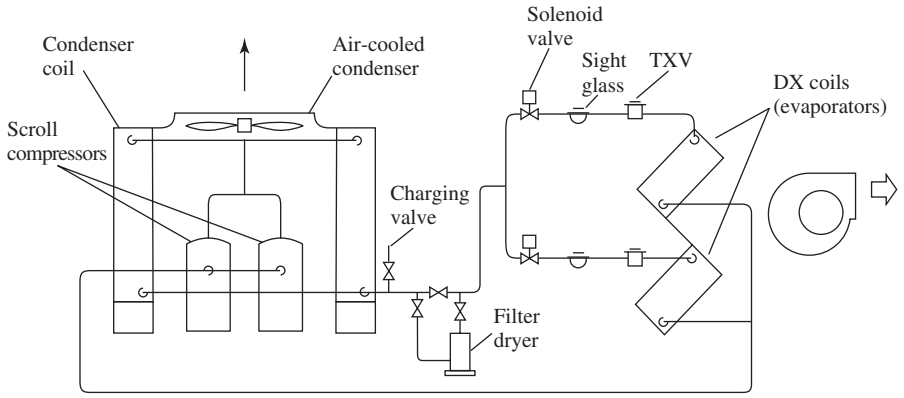


FIGURE 11.25 An air-cooled scroll DX cooler.

Figure 11.25 shows an air-cooled, scroll DX cooler. It consists of two scroll compressors, an air-cooled condenser, two DX coils, and two thermostatic expansion valves (TXVs). A scroll refrigeration system has almost the same refrigerant piping and safety controls as the reciprocating system except there is no crankcase heater.

An air-source scroll heat pump has nearly the same system structure as an air-cooled, scroll DX cooler except that a four-way reversing valve is added in the heat pump. Heat pumps are discussed in the next chapter.

Scroll Compressors

In 1987, the first U.S. manufacturer using a computer-aided manufacturing process produced a scroll compressor for residential air conditioning with tolerances up to 0.0001 in. (0.0025 mm). A scroll compressor consists of two identical spiral scrolls mated face to face, as shown in Fig. 11.26. Each scroll is bound on one side to a flat plate. The upper scroll is stationary and contains the discharge port. The lower one moves in an orbit around the shaft center of the motor at an amplitude equal to the orbit radius. The two scrolls are fitted together to form vapor pockets between their respective baseplates and lines of contacts of their vane walls. During suction, vapor refrigerant enters the space between the two scrolls through the lateral openings. The openings are then sealed off to form trapped vapor pockets, thus completing the intake process.

During the successive revolutions of the motor shaft, the volume of the trapped vapor pockets is reduced, and the compression process is complete when the gaseous refrigerant is compressed to its maximum pressure and just before the compressed gaseous pocket is opened to the discharge port. During discharge, the trapped gas pockets opens to the discharge port. All compressed hot gas is discharged through the discharge port to the discharge line. The volume of the gas pockets is reduced to zero. The sequence of intake, compression, and discharge occurs simultaneously in the two trapped gas pockets.

Radial and Axial Compliance

In scroll compressors, the scrolls touch one another with force sufficient to create a seal but not sufficient to cause wear. This is the result of radial and axial compliance. Continuous flank contact because of the centrifugal force minimizes the gas leakage.

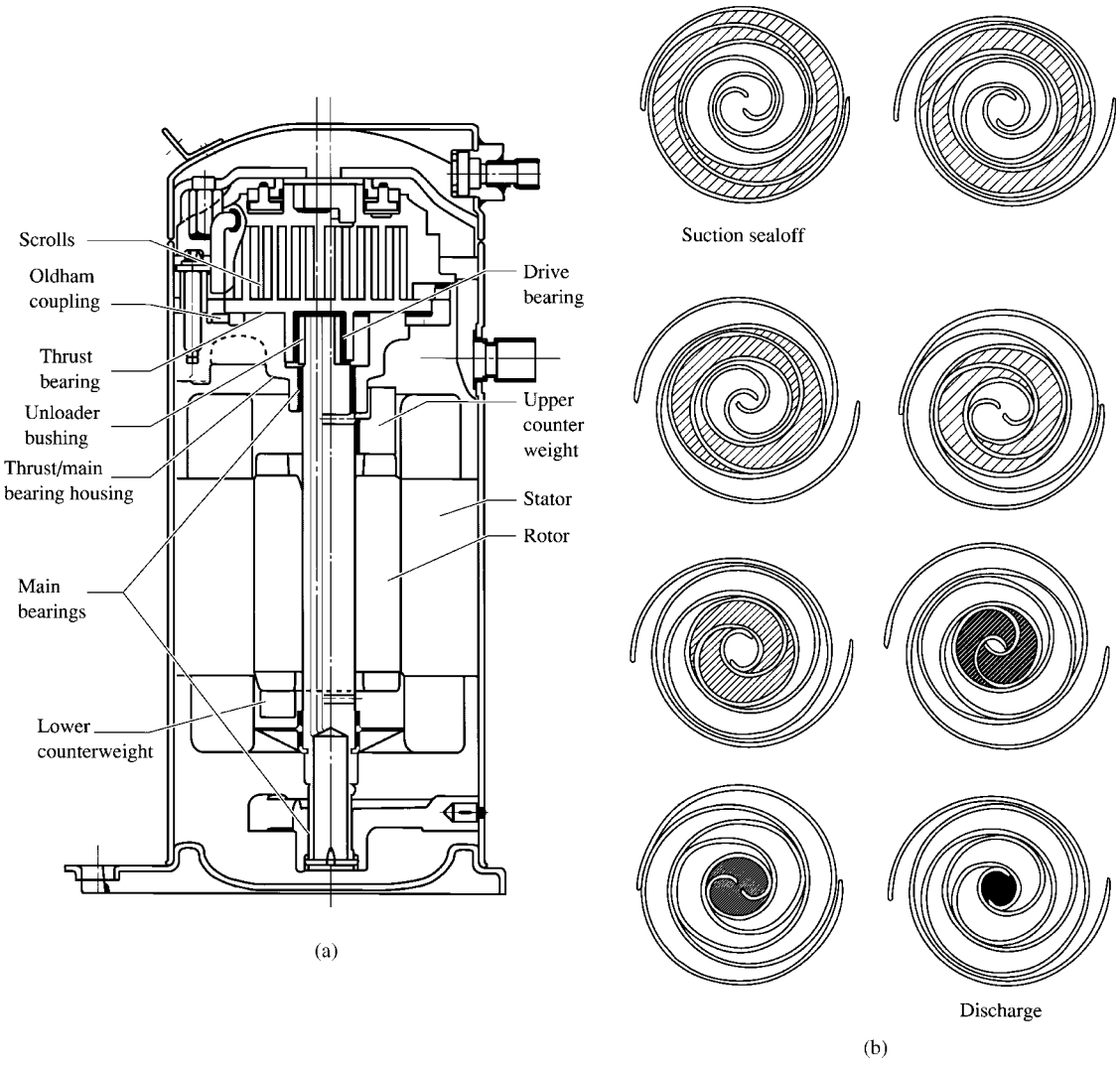


FIGURE 11.26 (*a*) Scroll compressor and (*b*) scroll compression processes. (*Source: ASHRAE Handbook 1996, HVAC Systems and Equipment. Reprinted with permission.*)

In one type of radial compliance, the orbiting scroll is mounted on the sliding unloader bushing that is fitted onto the eccentric pin of the motor shaft and thus directs the radial motion of the orbiting scroll in and out to accommodate variations in orbit radius due to discrepancies. Radial compliance allows the scroll members to separate in the presence of liquid refrigerant or debris and increases system reliability.

Axial compliance is accomplished by an adjustable axial pressure that provides sealing contact between the scroll tips and bases when the scroll is orbiting. This pressure is released when the orbiting scroll ceases to operate. As the scroll tip bears directly on both the baseplates and the lubrication oil between the scroll and the plates, both form an effective oil seal.

Compressor Performance

The cooling capacity of scroll compressors for a DX system is often between 1 and 15 tons (3.5 and 53 kW). HFC-407C and HFC-410A are the long-term alternative refrigerants, and HCFC-22 is the interim refrigerant that can be used before 2020.

Scroll compressors have a very high volumetric efficiency η_v. Levins et al. (1996) showed that η_v for the scroll compressor varied from 96.9 to 93.6 percent when the pressure ratio increased from 2.77 to 3.58. The pressure ratio (or compression ratio) is defined as the ratio of discharge pressure to the suction pressure p_{dis}/p_{suc}, both in psia (kPa abs.).

Scroll compressors have an isentropic efficiency η_{isen} of 70 percent when their pressure ratios are around 3. Most of the scroll compressors produced in the late 1990s have an EER of about 11.5 (COP = 3.35) which is higher than that of the rotary compressors and a standard reciprocating compressor.

ASHRAE/IESNA Standard 90.1-1999 mandates minimum energy requirement of air-cooled, water-cooled, and evaporatively cooled scroll DX packaged units which are the same as the reciprocating DX packaged units as listed in Table 11.6. The minimum efficiency requirements EER of water-cooled and evaporatively cooled DX packaged units are about 13 to 18 percent higher than air-cooled DX packaged units at various cooling capacities listed.

Concentration Shift and Temperature Glide

Basic Mechanism. In scroll, reciprocating, and screw refrigeration systems, refrigerant blends HFC-404A, HFC-410A, HFC-407A, and HFC-407C are used as alternative refrigerants to replace HCFC-22 and CFC-502 for zero ozone depletion and better system performance. HFC-404A and HFC-410A are near-azeotropic, whereas HFC-407A and HFC-407C are zeotropic or nonazeotropic and produce significant temperature glide in condensation and evaporation.

During the phase change in the two-phase region, the evaporating liquid changes its composition continuously and thus its bubbling point; also the condensing vapor changes its composition and thus its dew point, as shown on the temperature-concentration diagram in Fig. 11.27. These result in a concentration shift and temperature glide which is a temperature drop during condensation and a temperature gain during evaporation. There are two additional causes of concentration shifts:

- Differential solubility of lubrication oil of the components of the refrigerant mixture.
- Leakage of refrigerant from the system.

Circulating Concentration Shift and Its Effects on Performance

Kruse and Wieschollek (1997) measured the concentration and mass of refrigerant in the two-phase region in the evaporator and condenser of a DX refrigeration system during circulation of refrigerant. The concentration shift can be calculated by means of the mass balance. For HFC-407C, a blend of HFC-32/HFC-125/HFC-134a with original concentration of 23/25/52 percent, HFC-32 had a circulating concentration of 24.76 percent; HFC-125, 26.44 percent; and HFC-134a, 48.81 percent. The calculated concentration shifts during circulation were 1.76, 1.44, and −3.19 percent in weight for components HFC-32, HFC-125, and HFC-134a, respectively. The calculated performance changes because of the circulating concentration shift of HFC-407C are as follows:

	Original concentration	Concentration shift
Cooling capacity, Btu/h	10,529	10,625
Mass flow rate, lb/min	2.67	2.69
Evaporating pressure, psia	62.98	64.30
Condensing pressure, psia	315.3	324.3

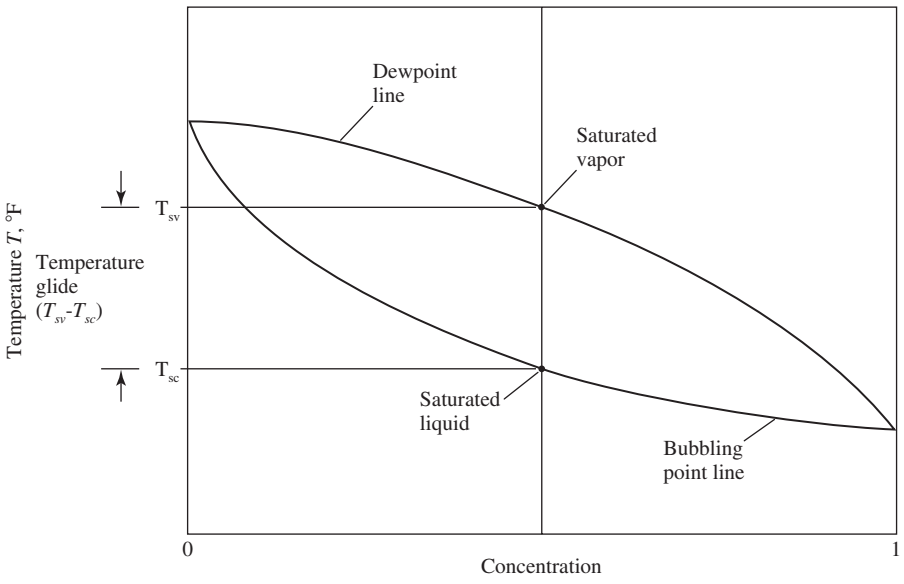


FIGURE 11.27 Temperature-concentration diagram of a zeotropic refrigerant blend.

As for a near-azeotropic refrigerant mixture HFC-404A, the concentration shift is small, the calculated performance change due to circulation concentration shift is less than 1 percent and can be neglected.

Corr et al. (1994) showed that direct-expansion evaporators typically have liquid fractions of 2 to 10 percent and result in a small shift in circulation concentration.

Heat Exchanger Flow Configuration

The temperature glide of zeotropic refrigerant blends can be utilized to increase the heat exchange efficiency in the evaporator where the evaporating temperaure increases and in the condenser where the condensing temperature drops. Rice (1993), using a modified Carnot cycle analysis, found that for an air-source heat pump using zeotropic blend as refrigerant at cooling mode with a relative *UA* (overall conductance-area product) level of 3, many-row crossflow has a potential COP gain of about 7 percent, two-row countercrossflow about 10 percent, three-row countercrossflow 15 percent, counterflow (or $>$ three-row countercrossflow) 20 percent. Hwang and Radermacher (1997) showed that the capacity of the near-counterflow heat exchanger increases about 10 percent compared to that of the near-parallel flow heat exchanger.

Capacity Control and Part-Load Performance

There are two types of capacity control: variable-speed and variable-displacement compressors. In a variable-speed scroll compressor, a variable-speed drive is used to reduce the speed at part-load operation so that the cooling capacity is reduced to meet the reduced system load. The operating frequency often varies from 15 to 150 Hz.

In a variable-displacement scroll compressor, there are porting holes in the fixed scroll. By connecting or disconnecting the compression chamber to the suction side through the closing and

opening of the porting holes, the cooling capacity of the scroll compressor is varied. If all porting holes are closed, the compressor is operating at full capacity. If all the porting holes are open, the cooling capacity drops to the minimum.

Takebayashi et al. (1994) showed the isentropic efficiency of a variable-speed scroll compressor at a pressure ratio range between 2 and 5 during part-load operation, cooling, and heating mode operation. Isentropic efficiency drops about 15 percent when the pressure ratio increases from 2 to 5.

System Characteristics

Compared to reciprocating and rotary compressors, the system characteristics of air-cooled, DX scroll refrigeration systems are as follows:

- Scroll compressors tolerate liquid refrigerant and debris during compression. For most of the air-cooled DX scroll systems, the crankcase heater and suction line accumulator are often not required.
- A scroll compressor has less than 50 percent of the parts of a reciprocating compressor of the same size. The fewer the parts of a compressor, the higher the reliability.
- Most scroll compressors provided an EER of 11.5 (3.37 COP) in the late 1990s, which is higher than that of a rotary compressor and is similar to that of a high-efficiency reciprocating compressor.
- A scroll heat pump shows a flatter capacity curve when outdoor air temperature T_o varies than a reciprocating heat pump does. Its cooling capacity drops less when the outdoor ambient temperature increases during summer, and the heating capacity of a scroll heat pump reduces less when the outdoor ambient temperature decreases during winter. The advantage of a flatter capacity curve is discussed in the next chapter.
- A scroll compressor has a lower noise than a reciprocating compressor and is about 5 dBA lower than the reciprocating compressor of the same size and operated in the same environment.

On the other hand, scroll compressors need computer-aided precision manufacturing and have been used in the field of air conditioning in the United States since 1987 only. There will be new problems to be solved and potential to be improved and developed.

Scroll compressors are gradually replacing reciprocating and rotary compressors in room air conditioners, heat pumps, and packaged units for residential and commercial buildings.

Scroll Chillers

Scroll chillers are often available with a cooling capacity of 10 to 60 tons (35 to 210 kW). They are gradually replacing reciprocating chillers in new and retrofit installations. Scroll chillers can again be divided into air-cooled scroll chillers and water-cooled scroll chillers.

Air-Cooled Scroll Chillers. An air-cooled scroll chiller consists of mainly an air-cooled condenser, two or more scroll compressors, a single or dual refrigerant circuit, a thermostatic expansion valve, a distributor, a direct-expansion liquid cooler, and controls. A direct-expansion liquid cooler can be easily connected to dual refrigerant circuits and multiple compressors. An air-cooled scroll chiller is usually located on the rooftop of an air conditioned building. Chilled water freeze protection should be provided during winter operation in locations where the outdoor temperature falls below 32°F (0°C).

An air-cooled scroll chiller always has a microprocessor-based control of the chilled water temperature leaving the liquid cooler. There are usually two capacity control steps for small air-cooled scroll chillers and four capacity steps for large scroll chillers. Air-cooled scroll chillers are designed to be operated between outdoor temperatures of 40 and 115°F (4.4 and 46°C); with an additional head pressure control damper assembly, they are permited to operate at a lower ambient temperature down to 0°F (−17.8°C).

In addition, there are safety controls of compressor overload protection, high- and low-pressure cutouts, low-temperature freeze protection, oil pressure control, elapse-time meter to record compressor operating hours and number of starts, chilled water flow switch, as well as a diagnostic display.

Air-cooled scroll chillers are rated at a chilled water temperature leaving the liquid cooler at 44°F (6.7°C), $\Delta T = 10$°F (5.6°C), and ambient air temperature entering the air-cooled condenser at 95°F (35°C). ASHRAE/IESNA Standard 90.1-1999 mandates the minimum performance of air-cooled chillers of cooling capacity less than 150 tons (527 kW) with a COP $\geq$ 2.7 (EER $\geq$ 9.22) and an $IPLV_{COP} \geq 2.8$ ($IPLV_{EER} \geq 9.55$). The EER of the air-cooled scroll chillers of one of the leading HVAC&R manufacturers in the United States is 9.1 to 9.6, and $IPLV_{EER}$ is 10 to 12.5.

Water-Cooled Scroll Chillers. A water-cooled scroll chiller is similar in construction and controls to an air-cooled scroll chiller except that a shell-and-tube water-cooled condenser is used instead of an air-cooled condenser. A water-cooled scroll chiller is usually located in the machinery room (refrigeration plant room) of the air conditioning building. According to ARI Standard 550/590-98, air-cooled and water-cooled scroll chillers are rated at the same conditions as in a centrifugal chiller, discussed in Sec. 13.2.

ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency requirements of air-cooled and water-cooled reciprocating, scroll, and screw water chillers as listed in Table 11.7. In Table 11.7, air-cooled reciprocating, scroll, and screw water chillers (with condensers) are specified with the same 2.7 COPs and 2.8 IPLVs when cooling capacities are less than 150 tons (527 kW), and

TABLE 11.7 Water Chillers — Minimum Efficiency Requirements

Equipment type	Size category	Rating condition	Minimum efficiency*	Efficiency as of 10/29/2001*	Test procedure
Air cooled, with condenser, electrically operated	< 150 tons		2.70 COP 2.80 IPLV	2.80 COP 2.80 IPLV	ARI 550 or ARI 590 as appropriate
	≥ 150 tons		2.50 COP 2.50 IPLV		
Air cooled, without condenser, electrically operated	All capacities		3.10 COP 3.20 IPLV	3.10 COP 3.10 IPLV	
Water cooled, electrically operated, positive displacement (reciprocating)	All capacities		3.80 COP 3.90 IPLV	4.20 COP 4.65 IPLV	ARI 590
Water cooled, electrically operated, positive displacement (screw and scrolls)	< 150 tons		3.80 COP 3.90 IPLV	4.45 COP 4.50 IPLV	ARI 550 or ARI 590 as appropriate
	≥ 150 tons and < 300 tons		4.20 COP 4.50 IPLV	4.90 COP 4.95 IPLV	
	≥ 300 tons		5.20 COP 5.30 IPLV	5.50 COP 5.60 IPLV	
Water cooled, electrically operated, centrifugal	< 150 tons		3.80 COP 3.90 IPLV	5.00 COP 5.00 IPLV	ARI 550
	≥ 150 tons and < 300 tons		4.20 COP 4.50 IPLV	5.55 COP 5.55 IPLV	
	≥ 300 tons		5.20 COP 5.30 IPLV	6.10 COP 6.10 IPLV	

*The requirements do not apply for chillers used in low-temperature applications where the design leaving temperature is less than or equal to 40°F (≤4.4°C).

***Source*:** ASHRAE/IESNA Standard 90.1-1999. Reprinted by permission.

are specified with 2.5 COP and also IPLVs when cooling capacity is greater than or equal to 150 tons (527 kW). For water-cooled water chillers, when cooling capacities are less than 150 tons (527 kW), minimum efficiency requirements 3.8 COP and 3.9 IPLV are the same for reciprocating, scroll, and screw water chillers. According to the data for water-cooled scroll chillers of a capacity range between 20 to 60 tons (70 to 211kW), a leading HVAC&R manufacturer in late 1990s in the United States listed their COPs at between 3.86 and 4.04 corresponding to a kW/ton between 0.91 and 0.87.

11.11 SCREW REFRIGERATION SYSTEMS

Types of Screw Refrigeration System

Currently used screw refrigeration systems are mostly screw chillers. Screw chillers are either water-cooled or air-cooled. Energy-efficient water-cooled screw chillers are widely used to replace the water-cooled reciprocating chillers employed in thermal storage systems. More energy-efficient and more compact air-cooled screw chillers are used to replace air-cooled centrifugal chillers and air-cooled reciprocating chillers in new and retrofit installations.

Water-cooled chillers often have a cooling capacity between 50 and 500 tons (176 and 1758 kW). A water-cooled screw chiller has five main components, as shown in Fig. 11.28. In addition to a screw compressor, a shell-and-tube liquid cooler as evaporator, a shell-and-tube water-cooled condenser, and orifice plates or float valves as throttling device, an oil cooler is used to cool and supply oil to the compressor. According to ARI Standard 550/590-98, water-cooled and air-cooled screw chillers can be rated the same as a centrifugal chiller, discussed in Sec. 13.2.

Screw Compressors

Screw compressors are designed to be incorporated with oil injection and have been used in industrial applications since the 1950s. Screw compressors can be classified as twin-screw compressors or single-screw compressors.

Twin-Screw Compressor. Twin-screw compressors are more widely used than single-screw compressors because their isentropic efficiency is slightly higher than that of single-screw compressors (about 3 to 4 percent). A typical twin-screw compressor, shown in Fig. 11.29*a*, consists of a four-lobe male rotor and a six-lobe female rotor (4 + 6), or five-lobe male rotor and a seven-lobe female rotor (5 + 7), a housing with suction and discharge ports, and a sliding valve for capacity control. When the lobes are separated at the suction port, as shown in Fig. 11.29*b*, vapor refrigerant is drawn in and the intake process continues until the interlobe space is out of contact with the suction port. The volume trapped in the interlobe space within the meshing point is compressed during successive rotations of the rotor. When the interlobe space makes contact with the discharge port, the compressed gas discharges through the outlet.

The ratio of the volume of vapor trapped during the intake process V_{in}, ft^3 (m^3), to the volume of trapped gaseous refrigerant discharged V_{dis}, ft^3 (m^3), is called the *built-in volume ratio* V_i. The application of variable V_i instead of fixed V_i eliminates the losses associated with over- or undercompression, especially during part-load operation.

Oil injection plays an important role in twin-screw compression. Injected oil effectively cools the compressor and results in a lower discharge temperature. A small clearance of 0.0005 in. (0.0127 mm) between lobes and the sealing oil minimizes leakage loss. Another important function of the injected oil is lubrication. The cooling capacity of twin-screw compressors ranges from 20 to 1300 tons (70 to 4570 kW). Cooling capacities between 50 and 350 tons (176 and 1230 kW) are most widely used. The pressure ratio of a screw compressor can be as high as 20 : 1 (single-stage). HCFC-22, HFC-134a, and HFC-407C for comfort air conditioning, and R-717 for industrial applications, are the most widely used refrigerants for twin-screw compressors.

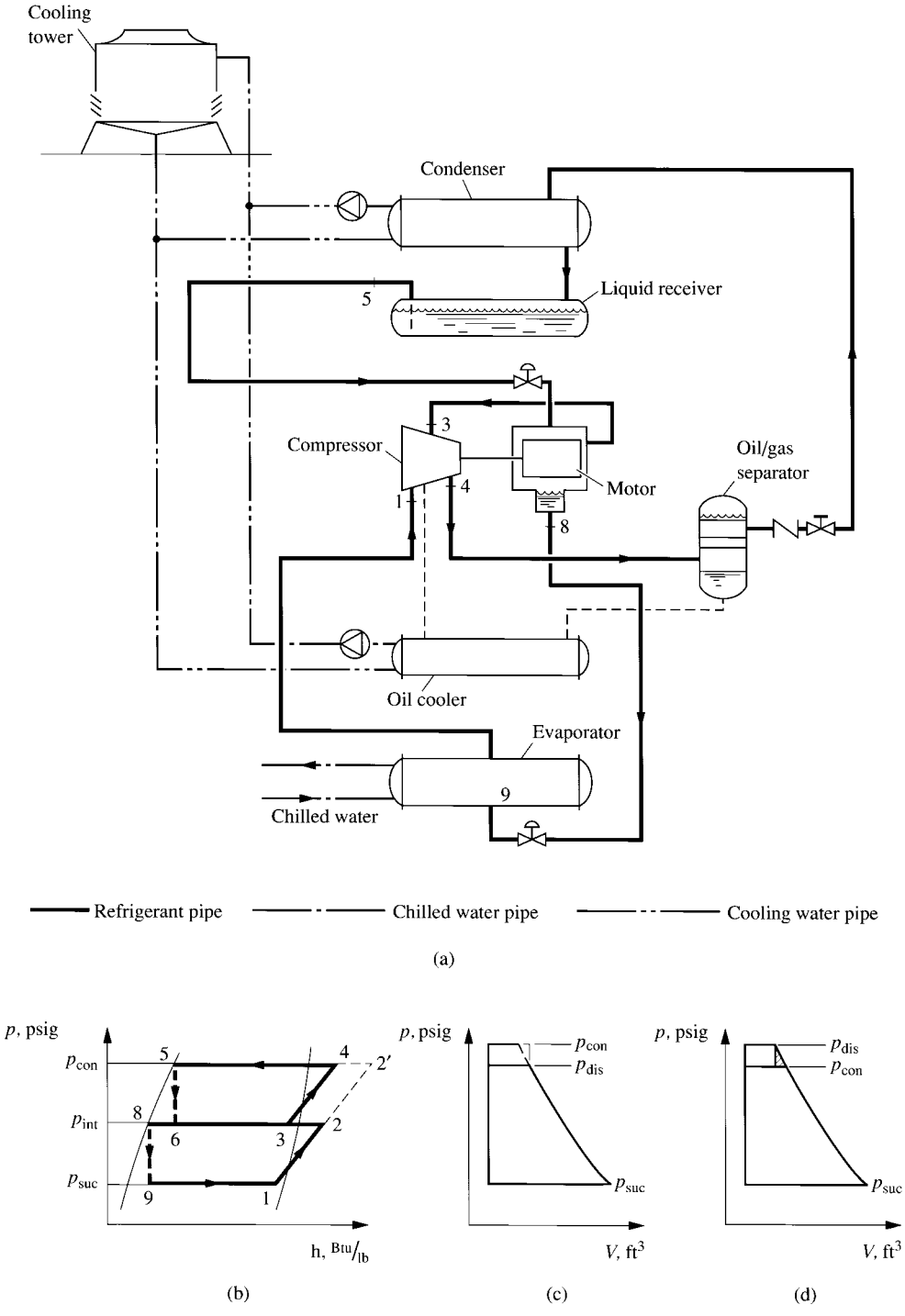


FIGURE 11.28 A typical screw (twin-screw compressor refrigeration system): (*a*) schematic diagram; (*b*) refrigeration cycle; (*c*) undercompression; (*d*) overcompression.

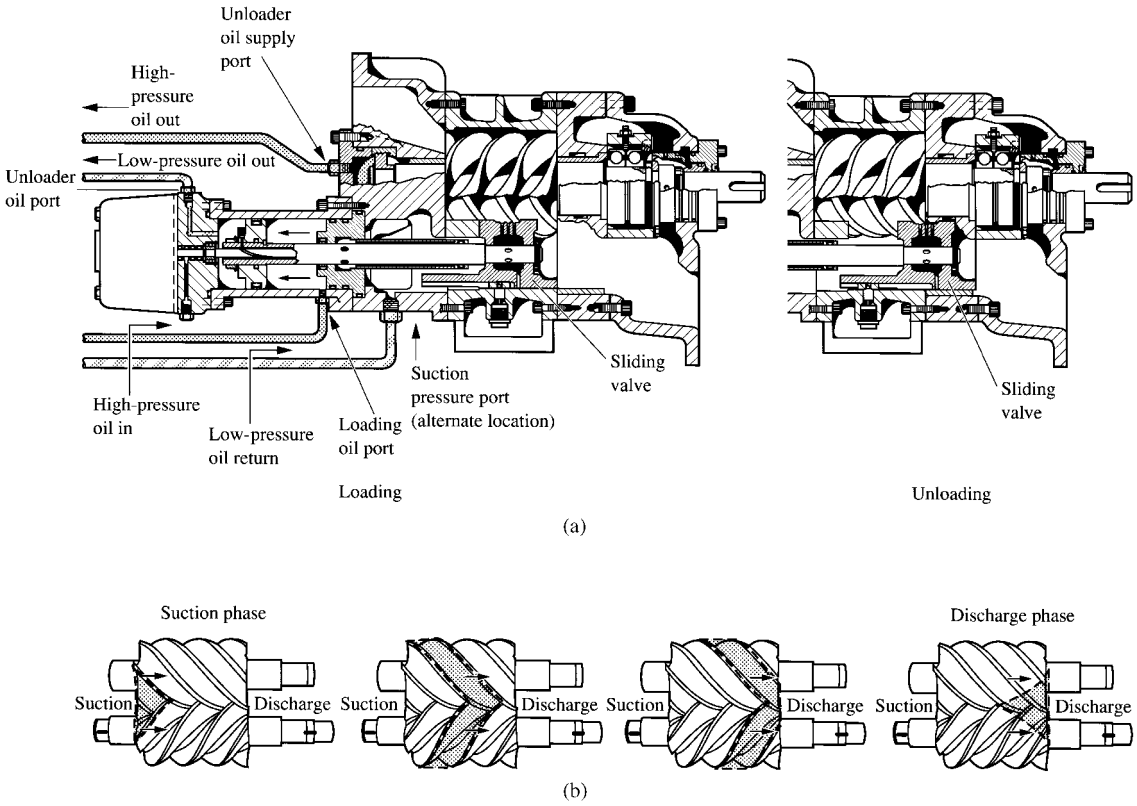


FIGURE 11.29 Twin-screw compressor: (*a*) loading and unloading of screw compressor; (*b*) suction-compression-discharge processes. (*Source: Dunham-Bush Compressors. Reprinted by permission.*)

Single-Screw Compressor. The single-screw compressor was developed in the 1960s and has been used in refrigeration systems since the 1970s. The single-screw compressor has a single main rotor and two gate rotors, as shown in Fig. 11.30. As the main rotor rotates, one of the flutes opens to the suction port and is filled with suction vapor until its suction end meshes with the gate rotor. The discharge end of this flute is covered by the rotor casing. When the main rotor turns, the meshing of gate rotor *A* with the flute compresses the trapped gas and raises its pressure. When the discharge end of this flute opens to the discharge port, compression stops and the hot gas is discharged. In a single-screw compressor, compression occurs simultaneously in both the top and bottom of the main rotor.

Performance of Twin-Screw Compressor

In a typical twin-screw compressor using HCFC-22 as refrigerant with $V_i = 2.6$, η_v drops from 0.92 to 0.83 and η_{isen} decreases from 0.82 to 0.60 when the pressure ratio p_{dis}/p_{suc} increases from 3 to 8. A twin-screw compressor is more energy-efficient than a reciprocating or a scroll compressor of the same size, operating at the same conditions, and is less energy-efficient than a centrifugal compressor using a water-cooled condenser. Some recently developed screw compressors can even tolerate liquid slugging.

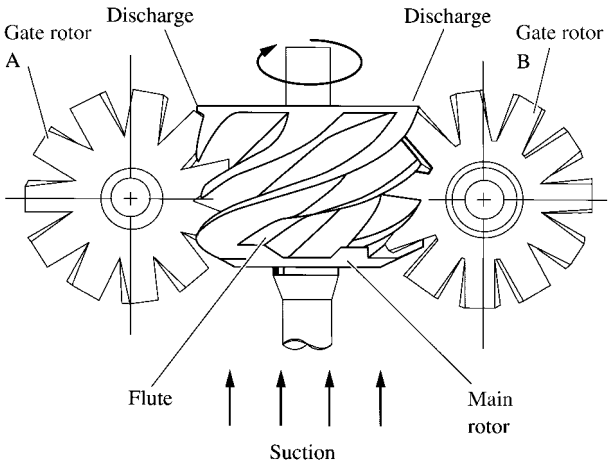


FIGURE 11.30 Single-screw compressor.

Bivens et al. (1997) tested of a screw chiller of 150-ton (527-kW) cooling capacity with both shell-and-tube evaporator and condenser with water inside the tubes and using a zeotropic refrigerant mixture of HFC-32/HFC-125/HFC-134a (30/10/60 percent). The result was that the circulating concentration shift of HFC-32/HFC-125/HFC-134a was 41/17/42 percent; and the cooling capacity of the screw chiller dropped 36 percent, input power increased 14 percent, and COP decreased 44 percent when it was compared with HCFC-22. Instead of a flooded evaporator, when a screw chiller employed a direct-expansion evaporator with a refrigerant blend HFC-32/HFC-125/HFC-134a (30/10/60), the screw chiller's cooling capacity dropped 7 percent, power input increased 8 percent, and COP decreased 14 percent when compared with HCFC-22.

Oil Cooling

As shown in Fig. 11.28, the hot-gas and oil mixture discharged from the screw compressor enters an oil-gas separator, in which oil is separated from the gaseous refrigerant. The oil is cooled in the oil cooler, flows through a filter, and divides into three circuits to lubricate the bearings, provides hydraulic power to the capacity control system, and flows to the compressor. Oil in the compressor provides a rotor-to-rotor and rotor-to-housing seal, lubricates the rotor driving mechanism, and cools the refrigerant during compression.

In an oil cooler, oil is usually cooled by cooling water from the cooling tower or evaporative cooler. It is sometimes cooled by river, lake, or well water, or by direct expansion of liquid refrigerant.

Capacity Control

Cooling capacity is controlled by moving a sliding valve between the discharge end of the male rotor and the discharge port, as shown in Fig. 11.29*a.* The sliding valve is driven by the oil pressure. When the sliding valve moves toward the discharge port, it opens a bypass passage on the rotor housing. A portion of the trapped gaseous refrigerant returns to the suction port through this bypass passage, so the mass flow rate of refrigerant and thus the cooling capacity are reduced. This kind of capacity control is a stepless continuous modulation.

Variable Volume Ratio

There are no discharge valves in a twin-screw compressor. If the pressure drop in the discharge line is ignored, a screw compressor using built-in volume ratio V_i may cause over- or undercompression when the discharge pressure from the compressor p_{dis} is different from the condensing pressure p_{con}, both in psig (kPa · g). Figure 11.28*c* shows undercompression. Undercompression causes a volume of gas at condensing pressure to reenter the trapped volume at the beginning of the discharge process. Figure 11.28*d* shows overcompression. The discharged gas reexpands to match the condensing pressure.

In the late 1980s, twin-screw compressors with variable volume ratios were developed. The common arrangement includes a split sliding valve, which consists of a sliding valve and a movable slide. As the slide moves back and forth, the radial discharge port is relocated and thus varies the volume ratio to meet the optimum requirements without over- and undercompression. For a screw refrigeration system in which suction and condensing pressures change frequently, efficiency can be improved by up to 20 percent by using a variable volume ratio twin-screw compressor instead of a built-in volume ratio.

Economizer

In Fig. 11.28*b*, if a small portion of liquid refrigerant at point 5 is flashed at an intermediate pressure p_{int} so that the remaining refrigerant is cooled to saturated condition at intermediate pressure, point 8, such an arrangement significantly increases the refrigeration effect of the cycle. At the same time, the flashed vapor refrigerant is introduced to the partially compressed gas at p_{int}. The compression work needed for the flashed vapor at p_{int} is less than the compression work needed if the liquid refrigerant is flashed at the evaporator at p_{suc}.

Figure 11.28*a* is a schematic diagram of an economizer. Liquid refrigerant from the receiver at point 5 flows through the throttling device and enters the motor shell. In the motor, a portion of liquid refrigerant at p_{int} is flashed into vapor to cool the motor windings and the remaining liquid refrigerant to saturated condition point 8. The flashed vapor is then introduced to the compressed gaseous refrigerant at p_{int} in the compressor. The remaining liquid refrigerant flows through another throttling device. Its pressure drops to the evaporating pressure p_{ev} and enters the evaporator at point 9. In the evaporator, all liquid refrigerant is vaporized. Vapor refrigerant from the evaporator is extracted by the compressor at suction pressure p_{suc} denoted by point 1.

Controls

In a water-cooled screw chiller, a DDC system includes the following:

Status monitoring and specific controls:

- Entering and leaving chilled water temperatures
- Evaporator and condenser refrigerant temperatures and pressures
- Compressor suction temperature
- Electric current limit

Safety controls:

- Motor overloading, high- and low-pressure cutouts, water freeze protection, and oil pressure
- Diagnostics: operating conditions and diagnostics display

Minimum Performance

ASHRAE/IESNA Standard 90.1-1999 mandates different minimum efficiency requirements for water-cooled screw water chillers as listed in Table 11.7:

	Minimum Efficiency		Efficiency as of 10/29/2001	
Sizes	COP*	IPLV	COP	IPLV
< 150 tons	3.8	3.9	4.45	4.5
≥ 150 tons and < 300 tons	4.2	4.5	4.9	4.95
≥300 tons	5.2	5.3	5.5	5.6

* For a COP of 5.2, the equivalent is 0.68 kW/ton.

One of the U.S. leading HVAC&R manufacturer offered an energy use of as low as 0.59 kW/ton (5.96 COP) for their 150- to 360-ton (527- to 1266-kW) water-cooled screw chillers at full-load and 0.52 kW/ton (6.76 COP) at part-load. These energy consumption indices were even less than the values projected as of October 29, 2001.

Air-Cooled Screw Chillers

Air-cooled screw chillers are widely used in locations where water is scarce and the use of city water as makeup water for cooling towers is prohibited by local codes. Because the air-cooled screw chillers are more energy-efficient and more compact than air-cooled centrifugal chillers, and more energy-efficient than recprocating and scroll chillers, air-cooled screw chillers become the dominant air-cooled chillers in a cooling capacity range of 50 to 400 tons (176 to 1400 kW).

System Components. An air-cooled screw chiller consists of mainly two, three, or four screw compressors, dry-expansion liquid cooler, air-cooled condenser, expansion valve, and a DDC unit controller. There are two independent refrigerant circuits; each has its own compressors, expansion valve, and refrigerant distributor and liquid cooler. This kind of setup provides greater reliability in case there is a compressor problem.

Electric Expansion Valves. Some manufacturers offer electric expansion valves that can work over a wider operating range, conditions such as at low ambient air temperature and at a high refrigeration load, causing thermostatic expansion valve (TXV) hunting. Operating at a lower ambient always saves energy. An electric expansion valve provides more precise control and gives a better part-load performance than does TXV.

Microprocessor-Based DDC. A DDC unit controller and the correponding functional, optimizing, and safety controls as well as diagnostics similar to that discussed for water-cooled screw chillers are always equipped in air-cooled screw chillers.

System Performance. Air-cooled screw chillers are often manufactured at a capacity of 50 to 400 tons (176 to 1400 kW). They are rated at a chilled water leaving evaporator temperature at 44°F (6.7°C) and a cooling air temperature entering the air-cooled condenser at 95°F (35°C). In ASHRAE/IESNA Standard 90.1-1999, the minimum performance of air-cooled chillers with a cooling capacity < 150 tons (527 kW) is the same as that for air-cooled scroll chillers of the same cooling capacity (see Sec. 11.10). For air-cooled chillers in a cooling capacity ≥ 150 tons, COP ≥ 2.5 and IPLV ≥ 2.5 (EER ≥ 8.5 and $IPLV_{EER}$ ≥ 8.5). In the early 1990s, the EER and $IPLV_{EER}$ of an air-cooled screw chiller of a U.S. leading HVAC&R manufacturer were 10.0 (2.93 COP) and 13.2 (3.87 $IPLV_{COP}$), respectively.

Location of Installation. Air-cooled screw chillers are usually located on the roof outdoors. The key point for efficient operation is to have sufficient clearance between the condensing coils and the surroundings, and between units, according to the manufacturers' specifications. If the chiller is shut down during winter, water should be drained from the evaporator in areas where outdoor temperatures fall below 32°F (0°C). Otherwise, glycol should be added to the chilled water for freeze protection.

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CHAPTER 12
HEAT PUMPS, HEAT RECOVERY, GAS COOLING, AND COGENERATION SYSTEMS

12.1 BASICS OF HEAT PUMP AND HEAT RECOVERY 12.1
- Heat Pumps 12.1
- Heat Pump Cycle 12.2
- Classification of Heat Pumps 12.3
- HVAC&R Heat Recovery Systems 12.3
- Heat Balance and Building Load Analysis 12.4

12.2 AIR-SOURCE HEAT PUMP SYSTEMS 12.5
- System Components 12.6
- Suction Line Accumulator 12.8
- Operating Modes 12.9
- System Performance 12.9
- Cycling Loss and Degradation Factor 12.11
- Minimum Performance 12.12
- Defrosting 12.12
- Controls 12.13
- Capacity and Selection 12.13

12.3 GROUNDWATER HEAT PUMP SYSTEMS 12.13
- Groundwater Systems 12.14
- Groundwater Heat Pump System for a Hospital 12.14
- Groundwater Heat Pump Systems for Residences 12.15

12.4 GROUND-COUPLED AND SURFACE WATER HEAT PUMP SYSTEMS 12.17

12.5 AIR-TO-AIR HEAT RECOVERY 12.19
- Types of Air-to-Air Heat Recovery 12.19
- Effectiveness 12.19
- Fixed-Plate Heat Exchangers 12.20
- Runaround Coil Loops 12.21
- Rotary Heat Exchangers 12.21
- Heat Pipe Heat Exchangers 12.22
- Comparison between Various Air-to-Air Heat Exchangers 12.25

12.6 GAS COOLING AND COGENERATION 12.25
- Gas Cooling 12.25
- Cogeneration 12.25
- Gas-Engine Chiller 12.27
- Gas Engines 12.27
- Exhaust Gas Heat Recovery 12.28
- Engine Jacket Heat Recovery 12.28
- Cogeneration Using a Gas Turbine 12.29

REFERENCES 12.29

12.1 BASICS OF HEAT PUMP AND HEAT RECOVERY

Heat Pumps

A heat pump extracts heat from a heat source and rejects heat to air or water at a higher temperature. During summer, the heat extraction, or refrigeration effect, is the useful effect for cooling, whereas in winter the rejected heat alone, or rejected heat plus the supplementary heating from a heater, forms the useful effect for heating.

A heat pump is a packaged air conditioner or a packaged unit with a reversing valve or other changeover setup. A heat pump has all the main components of an air conditioner or packaged unit: fan, filters, compressor, evaporator, condenser, short capillary tube, and controls. The apparatus for changing from cooling to heating or vice versa is often a reversing valve, in which the refrigerant flow to the condenser is changed to the evaporator. Alternatively, air passage through the evaporator may be changed over to passage through the condenser. A supplementary heater is often provided when the heat pump capacity does not meet the required output during low outdoor temperatures.

A heat pump system consists of heat pumps and piping work; system components include heat exchangers, heat source, heat sink, and controls to provide effective and energy-efficient heating and cooling operations. HCFC-22, HFC-134a, and HFC-407C are the most widely used refrigerants in new heat pumps. According to the data in the EIA's Commercial Buildings Characteristics, for the 57 billion ft^2 (5.3 m^2) of air conditioned commercial building floor area in the United States in 1992, the use of heat pumps for heating and cooling was about 15 percent (by floor space).

Heat Pump Cycle

A heat pump cycle comprises the same processes and sequencing order as a refrigeration cycle except that the refrigeration effect $q_{1'4}$ or q_{rf}, and the heat pump effect $q_{2'3'}$, both in Btu/lb (J/kg), are the useful effects, as shown in Fig. 12.1. As defined in Eqs. (9.7) and (9.9), the coefficient of performance of a refrigeration system COP_{ref} is

$$\text{COP}_{ref} = \frac{h_{1'} - h_4}{W} = \frac{q_{1'4}}{W_{in}} \tag{12.1}$$

where $h_4, h_{1'}$ = enthalpy of refrigerant entering and leaving evaporator, respectively, Btu/lb (J/kg)
W_{in} = work input, Btu/lb (J/kg)

The coefficient of performance of the heating effect in a heat pump system COP_{hp} is

$$\text{COP}_{hp} = \frac{q_{2'3'}}{W_{in}} \tag{12.2}$$

and the useful heating effect $q_{2'3'}$ can be calculated as

$$q_{2'3'} = h_{2'} - h_{3'} = h_{1'} - h_4 + h_{2'} - h_{1'} \tag{12.3}$$

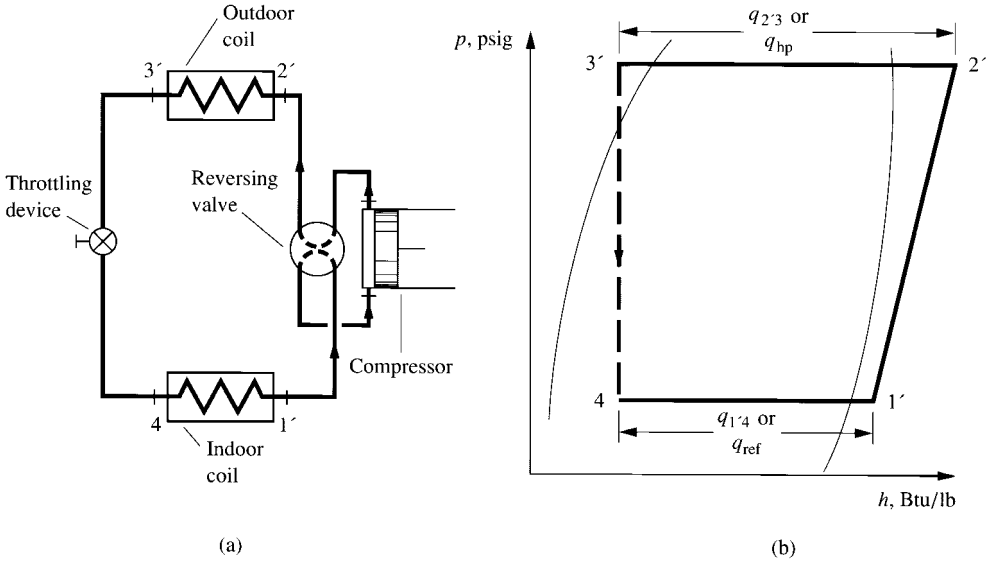


FIGURE 12.1 Heat pump cycle: (*a*) schematic diagram; (*b*) cycle on *p*-*h* diagram.

where $h_{2'}$ = enthalpy of hot gas discharged from compressor, Btu/lb (J/kg)
$h_{3'}$ = enthalpy of subcooled liquid leaving condenser, Btu/lb (J/kg)

Here polytropic compression is a real and irreversible process. Both the subcooling of the liquid refrigerant in the condenser and the superheating of the vapor refrigerant after the evaporator increase the useful heating effect $q_{2'3'}$. Excessive superheating, which must be avoided, leads to a too-high hot-gas discharge temperature and to a lower refrigeration capacity in the evaporator.

Classification of Heat Pumps

According to the types of heat sources from which heat is absorbed by the refrigerant, currently used heat pump systems can be mainly classified into two categories: air-source and water-source heat pump systems. Water-source heat pumps can again be subdivided into water-source, groundwater, ground-coupled, and surface water heat pump systems. Water-source heat pump systems are discussed in Chap. 29.

Heat pump systems are often energy-efficient cooling/heating systems. Many new technologies currently being developed, such as engine-driven heat pumps, may significantly increase the system performance factor of the heat pump system. Ground-coupled heat pumps with direct-expansion ground coils provide another opportunity to increase the COP of the heat pump system.

HVAC&R Heat Recovery Systems

An HVAC&R heat recovery system converts waste heat or cooling from any HVAC&R process to useful heat or cooling. Here heat recovery is meant in a broad sense. It includes both waste heat and cooling recovery. An HVAC&R heat recovery system includes the following:

- The recovery of internal heat loads—such as heat energy from lights, occupants, appliances, and equipment inside the buildings—by reclaiming the heat rejected at the condenser and absorber of the refrigeration systems
- The recovery of heat from the flue gas of the boiler
- The recovery of heat from the exhaust gas and water jacket of the engine that drives the HVAC&R equipment, especially engine-driven reciprocating vapor compression systems
- The recovery of heat or cooling from the exhaust air from air conditioning systems

Although heat pump systems sometimes are used to recover waste heat and convert it into a useful effect, a heat recovery system is different from a heat pump system in two ways:

- In a heat pump system, there is only one useful effect at a time, such as the cooling effect in summer or the heating effect during winter. In a heat recovery system, both its cooling and heating effects may be used simultaneously.
- From Eq. (12.2), the coefficient of performance of the useful heating effect in a heat pump is $\text{COP}_{\text{hp}} = q_{2'3'}/W_{\text{in}}$, whereas for a heat recovery system, the coefficient of performance COP_{hr} is always higher if both cooling and heating are simultaneously used and can be calculated as

$$\text{COP}_{\text{hr}} = \frac{q_{1'4} + q_{2'3'}}{W_{\text{in}}} \tag{12.4}$$

A heat pump is an independent unit. It can operate on its own schedule, whereas a heat recovery system in HVAC&R is usually subordinate to a refrigeration system or to some other system that produces the waste heat or waste cooling.

Heating or cooling produced by a heat recovery system is a by-product. It depends on the operation of the primary system. A heat recovery system can use waste heat from condenser water for winter heating only if the centrifugal chiller is operating.

A centrifugal chiller that extracts heat from a surface water source (e.g., a lake) through its evaporator and uses condensing heat for winter heating is a heat pump, not a heat recovery system. Heat recovery systems that are subordinate to a centrifugal or absorption refrigeration system are discussed in Chaps. 13 and 14, respectively. Recovery of waste heat from industrial manufacturing processes to provide heating shows great potential for saving energy. These heat recovery systems must be closely related to the specific requirements of corresponding manufacturing processes and are not discussed here.

Heat Balance and Building Load Analysis

The building load consists of transmission gain or loss, solar radiation, ventilation load and infiltration load, people, electric lights, appliances (or equipment), and heat gains from fans and pumps. Building load is actually the load of the cooling or heating coil in an air-handling unit, or DX coil in a packaged unit or air conditioner. On hot summer days, solar radiation and the latent ventilation load must be included in the building load. However, both sunny and cloudy days may occur in cold weather; therefore, building load is calculated and analyzed with and without solar radiation in cold weather, especially for the control zones facing south in a building in northern latitudes, or facing north in southern latitudes, where solar radiation is often a primary cooling load on sunny days.

Figure 12.2 shows the building load analysis of a typical floor of a multistory building without solar radiation in winter. In this figure, line *ABCDE* represents the building load curve at various outdoor temperatures T_o, in °F (°C). Point *A* on this curve represents the summer design refrigeration load Q_{rl}, in MBtu/h (kW). During summer design load, all the space cooling loads are offset by the cold supply air, and the condensing heat of the refrigeration system is rejected to the cooling tower.

When the outdoor temperature T_o drops below 75°F (23.9°C) on the left side of point *B* on the building load curve, then the following things occur:

- The perimeter zone may suffer a transmission loss.
- Solar radiation is excluded.
- The latent load of the outdoor ventilation air is no longer included in the building load because the outdoor air is often drier than the space air.

If the outdoor temperature $T_o < T_F$, hot condenser water is supplied to the heating coils in the perimeter zone to satisfy the heating load. Here T_F indicates the outdoor temperature at point *F*.

When the outdoor temperature drops to T_D, the heat recovered from the interior zone plus the power input to the compressor is exactly equal to the heating load of the perimeter zone. No supplementary heating is needed. Point *D* is called the break-even point of the building. When $T_o < T_D$, supplementary heating is necessary to maintain a desirable space temperature.

As the outdoor temperature falls to T_H, the cooling coil load in the interior zone becomes zero. The refrigeration compressors are turned off. No recovery of condensing heat is possible. When $T_o \leq T_H$, all the heat needed for the perimeter zone will be provided by the supplementary heating.

The area of triangle *FGH* indicates the heat energy recovered from the internal loads in the interior zone, which is used to offset the heat losses in the perimeter zone by means of hot condenser water. Similar building load curves with solar radiation for the entire building and for the south-facing zones in the building should be calculated and analyzed in winter. For building load with solar radiation, break-even point *D* will move to a lower T_o. Recovered heat will be greater, and supplementary heating will be less. For south-facing zones in buildings in northern

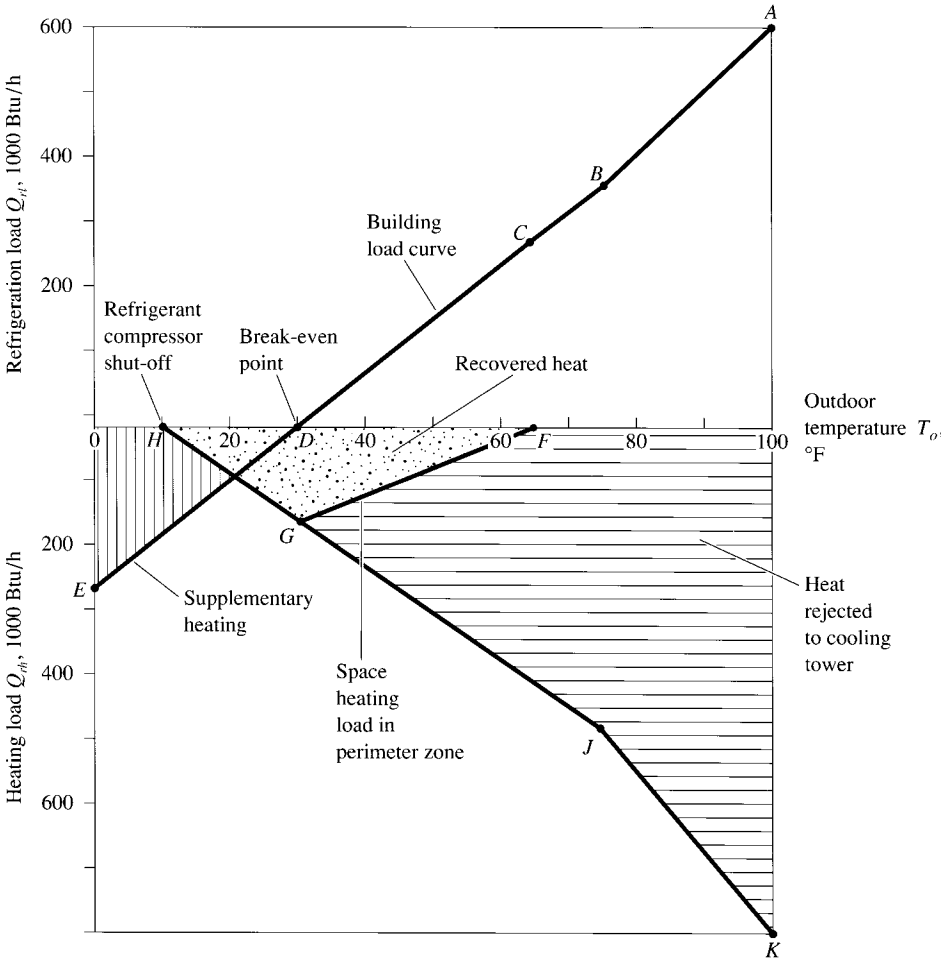


FIGURE 12.2 Building load analysis of a typical floor in a multistory building without solar radiation.

climates, cold supply air may be required during sunny days in the perimeter zone even in winter.

12.2 AIR-SOURCE HEAT PUMP SYSTEMS

In an air-source heat pump system, outdoor air acts as a heat source from which heat is extracted during heating, and as a heat sink to which heat is rejected during cooling. Since air is readily available everywhere, air-source heat pumps are the most widely used heat pumps in residential and many commercial buildings. The cooling capacity of most air-source heat pumps is between 1 and 30 tons (3.5 and 105 kW).

Air-source heat pumps can be classified as individual room heat pumps and packaged heat pumps. Individual room heat pumps serve only one room without ductwork. Packaged heat pumps can be subdivided into rooftop heat pumps and split heat pumps.

System Components

Most air-source heat pumps consist of single or multiple compressors, indoor coils through which air is conditioned, outdoor coils where heat is extracted from or rejected to the outdoor air, capillary tubes, reversing valves that change the heating operation to a cooling operation and vice versa, an accumulator to store liquid refrigerant, and other accessories. A typical rooftop packaged heat pump that uses HCFC-22 as refrigerant is shown in Fig. 12.3. In this heat pump, dual circuits, consisting of two compressors, two indoor coils, two outdoor coils, two throttling devices, and two reversing valves, are often used for better capacity control and better defrosting control for larger heat pumps.

Compressor. Reciprocating and scroll compressors are widely used in heat pumps. In Bucher et al. (1990) the median service life of the compressor in a heat pump is 14.5 years, and the median service life of a heat pump system is 19 years, depending on the conditions of operation and

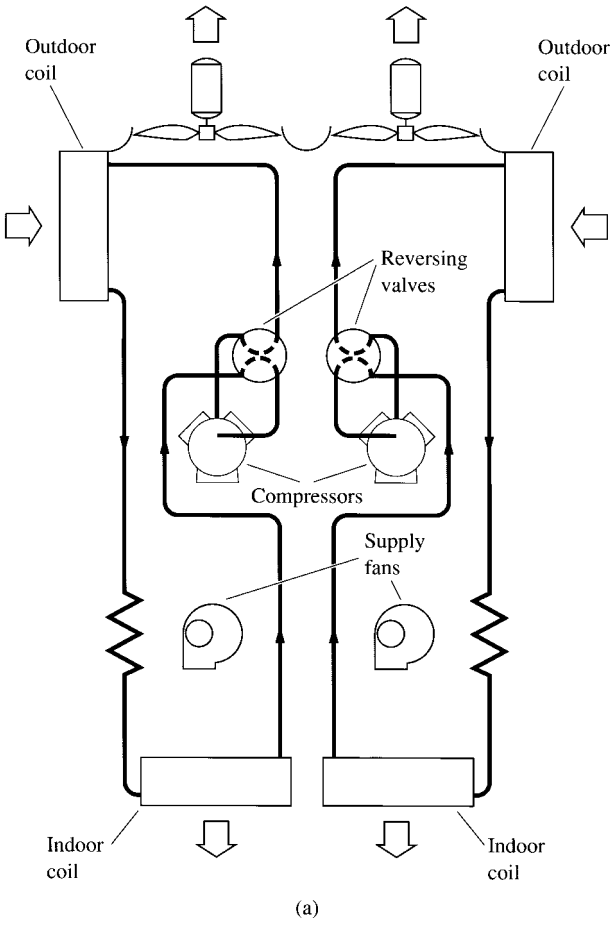
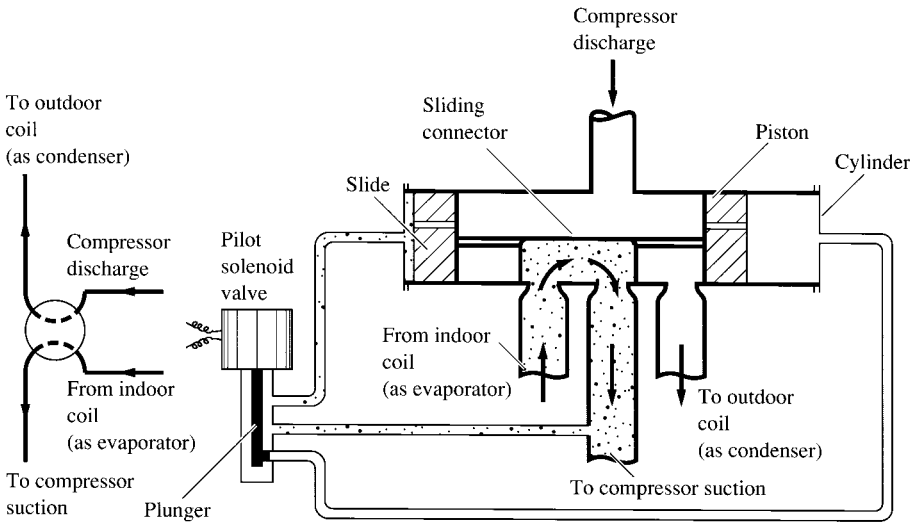
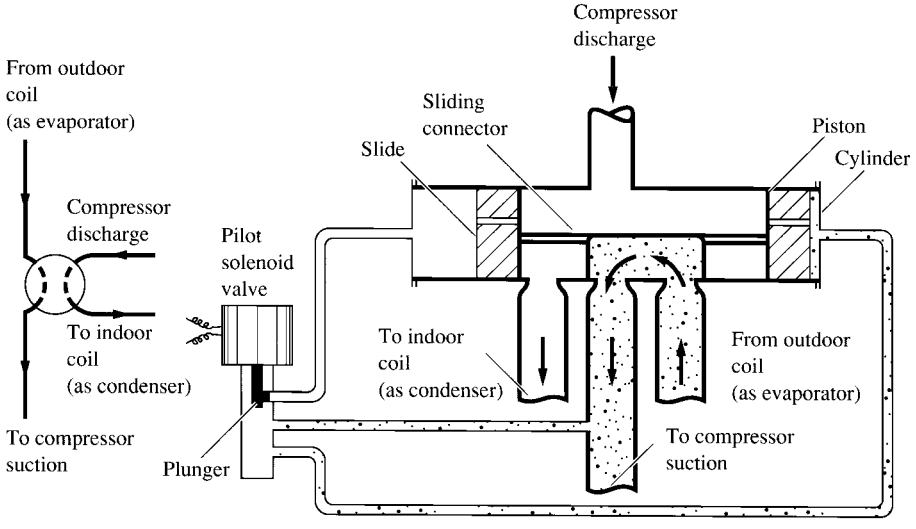


FIGURE 12.3 A typical rooftop heat pump: (*a*) schematic diagram; (*b*) reversing valve, cooling mode; (*c*) reversing valve, heating mode.



(b)



(c)

FIGURE 12.3 (*Continued*)

maintenance. Median service life is the age at which 50 percent of the units have been removed from service and 50 percent remain in service.

Indoor Coil. In an air-source heat pump, the indoor coil is not necessarily located inside the building. The indoor coil in a rooftop packaged heat pump is mounted on the rooftop. However, an indoor coil always heats and cools the indoor supply air. During cooling operation, the indoor

coil acts as an evaporator. It provides the refrigeration effect to cool the mixture of outdoor and recirculating air when the heat pump is operating in the recirculating mode. During heating operation, the indoor coil acts as a condenser. The heat rejected from the condenser raises the temperature of the conditioned supply air. For heat pumps using halocarbon refrigerants, the indoor coil is usually made from copper tubing and corrugated aluminum fins.

Outdoor Coil. The outdoor coil acts as a condenser during cooling and as an evaporator to extract heat from the outdoor atmosphere during heating. When an outdoor coil is used as a condenser, a series-connected subcooling coil often subcools the refrigerant for better system performance. An outdoor coil always deals with outdoor air, whether it acts as a condenser or an evaporator. Like the indoor coil, an outdoor coil is usually made of copper tubing and aluminum fins for halocarbon refrigerants. Plate or spine fins are often used instead of corrugated fins to avoid clogging by dust and foreign matter.

Reversing Valve. Reversing valves are used to guide the direction of refrigerant flow when cooling operation is changed over to heating operation or vice versa. The rearrangement of the connections between four ways of flow — compressor suction, compressor discharge, evaporator outlet, and condenser inlet — causes the functions of the indoor and outdoor coils to reverse. It is therefore called a four-way reversing valve.

A typical four-way reversing valve consists of a hollow cylinder with an internal slide to position the flow paths of the refrigerant. The slide is driven by the differential between the discharge and suction pressure. Both are introduced to opposite ends of the cylinder by a pilot valve that is energized by a solenoid coil. The operation of this typical four-way reversing valve is shown in Fig. 12.3*b* and *c*.

A reversing valve is a highly reliable system component. The efficiency losses altogether including leakage, heat transfer, and the pressure drop across the reversing valve cause a decrease of 4 to 7 percent in heat pump performance. Other accessories, such as a filter dryer, sight glass, strainer, liquid level indicator, solenoid valves, and manual shutoff valves as well as refrigerant piping are the same as described in Chap. 11.

Suction Line Accumulator

An accumulator is usually installed on the suction line prior to the inlet of the compressor in a heat pump system to store the liquid refrigerant prior to the defrosting process. Bivens et al. (1997) conducted tests on a 2.5-ton (8.8-kW) split residential heat pump with a scroll compressor and an accumulator using HFC-407C and HCFC-22 as refrigerant. Only a slight change in concentration shift occurred when there was no liquid refrigerant in the accumulator. There were significant changes in concentration shift during heating when the temperatures in the evaporator and accumulator were lower. The accumulator contained about 30 percent of the refrigerant as liquid during high-temperature heating (outdoor air temperature at 47°F, or 8.3°C) and about 50 percent of the refrigerant as liquid during low-temperature heating (outdoor air temperature at 17°F, or −8.3 °C). Compared with HCFC-22 as refrigerant, the capacity of the HFC-407C heat pump increased to 101 to 105 percent of the HCFC-22 capacity during high-temperature heating, and increased to 106 to 109 percent of the HCFC-22 capacity during low-temperature heating. The COP of the HFC-407C heat pump dropped to 0.96 to 0.97 of the HCFC-22 COP during cooling and dropped to 0.94 and 0.95 of the HCFC-22 COP during heating.

Nutter et al. (1996) conducted experiments on a scroll heat pump of 3-ton (10-kW) cooling capacity. The heat pump was operated at heating mode of ambient air of 1.7°F (−16.8°C) dry-bulb temperature and 80 percent relative humidity in a psychrometric room. Test results showed that the COP is higher for the heat pump without an accumulator first until a power surge at 40-min operation. This caused the integrated cyclic COP for the heat pump without an accumulator of 2.08 which fell below that of the COP of the heat pump system with an accumulator of 2.12. Such a

phenomenon may be the result of entrained liquid refrigerant that began to adversely affect the compressor performance by reducing discharge temperature and pressures.

Operating Modes

The operation of an air-source heat pump can be divided into cooling mode and heating mode.

Cooling Mode. When the discharge air temperature sensor detects an increase in the air temperature above a predetermined limit at the exit of the indoor coil, cooling is required in the air-source heat pump. The direct digital controller (DDC) unit deenergizes the pilot solenoid valve, as shown in Fig. 12.3*b*. The low-pressure suction vapor is now connected to the left-hand end of the slide, and the high-pressure hot gas between the pistons of the slide then pushes the piston toward the left end of the cylinder. The indoor coil now acts as an evaporator and extracts heat from the conditioned air flowing through the indoor coil. After evaporation, vapor refrigerant from the indoor coil passes through the sliding connector of the slide and flows to the suction line. Hot gas discharged from the compressor is led to the outdoor coil, which now acts as a condenser.

An economizer cycle can be used when an outdoor air sensor detects the outdoor temperature dropping below a specific limit during cooling mode. The DDC unit controller opens and modulates the outdoor damper to admit cold outdoor air to maintain a preset discharge air temperature.

Heating Mode. When the discharge air sensor detects a drop in air temperature below a predetermined limit at the exit of the indoor coil, heating is required. The DDC unit controller energizes the pilot solenoid valve as shown in Fig. 12.3*c*. The plunger of the solenoid valve moves upward and connects the right-hand end of the slide to the low-pressure suction line. High-pressure hot gas then pushes the pistons toward the right-hand end of the cylinder. Consequently, the hot gas from the compressor is discharged to the indoor coil, which now acts as a condenser. Heat is rejected to the recirculating air, and then the hot gas is condensed into liquid form. Liquid refrigerant flows through the capillary tube and then vaporizes in the outdoor coil, which extracts heat from outdoor air. The outdoor coil now acts as an evaporator.

Electric heating can be used for supplementary heating. When the discharge air temperature sensor detects a drop in air temperature further below preset limits, the electric heater can be energized in steps to maintain the required discharge air temperature. Supplementary heating is energized only when the space heating load cannot be offset by the heating effect of the heat pump.

ASHRAE/IESNA Standard 90.1-1999 mandates that heat pumps equipped with internal electrical resistance heaters shall have controls to prevent supplemental heater operation when the heating load can be met by the heat pump alone during heating or setback recovery.

System Performance

System performance of an air-source heat pump can be illustrated by plotting the heating coil load Q_{ch}, the cooling coil load Q_{cc}, the heating capacity of the heat pump Q_{hp}, the cooling capacity of the heat pump Q_{rc}, the coefficient of performance of the heat pump during heating COP_{hp}, and the energy efficiency ratio of the heat pump during cooling EER_{hp} against the outdoor temperature, as shown in Fig. 12.4. Coil loads and heat pump capacities all are expressed in Btu/h (W).

The heating capacity of an air-source heat pump using a reciprocating or a scroll compressor can be calculated as

$$Q_{\text{hp}} = 60\dot{V}_p\rho_{\text{suc}}\eta_v(q_{1'4} + q_{2'3'}) = 60\dot{V}_p\rho_{\text{suc}}\eta_v[(h_{1'} - h_4) + (h_{2'} - h_{3'})] \tag{12.5}$$

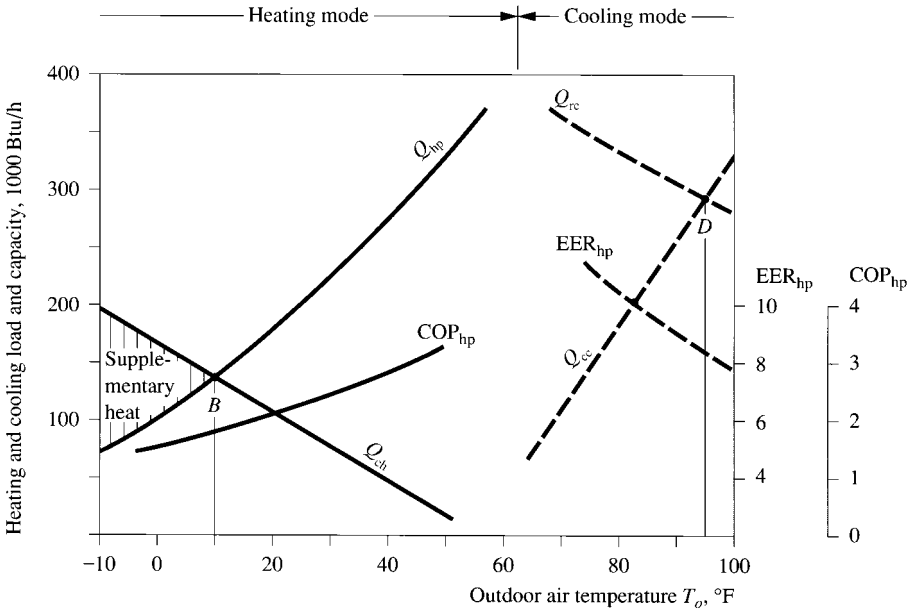


FIGURE 12.4 System performance of a rooftop heat pump.

where $\dot{V}_p$ = piston displacement of reciprocating compressor or suction volume flow rate, cfm [m/(60 s)]
$h_{1'}$, h_4 = enthalpy of refrigerant leaving and entering evaporator, Btu/lb (J/kg)
$h_{2'}$, $h_{3'}$ = enthalpy of hot gas discharged from compressor and enthalpy of refrigerant leaving condenser, Btu/lb (J/kg)
ρ_{suc} = density of suction vapor, lb/ft^3 (kg/m^3)
η_v = volumetric efficiency

The cooling capacity of an air-source heat pump Q_{rc} can be calculated from Eq. (9.25) as described in Chap. 9. The cooling coil load Q_{cc} should be offset by the cooling capacity Q_{rc} of the heat pump, and the heating coil load Q_{ch} should be offset by the heating capacity Q_{hp} provided by the heat pump. All Q_{cc}, Q_{rc}, Q_{ch}, and Q_{hp} are expressed in Btu/h (W).

When an air-source heat pump is operated in cooling mode during summer, the condensing temperature T_{con} and condensing pressure p_{con} drop as the outdoor temperature T_o falls. These decreases result in higher η_v and therefore an increase in the cooling capacity of the heat pump Q_{cc} as well as the energy efficiency ratio of the heat pump during cooling EER$_{hp}$. The fall of T_o also causes a decrease in the space cooling load and the cooling coil load Q_{cc}. The intersection of the Q_{rc} and Q_{cc} curves, point D, indicates the maximum cooling coil load that can be offset by the cooling capacity of the selected heat pump.

When the heat pump is operated in heating mode, a fall of T_o causes a decrease in the evaporating temperature T_{ev}. A lower T_{ev} results in a lower η_v, a smaller refrigeration effect $q_{1'4}$, and a lower density of suction vapor ρ_{suc}. All these effects result in a smaller heating capacity of the heat pump Q_{hp}. Although the work input W_{in} increases as T_o and T_{ev} decrease, the effect of the increase in W_{in} on the increase of Q_{hp} is small.

The fall of T_o also causes a rise in space heating load Q_{rh} and the heating coil load Q_{ch}. The intersection of the heating capacity Q_{hp} curve and the heating coil load Q_{ch} curve, at point B, is the balance point at which the heating capacity of the heat pump is equal to the required heating coil

load. When the outdoor temperature T_o drops below this balance point (such as $T_o < 10°F$ or $-12.2°C$), as shown in Fig. 12.4, supplementary heating from the electric heater or other heat source is required to maintain a preset discharge air temperature. The coefficient of performance of the heat pump during heating COP_{hp} also drops as T_o falls. The heat pump and supplementary heating operate simultaneously until the COP_{hp} drops below a certain value, such as below 1 when the use of an electric heater is more cost-effective than to operate the heat pump. The heat pump should be turned off. As discussed in Sec. 11.10, a scroll heat pump has a flatter capacity curve when outdoor air temperature T_o varies than a reciprocating heat pump. During heating mode operation, a flatter heating capacity Q_{hp} curve reduces the amount of supplementary heating. During cooling mode operation in hot summer, when T_o exceeds the design value, the increase of the space temperature will be less for a flatter capacity curve Q_{rc} than for a steeper one.

Cycling Loss and Degradation Factor

For split packaged air-source heat pumps, indoor coils are located inside the building and outdoor coils are mounted outdoors. When an on/off control is used for the compressor, during the off period, refrigerant tends to migrate from the warmer outdoor coil to the cooler indoor coil in summer and from the warmer indoor coil to the cooler outdoor coil during winter. When the compressor starts again, the transient state performance shows that a 2- to 5-min operating period of reduced capacity is required before the heat pump can operate at full capacity. Such a loss due to cycling of the compressor is called cycling loss.

According to the Department of Energy (DOE) test procedure (1986) and O'Neal et al. (1991), cycling losses are illustrated by the following parameters: part-load factor (PLF), cooling load factor (CLF), and degradation coefficient C_d. PLF can be calculated as

$$\text{PLF} = \frac{\text{EER}_{\text{cyc}}}{\text{EER}_{\text{ss}}} \tag{12.6}$$

where EER_{cyc} = energy-efficiency ratio (EER) of air-source heat pump during a whole cycle, Btu/h · W(COP)
EER_{ss} = EER of air-source heat pump if it is operated at steady-state, i.e., when compressors and fans are operated continuously, Btu/h · W(COP)

The cooling load factor can be calculated as

$$\text{CLF} = \frac{H_{\text{cyc}}}{Q_{\text{ss}} t} \tag{12.7}$$

where H_{cyc} = total energy (cooling or heating) delivered during a cycle, Btu
Q_{ss} = steady state cooling or heating rate, Btu/h (W)
t = time of a cycle, h

The degradation coefficient C_d can be calculated as

$$C_d = \frac{1 - \text{PLF}}{1 - \text{CLF}} \tag{12.8}$$

Cycling losses depend on (1) the cycling rate (whether it is 2, 3, 4, or 5 cycles per hour), (2) the indoor-outdoor temperature difference, and (3) the fraction of on-time per cycle. At design conditions, theoretically, PLF = 1. In Baxter and Moyers (1985), field tests of a typical heat pump in an unoccupied single-family house in Knoxville, Tennessee, between 1981 and 1983 showed that the degradation coefficient C_d in the heating season is 0.26, and C_d in the cooling season is 0.11.

Minimum Performance

To encourage the use of energy-efficient air conditioners and heat pumps, *ASHRAE/IESNA* Standard 90.1-1999 mandates the minimum efficiency requirements for air-source heat pumps at various cooling capacities during cooling and heating mode as follows:

Size	Cooling description	Minimum efficiency	Efficency as of 10/29/2001	Heating rating condition	Minimum efficiency	10/29/2001
< 65,000 Btu/h	Split system Single package	10.0 SEER 9.7 SEER	10.0 SEER 9.7 SEER	Split system Single package	6.8 HSPF 6.6 HSPF	6.8 HSPF 6.6 HSPF
≥ 65,000 Btu/h and < 135,000 Btu/h	Split system and single package	8.9 EER	10.1 EER	47°Fdb/43°Fwb outdoor air	3.0 COP	3.2 COP
≥ 135,000 Btu/h and < 240,000 Btu/h	Split system and single package	8.5 EER	9.3 EER	47°Fdb/43°Fwb outdoor air (≥ 135,000 Btu/h)	2.9 COP	3.1 COP
≥ 240,000 Btu/h	Split system and single package	8.5 EER 7.52 IPLV	9.0 EER 9.22 IPLV			

Single-phase packaged units < 65,000 Btu/h (19 kW) are regulated by U.S. National Appliance Energy Conservation Act (NAECA) of 1987, whose HSPF rating includes all usage of internal electric resistance heating and meets the requirements. For details, refer to ASHRAE/IESNA Standard 90.1-1999. ILPVs and part-load rating conditions are only applicable to equipment with capacity modulation. Deduct 0.2 from the required EERs and IPLVs for units with heating devices other than electric resistance heating.

Defrosting

Most air-source heat pumps use the reverse cycle to melt the frost that formed on the outdoor coil during heating mode operation in cold weather. The reverse cycle defrost switches the heating mode operation, in which the outdoor coil acts as an evaporator, to cooling mode operation, where the outdoor coil acts as a condenser. Hot gas is forced into the outdoor coil to melt the frost that accumulated there. After the frost is melted, the heat pump is switched back to normal heating mode operation.

O'Neal and Peterson (1990) described the defrosting process of an air-source heat pump using HCFC-22 as refrigerant with a short capillary tube of 0.059-in. (1.5-mm) diameter as the throttling device. When the reversing valve was energized, the outdoor fan stopped and the defrosting cycle began. The suction and discharge pressures equalized. The sudden decrease in pressure in the indoor coil caused the liquid refrigerant to vaporize. The compressor became temporarily starved and pulled the pressure in the indoor coil down to 23 psia (159 kPa abs.) about 1 min after defrosting started. Once the suction pressure had fallen low enough, refrigerant flow began to increase. In the interval from 1.0 to 3.5 min after defrosting, the refrigerant changed from vapor to saturated liquid upstream of the short capillary tube. This change caused a substantial increase in refrigerant flow, and frost was melted at the outdoor coil. After 6 min, the refrigerant flow fell 30 percent because of the subcooling in the outdoor coil. Defrosting usually lasts a few minutes up to about 10 min, depending on the frosting accumulation.

During defrost, a cold supply air from the indoor coil may cause a low space temperature and a draft. Supplementary electric heating should be considered to maintain an acceptable indoor temperature. ASHRAE Standard 90.1-1999 permits supplementary heating to be used during outdoor coil defrost cycles.

Defrosting only takes place on the outdoor coil. The initiation and control of defrosting can be performed by a clock or an intelligent or adaptive timer. It can also be controlled by measuring the

capacity of the unit or by measuring the temperature differential between the refrigerant inside the outdoor coil and the ambient air. Defrosting terminates when the temperature of liquid refrigerant leaving the outdoor coil (or the coil temperature) rises above 60°F (15.6°C).

Having or not having a suction line accumulator also affects the performance of the heat pump. Nutter et al. (1996) showed that the refrigerant flow rate averaged 18 percent higher and had a 7 percent shorter defrost cycle for heat pumps without an accumulator than those with an accumulator.

Controls

For reciprocating heat pumps, on/off, speed modulation, and cylinder unloading (as described in Sec. 11.5) capacity controls are generally used. For scroll heat pumps, on/off, variable-speed, and variable-displacement modulation capacity control are usually used.

Either the discharge air temperature or the return temperature can be used as the criterion to change automatically from cooling mode to heating mode and vice versa. A dead band of 2 to 3°F (1.1 to 1.7°C) and a time delay are always required between cooling and heating mode operations to prevent short cycling.

Most of the packaged heat pumps provide specific safety controls of high pressure, low pressure, head pressure, or low ambient control; freezing protection of indoor coil; protection from overloading; and supplementary heating. The principle and operation of these controls are the same as described in Sec. 11.5. A microprocessor-based DDC system controller may be used to integrate all the controls in one package and to add time delay, compressor lockout, loss of refrigerant charge, and short-cycling protection controls to the sequence control of heat pump and gas furnace in heating mode operation, and air economizer and refrigeration capacity control in cooling mode operation.

Capacity and Selection

Air-source heat pump capacity is selected according to its cooling capacity because supplementary heating may be required under winter design conditions. Also, the rated cooling capacity at summer design conditions is often greater than the rated heating capacity at winter design conditions.

When an air-source heat pump is installed directly inside or above the conditioned space, the noise generated by the heat pump must be taken into consideration. Attenuation remedies should be provided if necessary to maintain an NC curve at an acceptable level in the conditioned space. Sound control is discussed in Chap. 19. In 1992, air-source heat pump products were available ranging in cooling capacity from a fraction of a ton to about 40 tons (few kilowatts to 140 kW) with an indoor airflow of 16,000 cfm (7550 L/s).

12.3 GROUNDWATER HEAT PUMP SYSTEMS

Groundwater heat pump (GWHP) systems use well water as a heat source during heating and as a heat sink during cooling. When the groundwater is more than 30 ft (9 m) deep, its year-round temperature is fairy constant. Groundwater heat pump systems are usually open-loop systems. They are mainly used in low-rise residences in northern climates such as New York or North Dakota. Sometimes they are used for low-rise commercial buildings where groundwater is readily available and local codes permit such usage.

Groundwater Systems

For commercial buildings, the design engineer must perform a survey and study the site and surroundings to define the available groundwater sources. The design engineer should be fully aware of the legalities of water rights.

A test well should be drilled to ensure the availability of groundwater. If water is corrosive, a plate-and-frame heat exchanger may be installed to separate the groundwater and the water entering the water coil in the water-source heat pump (WSHP).

Usually, two wells are drilled. One is the supply well, from which groundwater is extracted by submersible pump impellers and supplied to the WSHPs. The other well is a recharge or injection well. Groundwater discharged from the WSHP is recharged to this well. The recharged well should be at least 100 ft (30 m) away from the supply well. Using a recharge well provides for resupply to the groundwater and prevents the collapse of the building foundation near the supply well due to subsidence. If the quality of groundwater meets the requirement and if local codes permit, groundwater discharged from the WSHP can be used as the service water or can be drained to the nearby river, lake, or canal. The groundwater intake water screen of the supply well may be located in several levels where water can be extracted. For a small supply well for residences, the pump motor is directly connected to the submersible pump underneath the impellers, whereas for large wells, the motor is usually located at the top of the supply well. Information regarding groundwater use regulations and guidelines for well separation and for the construction of supply and recharge wells are included in Donald's (1985) *Water Source Heat Pump Handbook*, published by the National Water Well Association (NWWA).

If the temperature of the groundwater is below 50°F (10°C), direct cooling of the air in the WSHP should be considered. If the groundwater temperature exceeds 55°F (12.8°C) and is lower than 70°F (21.1°C), precooling of recirculating air or makeup air may be economical.

Because the groundwater system is an open-loop system, it is important to minimize the vertical head to save pump power. Air-conditioning and Refrigeration Institute (ARI) Standard 325-85 recommends that the groundwater pump power not exceed 60 W/gpm (950 W·s/L). If the pump efficiency is 0.7, the allowable head for the well pump, including static head, head loss across the water coil or heat exchanger, valves, and piping work losses, should preferably not exceed 220 ft WC (66 m WC). If a recharge well is used and the discharge pipe is submerged under the water table level in the recharge well, as shown in Fig. 12.6, the groundwater system is most probably a closed-loop system, depending on whether the underground water passage between the supply and recharge well is connected or broken.

Groundwater Heat Pump System for a Hospital

A groundwater heat pump system described in Knipe (1983) was a 1980 retrofit project for a hospital in Albany, New York (see Fig. 12.5). Altogether, 540 gpm (34 L/s) of groundwater was supplied from a 12-in.- (300-mm-) diameter, 500-ft- (150-m-) deep well. During summer, the groundwater temperature was 58°F (14.4°C), and in winter it dropped to 46°F (7.8°C). About one-third of the groundwater was supplied to a makeup air-handling unit. The other two-thirds was sent to a heat pump which had the following operating characteristics:

Summer:

- Well → condenser → discharged to nearby river
- Evaporator → chilled water to AHU and terminals → evaporator

Winter:

- Well → evaporator → chilled water to AHU and terminals → discharged to nearby river
- Condenser → domestic hot water preheat → perimeter heating → condenser

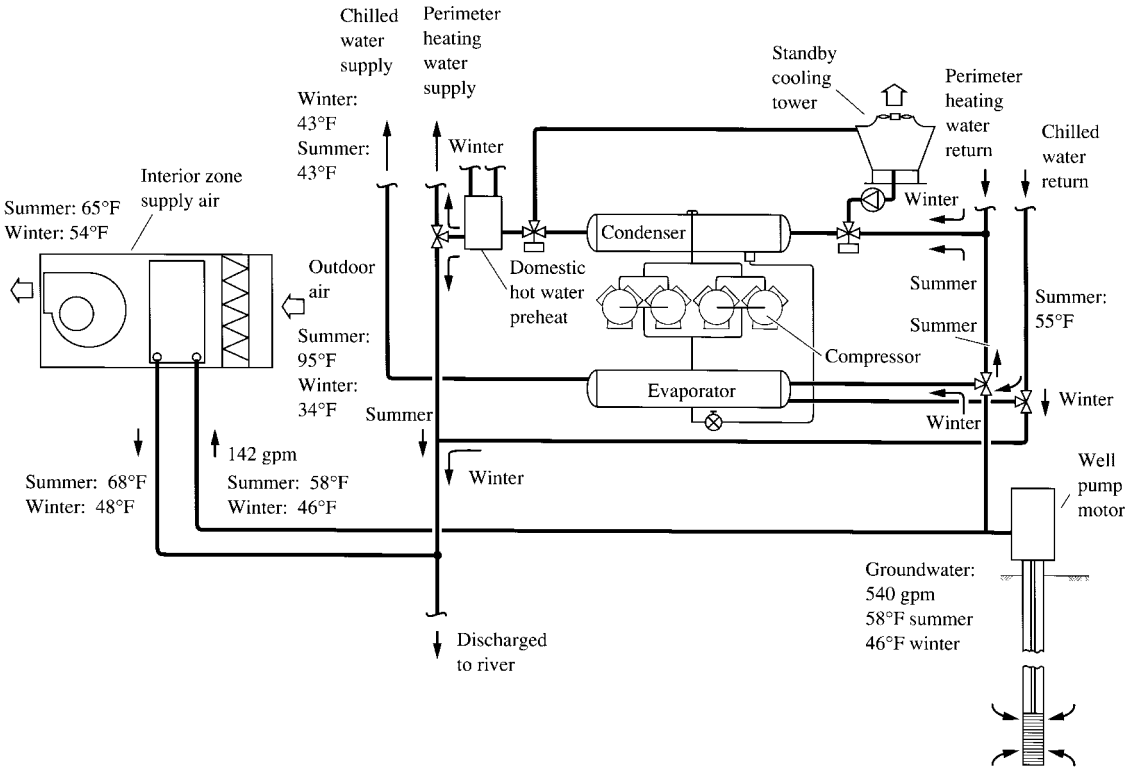


FIGURE 12.5 A typical groundwater heat pump system for a hospital.

In summer, groundwater entered the condenser at 58°F (14.4°C) and left at 68°F (20°C). In winter, groundwater entered the evaporator at 46°F (7.8°C), and left at 43°F (6.1°C).

The original installation had two 200-ton (700-kW) absorption chillers. Winter heating was supplied from gas-fired boilers. After retrofit, this groundwater heat pump system, including precooling and preheating, saved 30 percent of the energy used compared to the previous year's expense.

Groundwater Heat Pump Systems for Residences

A typical groundwater heat pump system for a residence is shown in Fig. 12.6. Such a heat pump system usually has a rated heating capacity from 24,000 to 60,000 Btu/h (7030 to 17,580 W). Groundwater is extracted from a supply well by means of a submersible well pump and is forced through a precooler or direct cooler. Then groundwater enters the water coil in the water-source heat pump. After that, groundwater is discharged to a recharge well. If the recharge pipe is submerged underneath the water table, as described in the previous section, such a groundwater system is most probably a closed-loop system. In the recharge well, the water level is raised in order to overcome the head loss required to force the groundwater discharging from the perforated pipe wall through the water passage underground. The vertical head required is the difference between the water levels in the supply and recharge wells, as shown in Fig. 12.6.

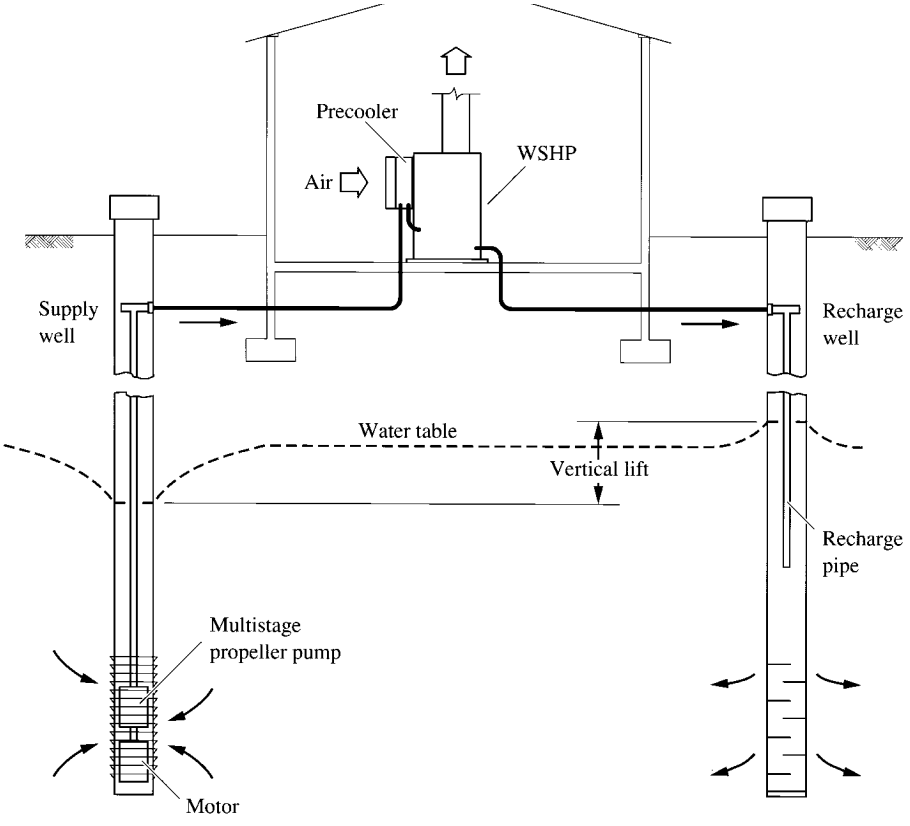


FIGURE 12.6 A typical residential groundwater heat pump system.

The water-source heat pump for residential GWHP systems has a structure similar to the WSHP in the water-source heat pump system (which is discussed in Chap. 29) except that a precooler or a direct cooler may be added.

Operating parameters and characteristics for a groundwater heat pump system are as follows:

- The groundwater flow rate for a water-source heat pump should vary from 2 to 3 gpm per 12,000 Btu/h (0.036 to 0.054 L/s per kW) heating capacity. The well pump must be properly sized. A greater flow rate and an oversized well pump are not economical.
- The pressure drop of the groundwater system should be minimized. The pressure drop per 100 ft (30 m) of pipe should be less than 5 ft/100 ft (5 m/100 m) of length. Unnecessary valves should not be installed. Gate valves or ball valves should be used instead of globe valves to reduce pressure loss. A water tank is not necessary.
- Direct cooling or precooling of recirculating air by means of groundwater increases significantly the EER of the GWHP system.
- Water containing excessive concentrations of minerals causes deposits on heat pump water coils that reduce the heat pump performance.
- In locations where the number of annual heating degree-days HDD65 exceeds 7000, more than 80 percent of the operating hours of the GWHP systems are for space heating.

- An electric heater may be used for supplementary heating in cold climates or in other locations where it is necessary. When the heating capacity of GWHP is equal to or even greater than the heating load operation of the electric heater must be avoided.
- Water-source heat pumps should be properly sized. They should not be operated for short-cycle durations (cycles less than 5 min) in order to prevent cycling losses and excessive wear and tear on the refrigeration system components. Cycle durations between 10 and 30 min are considered appropriate.
- Extraction and discharge of groundwater must comply with local codes and regulations.
- The temperature of groundwater tends to increase with its use.

A parameter called the seasonal performance factor (SPF) is often used to assess the performance of a groundwater heat pump system. SPF is defined as

$$\text{SPF} = \frac{Q_{\text{sup}}}{Q_{\text{cons}}} \tag{12.9}$$

where Q_{sup} = all heat supplied by GWHP during heating season, Btu (kJ)
Q_{cons} = energy consumed during heating season, Btu (kJ)

For groundwater heat systems with a cooling capacity $Q_{rc} < 135{,}000$ Btu/h (40 kW), ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency requirements of 11.0 EER for water-source heat pumps using groundwater during cooling mode when the entering water is 70°F (21.1°C). As of October 29, 2001, the minimum efficiency requirement is 16.2 EER when the entering water is 59°C (15°C). During heating mode, the minimum efficiency requirement is 3.4 COP when the entering water temperature is 70°F (21.1°C). As of October 29, 2001, the minimum efficiency requirement is 3.6 COP when the entering water temperature is 50°F (10°C).

A groundwater heat pump system has a fairly constant COP even if the outdoor air temperature varies. According to Rackliffe and Schabel (1986), the SPF and EER for 15 single-family houses in New York State from 1982 to 1984 were as follows:

System SPF (heating)	1.9 to 3	average 2.3
System average EER (cooling)	5.6 to 14	average 9.2

In many locations, groundwater heat pump systems usually have a higher SPF and seasonal EER than air-source heat pumps. GWHP system capacity remains fairly constant at very low and very high outdoor temperatures.

The main disadvantage of a groundwater heat pump system is its higher initial cost. More maintenance is required for systems using water with high mineral content. If the water table is 200 ft (60 m) or more below ground level, the residential groundwater heat pump system is no longer energy-efficient compared to high-efficiency air-source heat pumps.

12.4 GROUND-COUPLED AND SURFACE WATER HEAT PUMP SYSTEMS

Ground-coupled heat pump systems can be categorized as ground coil heat pump systems and direct-expansion ground-coupled heat pump systems. Of the ground coil heat pump systems, both horizontal and vertical coils are used. Many types of direct-expansion ground-coupled heat pump systems are still being developed. In fact, horizontal ground coil heat pump systems are the most widely used ground-coupled heat pump systems. A horizontal ground coil heat pump system is shown in Fig. 12.7. It is actually a closed-loop water-source heat pump system. The water-source heat pump has a heating capacity between 24,000 and 60,000 Btu/h (7030 and 17,580 W).

A horizontal ground coil is often made of polyethylene or polybutylene tubes of 1- to 2-in. (25- to 50-mm) external diameter in serpentine arrangement. In Ball et al. (1983) the horizontal coil is

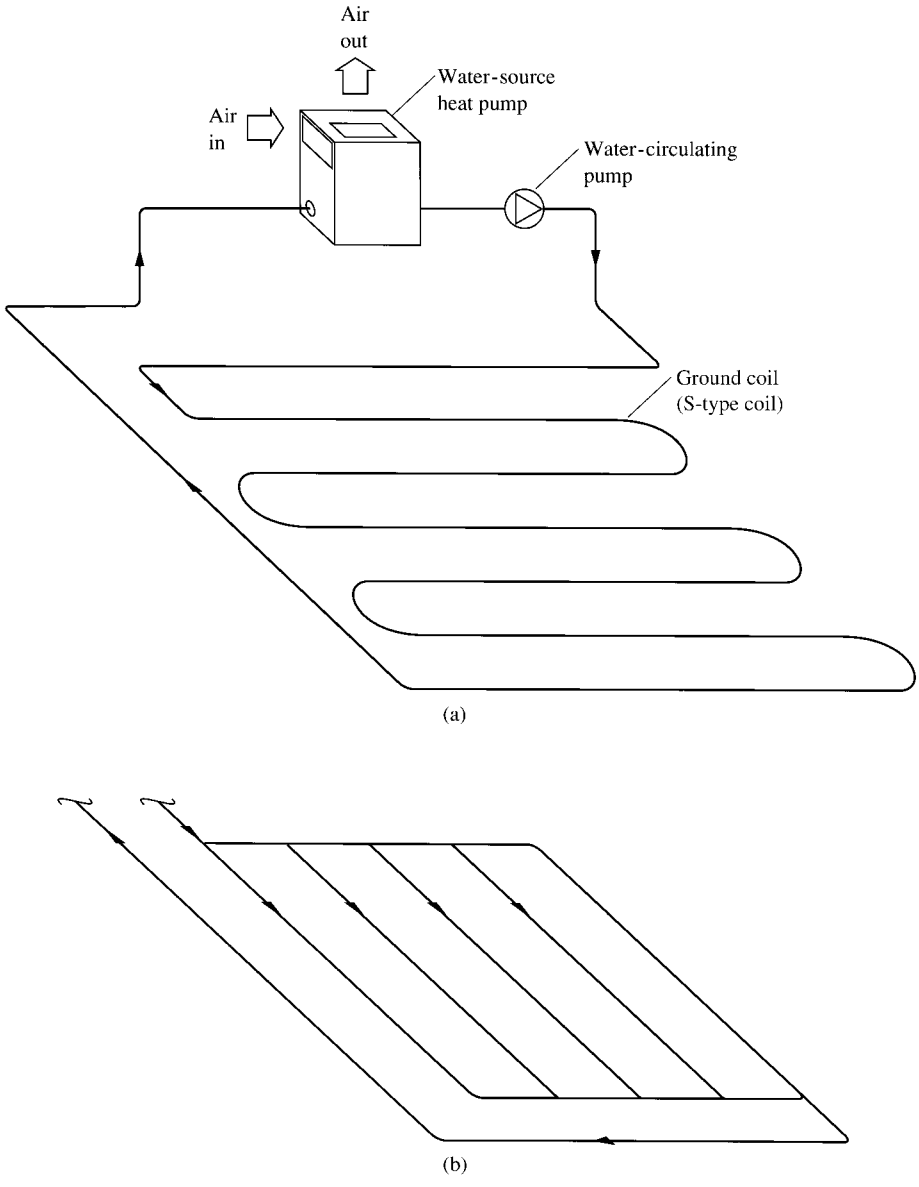


FIGURE 12.7 Ground-coupled heat pump system: (*a*) Schematic diagram; (*b*) grid-type ground coil.

buried in two layers in a trench, typically at 4 and 6 ft (1.2 and 1.8 m) deep. Spacing between the tubes varies from 2 to 8 ft (0.6 to 2.4 m). A general guideline is to use 215 to 430 ft (65 to 130 m) of copper/steel tubing for each ton (3.5 kW) of capacity for cooling. When a horizontal ground coil heat pump used in cooling mode was subjected to a heat rejection temperature that exceeded 100°F (37.8°C), some cracking of polybutylene pipe occurred.

A vertical ground coil is buried from 30 to 200 ft (6 to 60 m) deep in drilled holes. About 200 to 250 ft (60 to 75 m) of pipe is needed for each 12,000 Btu/h (3.5 kW) of heating capacity. A vertical ground coil requires less land area than a horizontal ground coil.

If the temperature of the fluid circulated in the water coil and the horizontal ground coil could drop below 32°F (0°C), then aqueous solutions of ethylene, propylene glycol, or calcium chloride should be used to protect the system from freezing. A water flow rate between 2 and 3.5 gpm per 12,000 Btu/h (0.036 and 0.063 L/s per kW) heating capacity is usually used.

In northern climates, horizontal ground coil heat pump systems have a heating SPF of 2.5 to 3 and a cooling seasonal EER of 10.5 to 13.5. There is no significant difference in SPF between horizontal and vertical ground coils. According to Hughes et al. (1985), for newly constructed projects in upstate New York, a ground coil heat pump system has a simple payback period of 5 to 10 years.

Based on ASHRAE RP-863 data, Cane et al. (1996) summarized and analyzed 12 ground-coupled heat pump systems. The average installed heat pump capacity per 1000 ft^2 (93 m^2) of gross floor area is 2.4 tons/1000 ft^2 (0.091 kW/m^2), the average length of the ground coil pipe per ton of cooling capacity is 287 ft (88 m), the average electric energy use was 14.7 $kWh/ft^2 \cdot$ year, and the capital cost for the HVAC&R system was \$6.6/ft^2.

For water-source heat pumps that use surface water such as lake water as a heat source and sink, a plastic or copper coil, or sometimes a plate-and-frame heat exchanger, is often used to form a closed-loop system to prevent fouling of the water coil.

12.5 AIR-TO-AIR HEAT RECOVERY

Types of Air-to-Air Heat Recovery

In HVAC&R systems, it is always beneficial if the cooling effect of the exhaust air can be used to cool and dehumidify the incoming outdoor air during summer, and if the heating effect can be used to heat the cold outdoor air during winter. Air-to-air heat recovery is used to recover the cooling and heating energy from the exhaust air for the sake of saving energy. In an air-to-air heat recovery system, the exhaust air is the airstream used to provide cooling capacity or heating energy. The other airstream used to extract heat energy from or release heat energy to the exhaust airstream is the outdoor airstream, which has a greater temperature and enthalpy difference between these airstreams than the supply airstream (which is often a mixture of outdoor and recirculating airstreams).

ASHRAE/IESNA Standard 90.1-1999 specifies individual air systems that have both a design air supply capacity of 5000 cfm (2360 L/s) or greater and have a minimum outside air supply of 70 percent or greater of the design supply volume flow rate shall have an energy (heat) recovery system with at least 50 percent recovery effectiveness. This means that the change of the enthalpy of the outdoor air supply is equal to 50 percent of the difference between the outdoor and exhaust air during design conditions.

Exceptions to having an energy recovery system include the contamination of the intake outdoor airstream or the energy recovery system is not cost effective.

Currently used air-to-air heat recovery devices are mainly divided into the following categories: fixed-plate heat exchangers, runaround coil loops, rotary wheel, and heat pipes.

Effectiveness

According to ASHRAE Standard 84-1991, *Method of Testing Air-to-Air Heat Exchangers*, the performance of the sensible energy transfer (dry-bulb temperature), latent energy transfer (humidity ratio), and total energy transfer of an air-to-air heat recovery device is assessed by its effectiveness

ε, which is defined as

$$\varepsilon = \frac{\text{actual transfer (moisture or energy)}}{\text{maximum possible transfer between airstreams}} \tag{12.10}$$

The effectiveness of sensible heat recovered ε_T can be calculated as

$$\begin{aligned} \varepsilon_T &= \frac{C_h(T_{he} - T_{hl})}{C_{\min}(T_{he} - T_{ce})} \\ &= \frac{C_c(T_{cl} - T_{ce})}{C_{\min}(T_{he} - T_{ce})} \end{aligned} \tag{12.11}$$

and $$C_h = \dot{m}_h c_{pa,h}, \qquad C_c = \dot{m}_c c_{pa,c}, \qquad C_{\min} = \dot{m}_{\min} c_{pa,\min} \tag{12.12}$$

where T_{he}, T_{hl} = air temperature of warm airstream entering and leaving heat exchanger, °F (°C)
T_{ce}, T_{cl} = air temperature of cold airstream entering and leaving heat exchanger, °F (°C)
C_h, C_c = heat capacity rate of warm and cold airstream, Btu/h · °F (W/°C)
$C_{\min}$ = heat capacity rate of airstream of smaller mass flow rate among these two airstreams, Btu/h · °F (W/°C)
$\dot{m}_h \dot{m}_c, \dot{m}_{\min}$ = mass flow rate of warm and cold airstreams and of airstream of smaller mass flow rate, lb/h (kg/s)
$c_{pa,\,h}$, $c_{pa,\,c}$, $c_{pa,\,\min}$ = specific heat of warm and cold airstreams and of airstream of smaller mass flow rate, Btu/lb · °F (J/kg · °C)

During summer cooling, the outdoor air intake in an AHU or packaged unit (PU) is the warm airstream in an air-to-air heat recovery heat exchanger and the cold airstream in the exhaust airstream. The exhaust airstreams is usually the smaller mass flow rate of these two airstreams because of the space positive pressure and additional space exhaust systems. During winter heating, the exhaust airstream is the warm airstream and is also the airstream of smaller mass flow rate. The outdoor airstream is the cold airstream.

The effectiveness of total heat recovered in an air-to-air heat recovery heat exchanger ε_h can also be calculated by Eq. (12.11), except that the air temperature T, in °F (°C), should be replaced by air enthalpy h, in Btu/lb (J/kg). If effectivenesses ε_T and ε_h are known (from manufacturer's data), the latent heat recovered can then be calculated and is the difference between the total and sensible heat recovered in the air-to-air heat recovery heat exchanger.

Fixed-Plate Heat Exchangers

The schematic diagram of a fixed-plate heat exchanger is shown in Fig. 12.8*a*. Alternate layers of fixed plates are separated and sealed and form two isolated airstream passages, outdoor and exhaust air. Only sensible heat is transferred from the warm airstream to the cold airstream. A counterflow or crossflow airflow arrangement is often used.

Aluminum and plastic are the most commonly used materials for plates because of their corrosive resistance and ease of fabrication. Steel alloys are used for higher-temperature applications. They are spaced from 0.1 to 0.5 in. (2.5 to 13 mm) apart. Heat resistance through the plates is small compared to their surface resistance. Condensate drains are often equipped in fixed-plate heat exchangers. These drains are also used to drain the wastewater after a water wash. Face air velocity through the fixed plates are usually between 200 and 1000 fpm (1 and 5 m/s) with a pressure drop from 0.1 to 1 in. (25 to 250 Pa) WC. Most fixed-plate heat exchangers have a capacity of 50 to 10,000 cfm (24 to 4720 L/s). Fixed-plate heat exchangers have a sensible heat recovered effectiveness between 0.5 and 0.8.

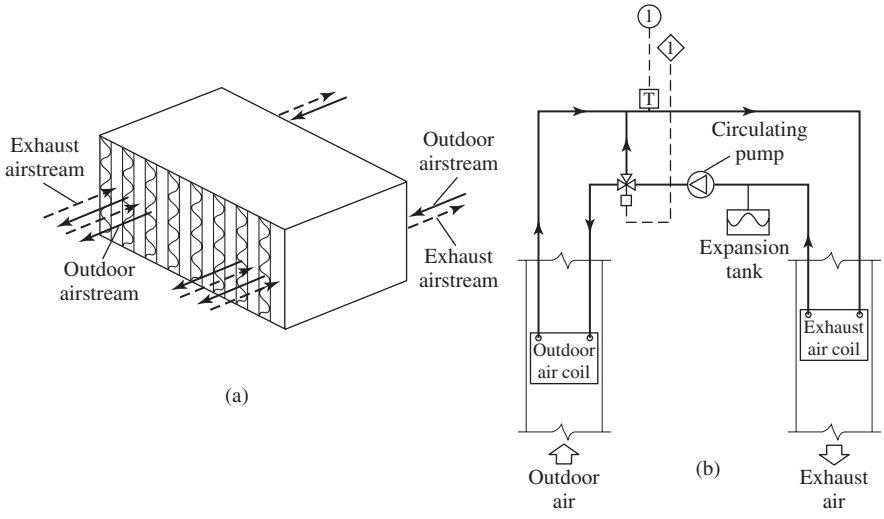


FIGURE 12.8 (*a*) A fixed-plate heat exchanger; (*b*) a runaround coil loop heat exchanger.

There is no moving part in a fixed-plate heat exchanger. Because two airstreams are in indirect contact with each other, neither airstream will contaminate the other. No cross-leakage occurs between these two airstreams.

Runaround Coil Loops

A runaround coil loop is also called a coil energy recovery loop and is shown in Fig. 12.8*b*. There are two extended surface finned-tube water coils. One is located in the outdoor air passage and the other in the exhaust air passage. These two coils are connected by water pipes and form a closed-loop system with a circulating water pump and an expansion water tank.

For locations where the outdoor air temperature is lower than 32°F (0°C), antifreeze fluid should be used to prevent freezing. A three-way control valve is often installed to prevent frosting of the exhaust air coil. This control valve will supply water to the exhaust air coil at a temperature of 30°F (−1.1°C) or above, or other preset temperatures by mixing a portion of bypassing warm water from the exhaust air coil with the cold water from the outdoor air coil during winter heating.

Runaround coil loops recover sensible heat only. They are more flexible vis-à-vis the layout of the outdoor air and exhaust air passages. The face velocity of outdoor air and exhaust air coils is usually between 300 and 600 fpm (1.5 and 3 m/s), and their pressure drops often range from 0.4 to 1.0 in. WC (100 to 250 Pa). Runaround coil loops usually have a sensible heat effectiveness between 0.45 and 0.65.

Rotary Heat Exchangers

Rotary heat exchangers, also called rotary wheels, can be classified into rotary sensible heat exchangers, in which only sensible heat is recovered, and rotary enthalpy exchangers, in which both

sensible heat and latent heat are recovered. A rotary heat exchanger is a rotary wheel with a depth of 4 to 8 in. (100 to 200 mm); Fig. 12.9*a* and *b* shows a rotary heat exchanger. The wheel has support spokes, typically 16, and its front surface is cut into segments. A rotary heat exchanger is often equally divided into two separately sealed sections: an outdoor airstream section and an exhaust airstream section. In a rotary sensible heat exchanger, it is filled with an air-penetrable medium with a large internal surface area such as aluminum, monel metal, or stainless-steel corrugated wire mesh, at a density of about 4 lb/ft^3 (64 kg/m^3). In an enthalpy exchanger, a desiccant such as lithium chloride is impregnated in the porous fiberglass matrix, or solid desiccants such as silica gels in granular form are lined with substrate sheets at a center-to-center spacing of about 30 mil. To prevent the dripping of LiCl from the fiberglass matrix, the ratio of absorbed water to the mass of desiccant should be less than 10 lb H_2O/lb LiCl (10 kg H_2O/kg LiCl). Each 1 lb (1 kg) of fiberglass matrix holds about 0.10 lb (0.10 kg) of LiCl. The rotating speed of a rotary enthalpy heat exchanger varies from 0.5 to 10 rph (revolutions per hour).

There are two airstreams in a rotary heat exchanger: an outdoor airstream and an exhaust airstream. In summer cooling, the outdoor airstream is the warm airstream and the exhaust airstream is the cold airstream, whereas in winter heating, the outdoor airstream is the cold airstream and the exhaust airstream is the warm airstream. Outdoor and exhaust airstreams flow in parallel but opposite directions for a counterflow heat transfer. The sensible heat-transfer process of a rotary sensible heat exchanger is shown in Fig. 12.9*c*, and the enthalpy transfer process of a rotary enthalpy heat exchanger is shown in Fig. 12.9*d*.

The blockage of the face area by desiccant, substrates, of support spokes for an enthalpy heat exchanger is about 28 percent. The face velocity of rotary heat exchangers varies from 500 to 700 fpm (2.5 to 3.5 m/s). At a face velocity of 500 fpm (2.5 m/s), the pressure drop of air flowing through the rotary heat exchanger varies from 0.45 to 0.65 in. WC (112 to 162 Pa), depending mainly on the structure of the heat-transfer medium.

At a face velocity of 500 to 1000 fpm (2.5 to 5 m/s), the sensible heat effectiveness of a rotary heat exchanger varies from 0.5 to 0.8, and the total heat effectiveness varies from 0.55 to 0.85. The rotating speed of a rotary sensible heat exchanger is between 10 and 25 rpm, whereas for a rotary total heat exchanger, its rotating speed varies between 0.5 and 10 rph. A higher face velocity means that a higher rate of transfer, higher pressure drop, and a larger volume flow rate result in a lower effectiveness. A lower face velocity results in a lower pressure drop, smaller volume flow rate, and a higher effectiveness.

The maximum diameter of a rotary heat exchanger is 14 ft (4.2 m). The upper limit of the volume flow rate for a single unit is about 65,000 cfm (30,670 L/s). A wheel larger than that would be too difficult to ship and install.

Heat Pipe Heat Exchangers

A heat pipe is often used as a sensible indirect heat exchanger. It consists of many heat pipes arranged in rows along the direction of airflow, as shown in Fig. 12.10*a*. Each sealed heat pipe contains a volatile fluid, as shown in Fig. 12.10*b*. When one end of the pipe, the hot end or evaporative section, absorbs heat from the airstream flowing over the pipe, then the volatile fluid inside the pipe vaporizes. The vapor then moves to the other end (the cold end or condensation section) because of high saturated pressure in the evaporative section. After condensing heat is released to another airstream that flows over the other end of the pipe, the vapor inside the pipe condenses to liquid form and is drawn back to the evaporative section by gravity. It has then completed an evaporation/condensation cycle. The heat pipes are often slightly tilted to enable the condensed liquid to flow back to the evaporation section.

Heat pipes have an inner capillary wick structure. The outer tube is often made of aluminum, with fins of the same material for a large heat-transfer surface. The volatile fluid inside the heat pipe is usually a halocarbon compound refrigerant.

When two separate airstreams flow over the heat pipe heat exchanger, the warmer airstream flows over the evaporation section and the colder airstream flows over the condensation section.

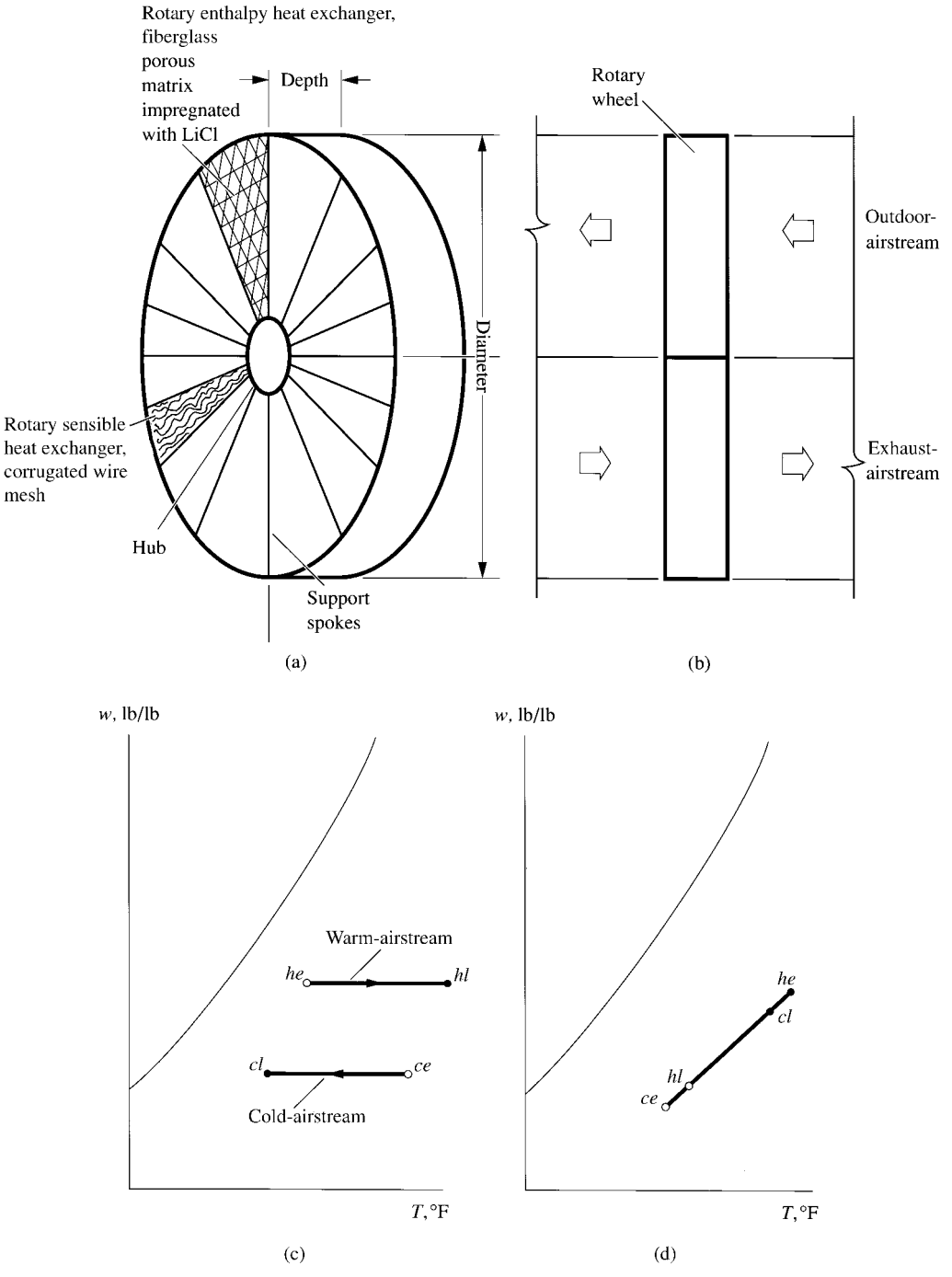


FIGURE 12.9 Rotary heat exchangers: (*a*) isometric view; (*b*) outdoor and exhaust airstreams; (*c*) sensible heat-transfer process; (*d*) enthalpy-transfer process.

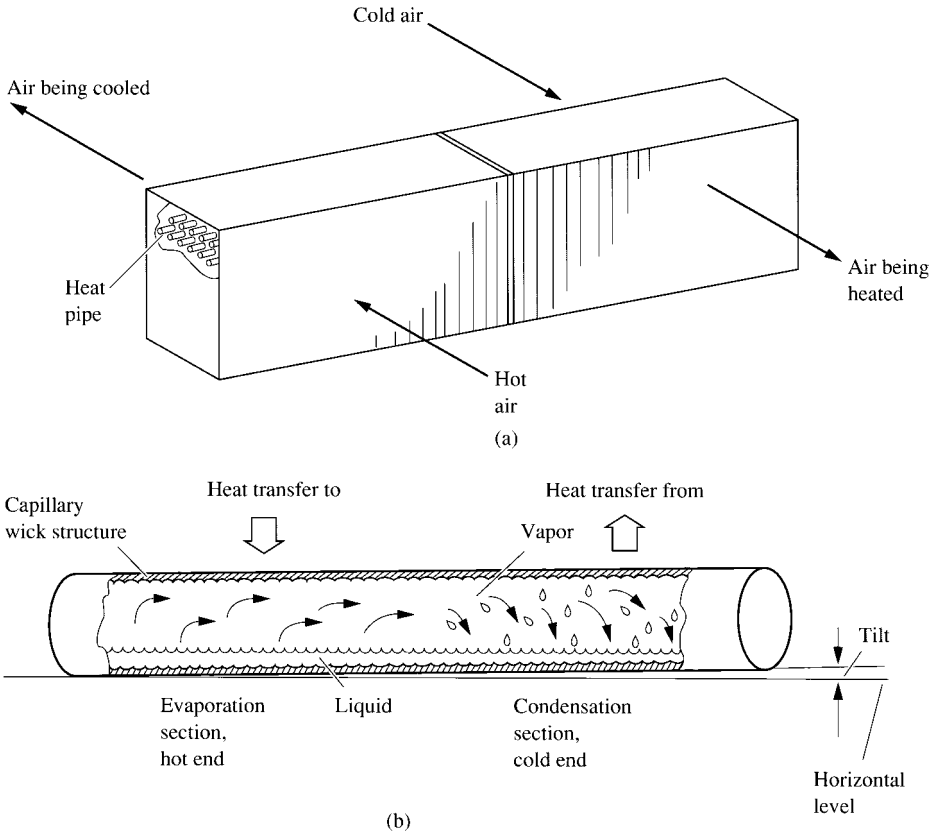


FIGURE 12.10 A heat pipe heat exchanger: (*a*) heat exchanger and airstreams; (*b*) heat pipe.

These two airstreams flow in a counterflow arrangement for greater effectiveness. The airstreams are separated by a sealed partition to prevent cross-contamination.

The performance of a heat pipe heat exchanger is indicated by its sensible heat effectiveness ε_T. Both the effectiveness and the airstream pressure drop Δp of a heat pipe heat exchanger depend mainly on its face velocity v_{face}, fin spacing, and the number of rows of heat pipes in the direction of airflow. If two heat pipe heat exchangers are connected in series, the number of rows of such a series is the total number of rows of the two heat exchangers.

The greater the total number of rows of heat pipe and fins per inch, the higher the ε_T and Δp. The lower the face velocity v_{face} of the heat pipe exchanger, the higher the ε_T and the lower the Δp. However, the volume flow rate and the capacity of the heat pipe heat exchanger will also be smaller.

The design face velocity of a heat pipe heat exchanger is between 400 and 800 fpm (2 and 4 m/s). The total number of heat pipe rows varies from 6 to 10 rows. Sensible heat effectiveness varies from 0.45 to 0.75. For a heat pipe with 14 fins/in. (1.8-mm fins) and a total of 8 rows with a face velocity v_{face} = 500 fpm (2.5m/s), its sensible heat effectiveness ε_T = 0.65 and its pressure drop Δp is about 0.6 in. WC (150 Pa). Refer to the manufacturer's data for detailed information.

The capacity and rate of heat transfer of a heat pipe can be controlled by varying the slope or tilt of the heat pipe. This adjustment increases or decreases the liquid flow inside the heat pipe and, therefore, its capacity.

Comparison between Various Air-to-Air Heat Exchangers

According to *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*, among the air-to-air heat recovery heat exchangers:

- Fixed-plate, runaround coil loop, and rotary sensible heat exchangers are all limited to sensible heat transfer and recovery; only the rotary enthalpy exchangers can recover latent heat from the exhaust airstream.
- There are no moving parts in fixed-plate and heat pipe heat exchangers.
- Both heat pipe and runaround coil loop heat exchangers show no cross-contamination or cross-leakage. Fixed-plate heat exchangers have a cross-contamination and cross-leakage from 0 to 5 percent, and rotary heat exchangers from 0 to 10 percent. If cross-contamination is strictly prohibited, a purge section can be installed to reduce it.
- The effectivenesses of fixed-plate and rotary heat exchangers are higher than those of heat pipe and runaround coil loop heat exchangers.
- Rotary heat exchangers show the lowest air pressure drop among the various types of heat exchanger.

12.6 GAS COOLING AND COGENERATION

Gas Cooling

High electricity demand charges, high peak electricity rates, and the development of high-efficiency direct-fired equipment, highly reliable gas engines, and more sophisticated and cost-effective desiccant-based air conditioning systems enable gas cooling systems to compete with electric compressors after the decline of gas cooling in the 1970s. Current gas cooling systems include the following:

- Double-effect, direct-fired, lithium bromide (LiBr) absorption chillers and chiller/heaters
- Desiccant-based air conditioning using evaporative cooling, refrigeration, and direct-fired gas heaters for regeneration
- Gas engine chiller systems and cogeneration

Usually, a gas cooling system has a higher initial cost and a lower operating cost than an electricity-driven refrigeration system. Accurate calculation of the operating costs of a gas cooling system is important in this comparison.

The LiBr absorption chillers are discussed in Chap. 14, and desiccant cooling systems are discussed in Chap. 29. Gas-engine chiller systems and cogeneration are discussed in this section.

Cogeneration

Cogeneration is the sequential use of energy from a primary source, including natural gas, oil, and coal, to produce power and heat. Power can be electric or mechanical power, or both. In a cogeneration system, the sequential use of the heat released from the flue gas and engine jacket significantly increases system efficiency and makes the cogeneration system economically attractive.

In 1978 in the United States, the Public Utility Regulatory Policies Act (PURPA) permitted the interconnection of electric power lines of cogeneration systems with electric utility systems. This provides flexibility for cogeneration plants. They can either use or sell their electric power to the

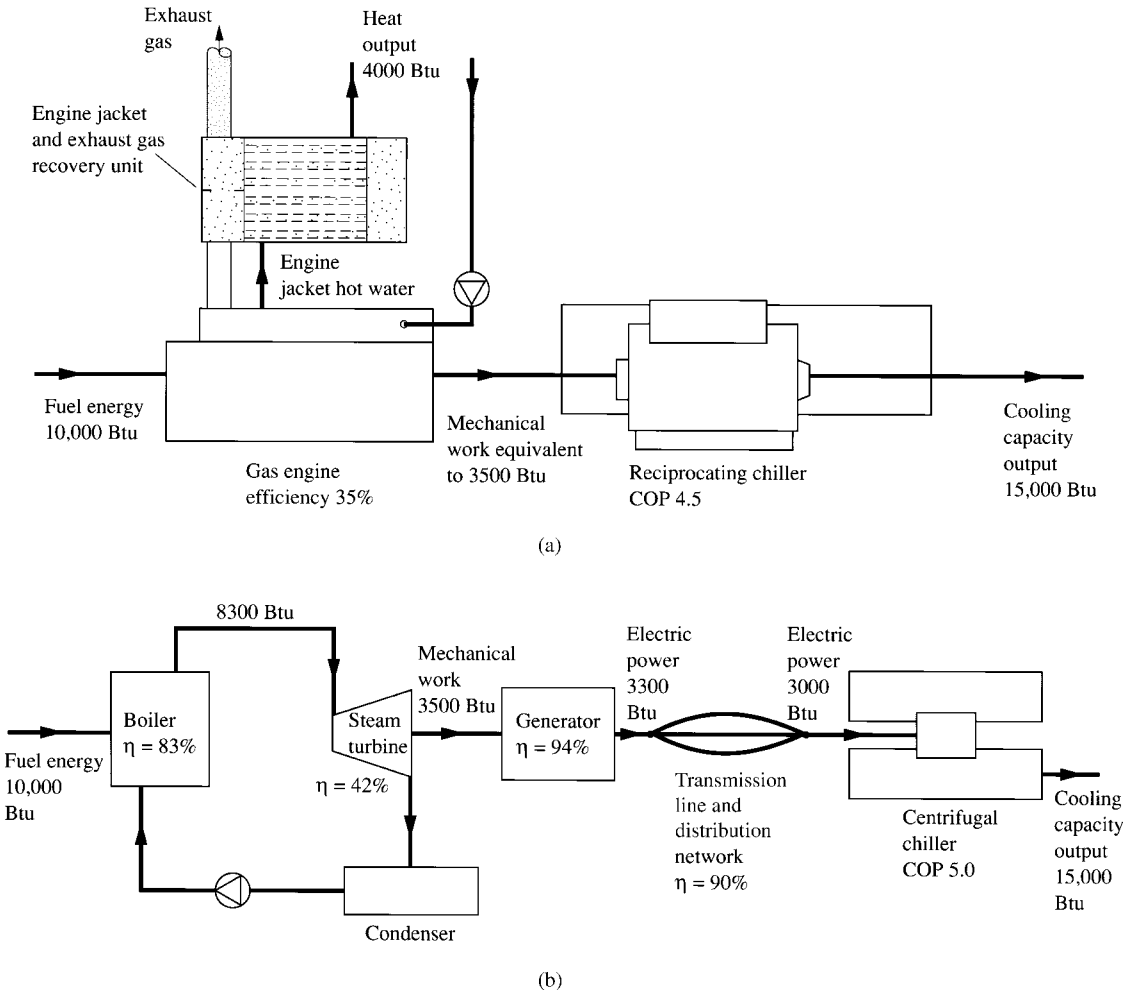

Exhaust gas
Heat output 4000 Btu
Engine jacket and exhaust gas recovery unit
Engine jacket hot water
Fuel energy 10,000 Btu
Gas engine efficiency 35%
Mechanical work equivalent to 3500 Btu
Reciprocating chiller COP 4.5
Cooling capacity output 15,000 Btu
(a)
8300 Btu
Boiler η = 83%
Fuel energy 10,000 Btu
Steam turbine η = 42%
Mechanical work 3500 Btu
Generator η = 94%
Electric power 3300 Btu
Transmission line and distribution network η = 90%
Electric power 3000 Btu
Centrifugal chiller COP 5.0
Cooling capacity output 15,000 Btu
Condenser
(b)

FIGURE 12.11 Energy flow in (*a*) gas-engine chiller and (*b*) electricity-driven chiller.

utility and optimize the size of the cogeneration plant by reducing or eliminating its standby generation capacity. During the 1990s, many states, such as California and New York, were leading the way in utility deregulation. Utility deregulation will stimulate competition in the electricity market for lower consumer rates.

Since 1990, hundreds of cogeneration systems were developed for internal use. Internal use means the production of both power and heat for use in settings such as hospitals, medical centers, university campuses, public buildings, and industrial facilities, and in their installed air conditioning systems. A successfully developed cogeneration system often relies on site technical and economic analysis, especially for local electricity demand and electricity rates. Two kinds of prime movers are widely used in these internal use cogeneration systems: gas engines and gas turbines.

Gas-Engine Chiller

A gas-engine chiller is often a combination of a gas cooling system and a cogeneration system. Heat released from the exhaust gas and the engine jacket cooling water are all recovered to increase system efficiency. Figure 12.11*a* shows the energy flow of a gas-engine chiller system.

If the efficiency of the gas engine is 35 percent, the mechanical efficiency including the transmission gear train is 95 percent, and the chiller's COP = 4.5, then for every 10,000 Btu (10,550 kJ) of fuel energy input to the gas-engine chiller, there is a cooling output Q_{rf} of

$$Q_{rf} = 10{,}000 \times 0.35 \times 0.95 \times 4.5 \approx 15{,}000 \text{ Btu } (15{,}825 \text{ kJ})$$

In addition, there is a heating output of 4000 Btu (4220 kJ) from the exhaust gas and engine jacket to supply hot water or low-pressure steam for an absoption, space heating, or domestic hot water unit.

A gas-engine chiller has a prominent advantage over a motor-driven chiller because the former can vary its speeds at various operating conditions: high speed at overloads and low speeds at part-load operation. Engine reliability is the key to user acceptance for gas-engine chillers. Several hundred engine-driven chillers were installed in the 1960s and early 1970s. According to reliability records of these systems reported in Davidson and Brattin (1986), the reliability of gas-engine chillers matches the requirements of HVAC&R systems.

Gas engines can be used to drive screw, reciprocating, or centrifugal compressors. Gas-engine-driven screw chillers are becoming more and more popular. Many manufacturers offer packaged units for easier field installation. The capacity of gas-engine chiller packaged units varies from 30 to 500 tons (105 to 1760 kW). Gas-engine cooling systems can also be coupled to direct-expansion (DX) refrigeration systems and rooftop packaged units. Gas-fired internal combustion engines for cooling in buildings follow the developments of gasoline and diesel engine technology. There are two kinds of gas engines: heavy-duty industrial applications and light-duty automotive engines. Industrial heavy-duty gas engines run a minimum of 30,000 h of full-load service between major overhauls, and cost about 5 times as much as automotive engines. The service life of an automobile engine is only 2000 to 5000 h.

Recently, manufacturers have produced packaged automotive gas-engine chillers of 150-ton (525-kW) capacity. One manufacturer also offers one 150-ton (525-kW) gas-engine chiller and integrated hot water absorption chiller to give a total maximum output of 180 tons (630 kW). Automotive gas engines are suitable for compressors that require speeds far above 1800 rpm. This packaged gas-engine chiller is also equipped with microprocessor-based controls to coordinate and monitor the operation of the engine and chiller. An operating cost as low as one-half that of similar-size electricity-driven units is claimed, depending on the local utility rate structure.

Engine-driven chillers are maintenance-intensive. The maintenance cost of a gas-engine chiller may be between 10 and 20 percent of the energy cost, and it should be added to the operating cost during economic analysis. D'zurko and Epstein (1996) made a cost analysis between gas-engine chiller and electric chillers for offices, schools, restaurants, retail stores, and hospitals in New York State. Most have a simple payback of 2.1 to 11.9 years.

For a gas-engine chiller with a cooling output of 15,000 Btu (15,825 kJ), if its by-product, 4000-Btu (4220-kJ) heat output from the exhaust gas and engine jacket is also counted, then the simple payback period is usually between 2 and 5 years.

Gas Engines

Two types of automotive engines are used to drive the chillers: gas engines operated on diesel cycle and operated on an Otto cycle. Because a diesel engine has a higher thermal efficiency than an Otto engine, more diesel engines are used in gas-engine chillers. Most diesel engines are operated on a four-stroke cycle, i.e., an intake stroke, a compression stroke, a power stroke, and an exhaust stroke, to produce power. A diesel engine can also operate on a two-stroke cycle, an intake stroke and an

exaust stroke, which is slightly cheaper than a four-stroke engine; however, a two-stroke engine is less efficient and experiences greater wear than a four-stroke engine.

Reciprocating gas engines can be categorized according to the manner in which the engine is aspirated: natural aspiration and turbocharging. A natural aspiration engine supplies air or the fuel/air mixture to the engine cylinder at atmopheric pressure, whereas a turbocharging engine supplies higher-pressure air or fuel/air mixture to the cylinder. Turbocharging uses the engine exhaust gas to drive a small turbine which is connected with a centrifugal compressor. This compressor raises the intake air pressure and quantity and thus the engine's capacity.

In a gas engine, the primary function of a heat recovery system is to reject excess heat produced and to exhaust flue gas during the power generation process. Effective heat rejection from the engine jacket and the exhaust of adequate flue gas must be emphasized. The secondary function is to recover heat from the exhaust gas and engine jacket and to use the recovered heat efficiently and economically.

Exhaust Gas Heat Recovery

According to Orlando's *Cogeneration Design Guide* (1996), the exhaust gases of the reciprocating gas engine contain about one-third of the engine's heat output, and approximately 50 to 75 percent of the sensible heat can be recovered. Heat may be recovered in the form of steam or hot water. The sensible heat recovered $Q_{ex,r}$, in Btu/h (W), can be calculated as

$$Q_{ex,r} = 60\dot{V}_{ex}\rho_{ex}c_{pa}(T_{ex} - T_{rec}) \tag{12.13}$$

where $\dot{V}_{ex}$ = volume flow rate of exhaust gas at outlet of gas engine, cfm [$m^3/(60\ s)$]
ρ_{ex} = density of exhaust gas at outlet of gas engine, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of exhaust gas, Btu/lb·°F (J/kg·°C)
T_{ex} = temperature of exhaust gas at outlet of gas engine, °F (°C)
T_{rec} = temperature of exhaust gas leaving heat recovery heat exchanger, °F (°C)

It is more economical to have a minimum temperature difference between the exhaust gas and the generated steam inside the recovery heat exchanger of 100°F (55°C).

According to *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*, the temperature of the recovered steam or hot water for absorbing chillers is between 190 and 245°F (88 and 118°C), for space heating between 120 and 250°F (49 and 120°C), and for domestic hot water between 120 and 200°F (49 and 93°C). For steam, a maximum pressure up to 8 psig (55 kPag) may be produced.

Exhaust gas heat recovery heat exchangers are often designed to reduce engine noise transmitted along with the exhaust gas and are called heat recovery mufflers. Heat recovery mufflers should minimize engine backpressure to 6 in. WG (1500 Pag) for natural aspiration reciprocating engines and to 25 to 30 in. WG (6.25 to 7.5 kPag) for turbocharged engines.

Many heat recovery mufflers are designed based on a minimum exhaust temperature of 300°F (149°C) to avoid condensation. The temperature of the exhaust gas at part-load operation is important. The construction of the exhaust heat recovery mufflers should provide access for inspection, cleaning, and soot removal for diesel engines.

Engine Jacket Heat Recovery

Approximately one-third of the heat input to the reciprocating gas engine is rejected from the engine block, heads, and exhaust manifolds to the engine jacket water coolant. Heat energy can be recovered by either the hot water with a temperature at 250°F (120°C) or low-pressure steam at a maximum pressure of 15 psig (103 kPag).

When the engine jacket water is routed through the heat recovery muffler where additional heat is extracted from the exhaust gas, an engine jacket heat recovery system is then combined with the exhaust gas heat recovery system. At least a water circulating pump is required to circulate the hot water flowing through the engine jacket and the heat recovery muffler as soon as the engine is operating. To avoid excessive thermal stress, the temperature difference between the cooling water entering and leaving the engine jacket should not exceed 15°F (8.3°C).

Steam can be produced at the top part of the heat recovery muffler with a steam separator and is distributed to various process loads. Hot water from the bottom of the heat recovery muffler is then mixed with the condensate returned from the remote steam loads. The mixture is forced through the engine jacket and the heat recovery muffler by the circulating pump.

Sometimes, the engine coolant circuit in a heat recovery loop is separated from the process loads by heat exchangers and forms a primary and secondary loop. A primary-secondary loop isolates and protects the engine coolant circuit from process loads, leaks, and failures in the distribution systems. A primary-secondary loop is especially useful for a multiple-engine installation. A cooling tower can be connected to a heat recovery loop through a heat exchanger for the sake of maintaining a required entering temperature of the engine jacket coolant at system part-load operation. A coolant heater can be installed to preheat the coolant during start-up.

For space heating, hot water from the heat recovery muffler can be supplied directly to the heaters in the conditioned space. For space cooling, an absorption chiller using recovered steam or hot water should be installed to provide cooling for the conditioned space.

Cogeneration Using a Gas Turbine

Many cogeneration plants use a combustion gas turbine instead of a gas engine as the prime mover. A gas turbine usually consists of a compressor section to raise the air pressure, a fuel/air mixing and combustion chamber, and an expanding turbine section. The compressor and turbine are joined by the same shaft. Capacity may vary from several hundred brakehorsepower (bhp) to more than 100,000 bhp (75,000 kW). Gas turbines are often connected to induction generators to produce electric power through gear trains.

Steam boilers are often used as heat recovery units to produce steam at a pressure typically 15 psig (103 kPag) from the gas-turbine exhaust gas. Recovered heat can often be used as process heat to operate an absorption chiller.

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CHAPTER 13
REFRIGERATION SYSTEMS: CENTRIFUGAL

13.1 CENTRIFUGAL VAPOR COMPRESSION REFRIGERATION SYSTEMS 13.1
- Refrigerants 13.2
- Compressor 13.3
- System Components 13.4
- Purge Unit 13.5
- Types of Centrifugal Chiller 13.7

13.2 WATER-COOLED CENTRIFUGAL CHILLER 13.7
- Refrigerant Flow 13.7
- Performance Rating Conditions 13.8

13.3 CENTRIFUGAL CHILLERS INCORPORATING HEAT RECOVERY 13.9
- System Description 13.9
- Operating Modes 13.9
- System Characteristics 13.12

13.4 AUTOMATIC BRUSH CLEANING FOR CONDENSERS 13.13
- Principles and Operation 13.13
- Effect of Brush Cleaning System 13.14

13.5 CENTRIFUGAL COMPRESSOR PERFORMANCE MAP 13.15
- Surge of Centrifugal Compressor 13.15
- Centrifugal Compressor Map at Constant Speed 13.16
- Centrifugal Compressor Map at Variable Speed 13.18

13.6 CAPACITY CONTROL OF CENTRIFUGAL CHILLERS 13.19
- Difference between Centrifugal Compressors and Centrifugal Fans 13.19
- Required System Head at Part-Load Operation 13.19
- Capacity Control Using Inlet Vanes 13.20
- Capacity Control by Variable Speed 13.20
- Comparison between Inlet Vanes and Variable Speed 13.21

13.7 CENTRIFUGAL CHILLER CONTROLS 13.22
- Functional Controls and Optimizing Controls 13.22
- Chilled Water Leaving Temperature Control 13.22
- Condenser Water Temperature Control 13.23
- Safety Controls 13.23
- Fault Detection and Diagnostics 13.24
- Operator Interface 13.24

13.8 OPERATING CHARACTERISTICS OF CENTRIFUGAL CHILLERS 13.24
- Sequence of Operation of a Typical Centrifugal Chiller 13.24
- System Balance at Full Load 13.25

13.9 PART-LOAD OPERATION OF CENTRIFUGAL CHILLER 13.25
- Part-Load Operating Characteristics 13.25
- Evaporating and Condensing Temperatures at Part-Load Operation 13.26
- Temperature Lift at Part-Load Operation for Centrifugal Chillers 13.29

13.10 FREE REFRIGERATION 13.31
- Principle of Operation 13.31
- Free Refrigeration Capacity 13.32

13.11 OPERATION OF MULTIPLE-CHILLER PLANT 13.33
- Multiple-Chiller Plant 13.33
- Parallel and Series Piping 13.33
- Chiller Staging 13.34
- Chiller Minimum Performance 13.35
- Design Considerations 13.35

REFERENCES 13.36

13.1 CENTRIFUGAL VAPOR COMPRESSION REFRIGERATION SYSTEMS

A centrifugal vapor compression refrigeration system uses a centrifugal compressor to compress the refrigerant and provides chilled water to cool the air in air-handling units and terminals. A

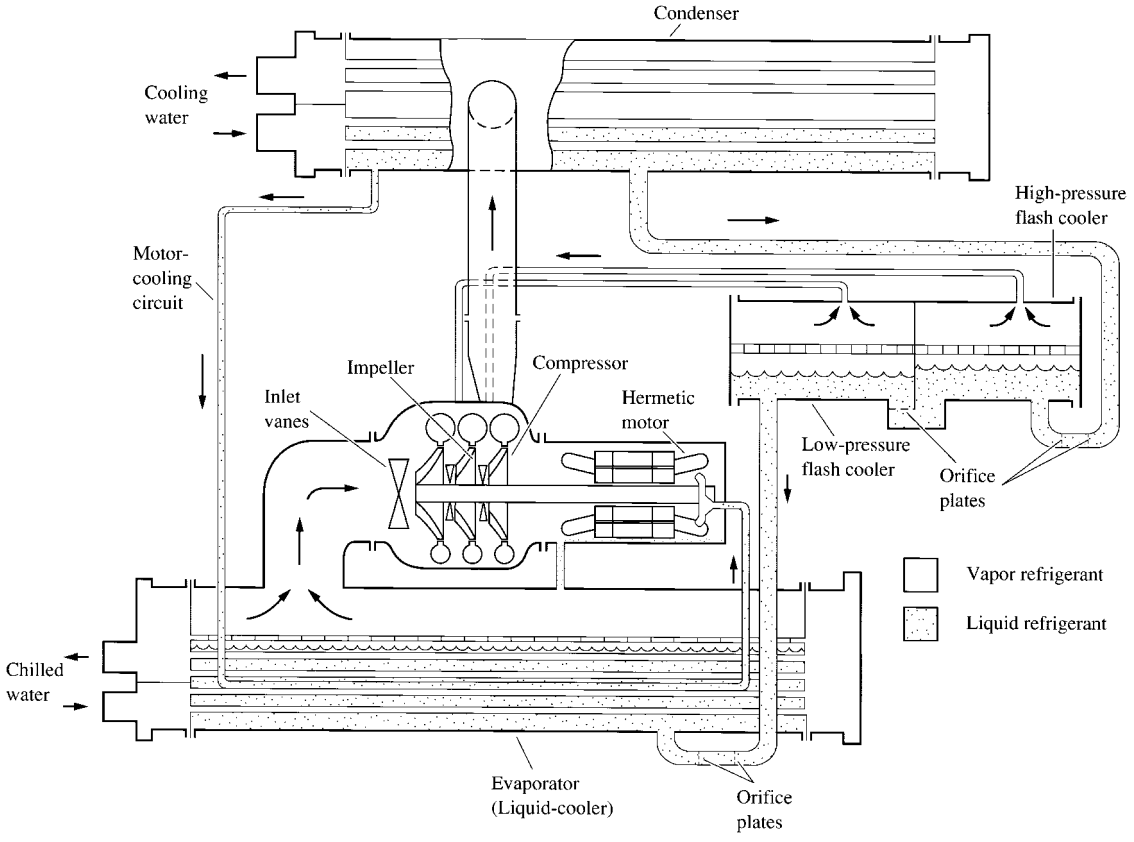


FIGURE 13.1 A typical three-stage, water-cooled centrifugal chiller.

centrifugal refrigeration machine usually comes as a factory-assembled packaged unit and is often called a centrifugal chiller, as shown in Fig. 13.1.

Most centrifugal chillers are driven by open and hermetic motors. Occasionally, they may be driven by internal combustion engines. Sometimes, large centrifugal chillers are driven by steam and gas turbines.

Refrigerants

According to the ARI survey, there were about 80,000 large-tonnage chillers in use in the United States in 1992. Of these, about 80 percent used CFC-11; the remaining used CFC-12, CFC-114, HCFC-22, and others. As described in Sec. 9.4, based on ARI's estimate (Dooley, 1997), 53 percent of the 80,000 CFC chillers will remain in service on January 1, 2000. The actual phaseout schedule is slower than expected.

As described in Secs. 9.4 and 9.5, HCFC-123, an interim refrigerant, will replace CFC-11 in centrifugal chillers. In 1997, DuPont Company (manufacturer of HCFC-123) specified an allowable exposure limit of 50 ppm for HCFC-123. HFC-134a, a long-term alternative refrigerant, will replace CFC-12. In centrifugal chillers, the direct conversion from CFC-12 to HFC-134a requires

an increase in compressor speed, a drop in cooling capacity and efficiency, and the use of synthetic polyolester lubrication oil instead of mineral oil. Ammonia R-717 is usually used in centrifugal compressors for industrial applications due to its lower flammability and higher toxicity.

Because a large amount of new non-CFC centrifugal chillers were installed to replace CFC centrifugal chillers, the energy use of the centrifugal chillers, expressed in kW/ton, is significantly reduced. The manufacturers produced new chillers with 0.49 kW/ton (7.17 COP) in 1997.

The 1990 Clean Air Act makes the knowing vent of CFCs and HCFCs illegal as of July 1, 1992. Recovery/recycle equipment must be used to prevent venting during installation, operation, and maintenance.

Compressor

A centrifugal chiller consists of a centrifugal compressor, an evaporator or liquid cooler, a condenser, a throttling device, piping connection, controls, and possibly a purge unit and a flashing cooler. A centrifugal compressor is a turbomachine. The total head lift, in psi (kPa), results from the conversion of velocity pressure to static pressure. A single-stage, two-stage, or three-stage compressor—according to the number of internally connected impellers—may be used. Although the compression ratio $R_{com} = p_{dis}/p_{suc}$ of a single-stage centrifugal compressor seldom exceeds 4, the impellers connected in series increase the compression ratio R_{com} and satisfy most compression requirements. For a refrigeration system with an evaporating temperature of 40°F (4.4°C) and a condensing temperature of 100°F (37.8°C), HCFC-123 has a compression ratio of 3.59. Use of a two-stage or even three-stage instead of a single-stage centrifugal compressor acts to increase the total head lift as well as the COP of the refrigeration system (refer to Sec. 9.12).

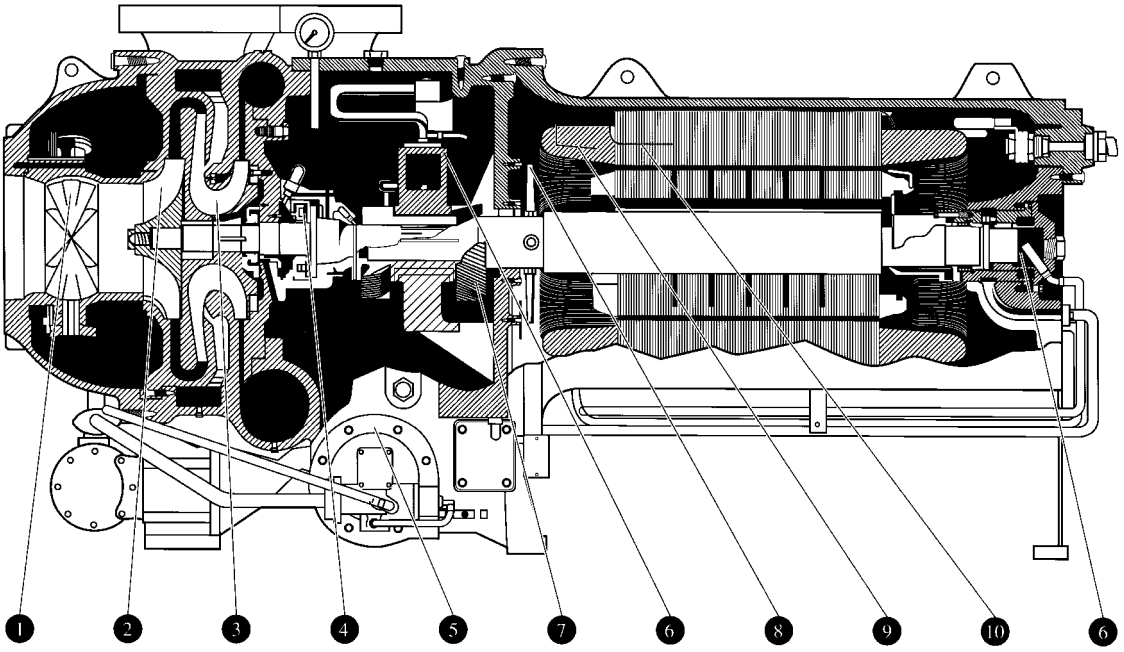
Figure 13.2 shows a two-stage centrifugal compressor. Vapor refrigerant is drawn through the inlet vanes and enters the first-stage impeller. The impeller compresses the vapor and discharges it through a diffuser, where it combines with flashed vapor refrigerant from the flash cooler. The mixture then enters the second-stage impeller, where the refrigerant is further compressed. The hot gas is discharged into the collecting volute through a diffuser. As the gaseous refrigerant flows through the compressor, the area of the flow passage increases. This causes a decrease in gas velocity and therefore a conversion of velocity pressure to static pressure.

Because a high head lift is required to raise the evaporating pressure to the condensing pressure, the hot gas discharge velocity at the exit of the second-stage impeller approaches the acoustic velocity v_{ac} of the saturated vapor. At atmospheric pressure and a temperature of 80°F (26.7°C), for HCFC-123, v_{ac} is 420 ft/s (128 m/s); and for HCFC-22, v_{ac} is about 600 ft/s (183 m/s). Centrifugal compressors need a high peripheral velocity and high rotating speed (from 1800 to 50,000 rpm). The refrigeration capacity of a centrifugal compressor ranges from 100 to 10,000 tons (350 to 35,000 kW). It is not economical to manufacture small centrifugal compressors. Because they are turbomachines, centrifugal compressors have higher volume flow than positive displacement compressors. Isentropic efficiency η_{isen} reaches a maximum of about 0.83 for a typical centrifugal compressor operated at design conditions. Its value may decrease to as low as 0.6 during part-load operation. The volumetric efficiency η_v of centrifugal compressors can be estimated at almost 1.

Inlet vanes are widely used as stepless capacity control for centrifugal compressors during part-load operation. Variable-speed drives, direct steam turbines, and engine drives also provide capacity control by varying the speed of centrifugal compressors. Sometimes, a combination of adjustable-frequency variable-speed drive and inlet vanes is used.

Many centrifugal chillers with a refrigeration capacity less than 1200 tons (4220 kW) are installed with hermetic compressors. Very large centrifugal chillers often employ open compressors in which an additional motor cooling system is used.

Centrifugal compressors need high peripheral velocity to produce head lift. They can be driven either directly by motor or through a gear train. A direct-drive compressor often has a larger-diameter impeller because of the limitation on the rotating speed of a two-pole motor. On the other hand, direct drive reduces energy loss because of the gear train driving mechanism. Such a loss is usually 2 to 3 percent of the total power input to the compressor.



1 – Variable inlet guide vanes
2 – First-stage impeller
3 – Second-stage impeller
4 – Compressor thrust bearing
5 – Lubrication package
6 – Shutdown oil reservoir
7 – Gear transmission
8 – Centrifugal demister system
9 – Internal motor protector
10 – Internal motor protector

FIGURE 13.2 A two-stage centrifugal compressor. (*Source: Carrier Corporation. Reprinted with permission.*)

Because centrifugal compressors are reliable and energy-efficient for water-cooled chillers, they are the most widely used refrigeration compressors in large central air conditioning systems.

System Components

Evaporator. A flooded shell-and-tube evaporator, as described in Sec. 10.3, is usually used for a centrifugal chiller because of its compact size and high heat-transfer characteristics.

Condenser. Water-cooled, horizontal shell-and-tube type of condensers, as described in Sec. 10.6, are widely used because of their lower condensing pressure and easier capacity control. Only in locations where local codes prohibit the use of city water as the makeup water in cooling towers are air-cooled centrifugal chillers adopted.

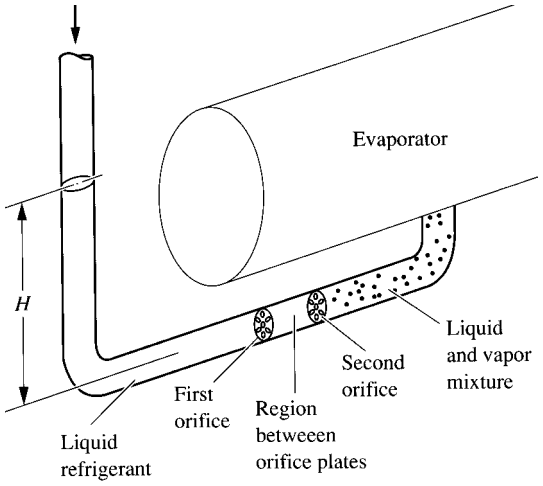


FIGURE 13.3 Orifice plates.

Flash Cooler. A flash cooler is sometimes called an economizer. It flashes a small portion of refrigerant at the intermediate pressure to cool the remaining liquid refrigerant to the saturated condition, in order to provide a greater refrigeration effect, as described in Secs. 9.13 and 9.14. For a two-stage compressor, a single-stage flash cooler is used. For a three-stage compressor, a two-stage flash cooler is used.

Orifice Plates or Float Valves. Orifice plates (arranged in series) or float valves are used as throttling devices in centrifugal chillers that employ flooded refrigerant feed. The use of multiple orifices as a throttling device is shown in Fig. 13.3. This device controls the amount of refrigerant feed to the liquid cooler according to the pressure of the liquid refrigerant in the condenser.

During full-load operation, a certain liquid pressure is maintained before the first orifice plate. When liquid refrigerant flows through the first orifice plate, it encounters a pressure drop. However, the fluid pressure between the two orifice plates is still higher than the saturated pressure of the liquid. No flash of vapor occurs in the region between the orifice plates. The second orifice plate meters the maximum refrigerant flow.

As the refrigeration load in the centrifugal chiller drops, less refrigerant is delivered to the condenser and evaporator, resulting in a lower condensing pressure and a higher evaporating pressure. This reduces the pressure exerted on the liquid refrigerant at the condenser due to lower condensing pressure and hydrostatic head. Because of the pressure drop across the first orifice plate, the fluid pressure between the orifice plates is less than the saturated pressure of the liquid refrigerant. Flashing of vapor occurs. Both liquid and vapor flow through the second orifice plate. Consequently, the mass flow rate of the refrigerant flow is reduced.

There are no moving parts in an orifice plate throttling device. It is simple and reliable in operation.

Purge Unit

At an evaporating temperature T_{ev} = 40°F (4.4°C), HCFC-123 has a saturated pressure of 5.8 psia (40 kPa abs.) which is far lower than atmospheric pressure. Noncondensable gas such as air and carbon dioxide may diffuse and leak into the evaporator through cracks and gaps. Because

condenser is always located at a higher level in a centrifugal chiller, any leakage of noncondensable gases into the system usually accumulates in the upper part of the condenser.

The presence of noncondensable gases reduces the refrigerant flow as well as the rate of heat transfer in the evaporator and the condenser. It also increases the pressure in the condenser. All these result in a loss of refrigeration capacity and an increase of power consumption. Wanniarachchi and Webb (1982) showed that at 60 percent load, the presence of 3 percent noncondensable gas causes a 2.6 percent increase in the power input to the compressor. The purpose of the purge unit is to purge periodically any noncondensable gases that have accumulated in the condenser.

According to the manufacturer's catalog in 1996, a purge unit developed in the mid-1990s is shown in Fig. 13.4. This purge unit includes mainly a purge tank, which is a low-temperature and low-pressure chamber with a DX coil to condense the vapor refrigerant; a condensing unit to serve the DX coil; a vacuum pump to purge the noncondensable gases; and a filter dryer to absorb moisture and to remove dirt. Noncondensable gases and gaseous refrigerant from the upper part of the condenser migrate to the low-temperature and low-pressure purge tank through a pipe connected to the bottom of the tank. When the latent heat of condensation is absorbed by the DX coil, the gaseous refrigerant condenses to liquid and falls to the bottom of the tank. Moisture contained in the refrigerant is separated by the water separation tube and forms the middle layer. All the noncondensable gases accumulate at the top of the tank because of their lower densities.

The purge vacuum pump operates periodically to extract the noncondensable gases and rejects them to the atmosphere. It may be operated automatically once a day according to the signal from the DDC unit controller; or it may be operated in adaptive mode, i.e., adjusting its operation to

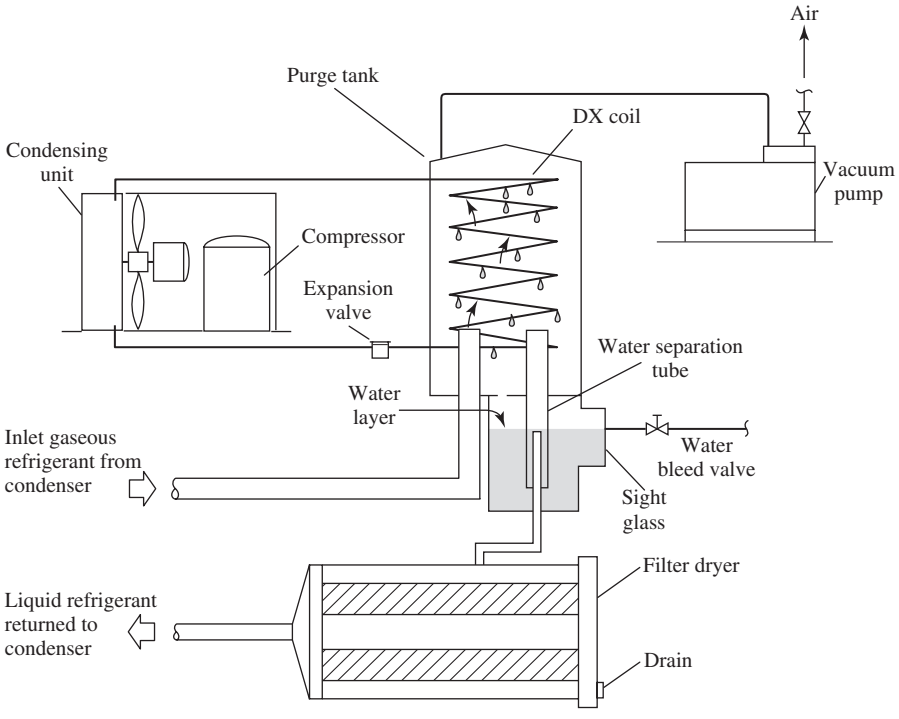


FIGURE 13.4 A purge unit in a centrifugal chiller.

match the chiller leak rate. The purge unit can be operated at any time independently of the chiller's operation.

The efficiency of this air purge unit is 0.0049 lb of refrigerant loss per lb of air purged (0.49 percent). Dietrich (1993) and Parsnow (1993) reported that many old type purge units need a refrigerant loss from 0.1 to 0.75 lb of refrigerant per lb of air purged. Sometimes their efficiencies are even worse than that. To select a high-efficiency air purge unit with less refrigerant loss causes less ozone depletion, which is always important for chillers using HCFC as refrigerant and is also economical.

Types of Centrifugal Chiller

Centrifugal chillers may be classified according to the configuration of their main components as follows:

- Single-stage or multistage
- Air-cooled, water-cooled, or double-bundle condenser (A double-bundle condenser has two bundles of condensing tubes for the purpose of heat recovery.)
- Open or hermetic
- Direct-drive or gear-drive
- Capacity control provided by inlet vanes or variable-speed drive

As mentioned before, because a centrifugal compressor needs a high peripheral velocity, it is not economical to fabricate small centrifugal machines. Their refrigeration capacities range from 100 to 10,000 tons (315 to 31,560 kW). Because of their large capacity, DX coils are not used as evaporators to condition air at such a large size and capacity, with their unacceptable air transporting and distribution losses.

13.2 WATER-COOLED CENTRIFUGAL CHILLER

Refrigerant Flow

A typical water-cooled centrifugal chiller that uses a three-stage, hermetic, and direct-drive compressor is illustrated in Fig. 13.1. As described in Sec. 9.14 and shown in Fig. 9.9*a* and *b*, the flow process of HCFC-123 is as follows:

Vapor refrigerant is extracted from the liquid cooler as a result of the suction of the compressor:

- It enters the first-stage impeller through inlet vanes.
- Hot gas from the first-stage impeller mixes with flashed vapor from the low-pressure (second-stage) flash cooler.
- The mixture enters the second-stage impeller.
- Hot gas from the second-stage impeller mixes with flashed vapor from the high-pressure (first-stage) flash cooler.
- The mixture enters the third-stage impeller.
- Hot gas discharged from the third-stage impeller enters the water-cooled condenser.
- Hot gas is desuperheated, condensed, and subcooled to liquid form in the condenser.
- A small portion of the liquid refrigerant is used to cool the hermetic motor.
- Most of the liquid refrigerant enters the high-pressure flash cooler through a multiple-orifice throttling device; and a small portion flashes into vapor and mixes with the hot gas discharged from the second-stage impeller.

- Most of the liquid refrigerant from the high-pressure flash cooler enters the low-pressure flash cooler through a multiple-orifice throttling device; and a small portion flashes into vapor and mixes with the hot gas discharged from the first-stage impeller.
- Most of the liquid refrigerant from the low-pressure flash cooler enters the evaporator through a multiple-orifice device.
- Liquid refrigerant vaporizes to vapor in the evaporator and produces its refrigeration effect there.

Performance Rating Conditions

According to ARI Standard 550/590-98, the cooling capacity of a centrifugal chiller is rated under the following conditions:

Water temperature leaving evaporator:

100 percent load	44°F (6.7°C)
0 percent load	44°F (6.7°C)
Chilled water flow rate:	2.4 gpm/ton (0.43 L/s·kW)

Percent load, entering condenser water temperature (ECWT), entering dry-bulb (EDB) temperature, and entering wet-bulb (EWB) temperature are as follows:

Percent load	Weighting of part-load, percent	Water-cooled ECWT, °F (°C)	Air-cooled EDB, °F (°C)	Condenser EWB, °F (°C)
100	1	85 (29.4)	95 (35.0)	75 (23.9)
75	42	75 (23.9)	80 (26.7)	68.75 (20.4)
50	45	65 (18.3)	65 (18.3)	62.5 (17.0)
25	12	65 (18.3)	55 (12.8)	56.25 (13.5)

Condenser water flow rate	3.0 gpm/ton (0.054 L/s·kW)
Fouling factor in evaporator:	water side 0.0001 h·ft^2·°F/Btu (0.000018 m^2·C/W), air side 0
Fouling factor in condenser:	water side 0.00025 h·ft·°F/Btu (0.000044 m^2·C/W), air side 0

The integrated part-load value (IPLV) of the centrifugal chiller at standard rating conditions can be calculated by the following formula, as specified by ARI 550/590-98:

$$\text{IPLV} = \frac{1}{0.01/A + 0.42/B + 0.45/C + 0.12/D} \tag{13.1}$$

where A, B, C, D = kW/ton or COP at 100, 75, 50, and 25 percent load, respectively. If the operating conditions are different from the standard rating conditions for the same centrifugal chiller, a general rule for purposes of estimation is as follows:

- When the temperature of chilled water leaving the chiller T_{el} is between 40 and 50°F (4.4 and 10°C), for each 1°F (0.56°C) increase or decrease of T_{el}, there is roughly a 1.5 percent difference in cooling capacity and energy use.
- When the entering temperature of condenser water T_{ce} is between 80 and 90°F (26.7 and 32.2°C), for each 1°F (0.56°C) increase or decrease of T_{ce}, there is roughly a 1 percent difference in cooling capacity; there is roughly a 0.6 percent difference in energy use.

For water-cooled centrifugal chillers with hermetic compressors, manufacturers' products that are currently available have a cooling capacity of 150 to 2000 tons (527 to 7032 kW). For water-cooled centrifugal chillers with open compressors, the cooling capacity varies from 150 to 10,000 tons (527 to 35,160 kW).

For hermetic compressors using vaporized liquid refrigerant to cool the motor, no additional motor cooling system is required. Leakage of refrigerant from the bearing seals is also less than that in the open compressor. Its main drawback is that 2 to 4 percent of the liquid refrigerant must be used to cool the motor. Its cooling capacity is reduced, and its energy use is increased.

Unfortunately, the IPLV in ARI Standard 550/590-98 is developed to represent the average part-load efficiency of of a single chiller. Most of the chillers are used in multichiller operation. Actual multichiller operation is different from the single-chiller operation.

13.3 CENTRIFUGAL CHILLERS INCORPORATING HEAT RECOVERY

System Description

Many large commercial applications often require heating in the perimeter zone but cooling in the interior zone during the winter. If the building is considered as a whole, a heat recovery system can be used to transfer the internal heat from the interior zone to the perimeter zone to offset the winter heating load there.

A typical centrifugal chiller incorporating a heat recovery system using a double-bundle condenser is presented in Fig. 13.5. In a double-bundle condenser, water tubes are classified as tower bundles and heating bundles. Heat rejected in the condenser may be either discharged into the atmosphere through the tower bundle and cooling tower or used for heating in the coils of the perimeter zones through the heating bundle. The tower bundle may be enclosed in the same shell with the heating bundle, as shown in Fig. 13.5*b*, but separate plates are needed to divide the water circuits.

A storage tank stores the excessive hot water from the heating bundle when the building is occupied. Hot water is then used for heating in the perimeter zone when the building is either occupied or unoccupied. A supplementary heater using, e.g., electric power may be used to raise the temperature of the hot water from the heating bundle, if necessary. A tower bypass line is always used to maintain the condenser water entering the condenser at a required value.

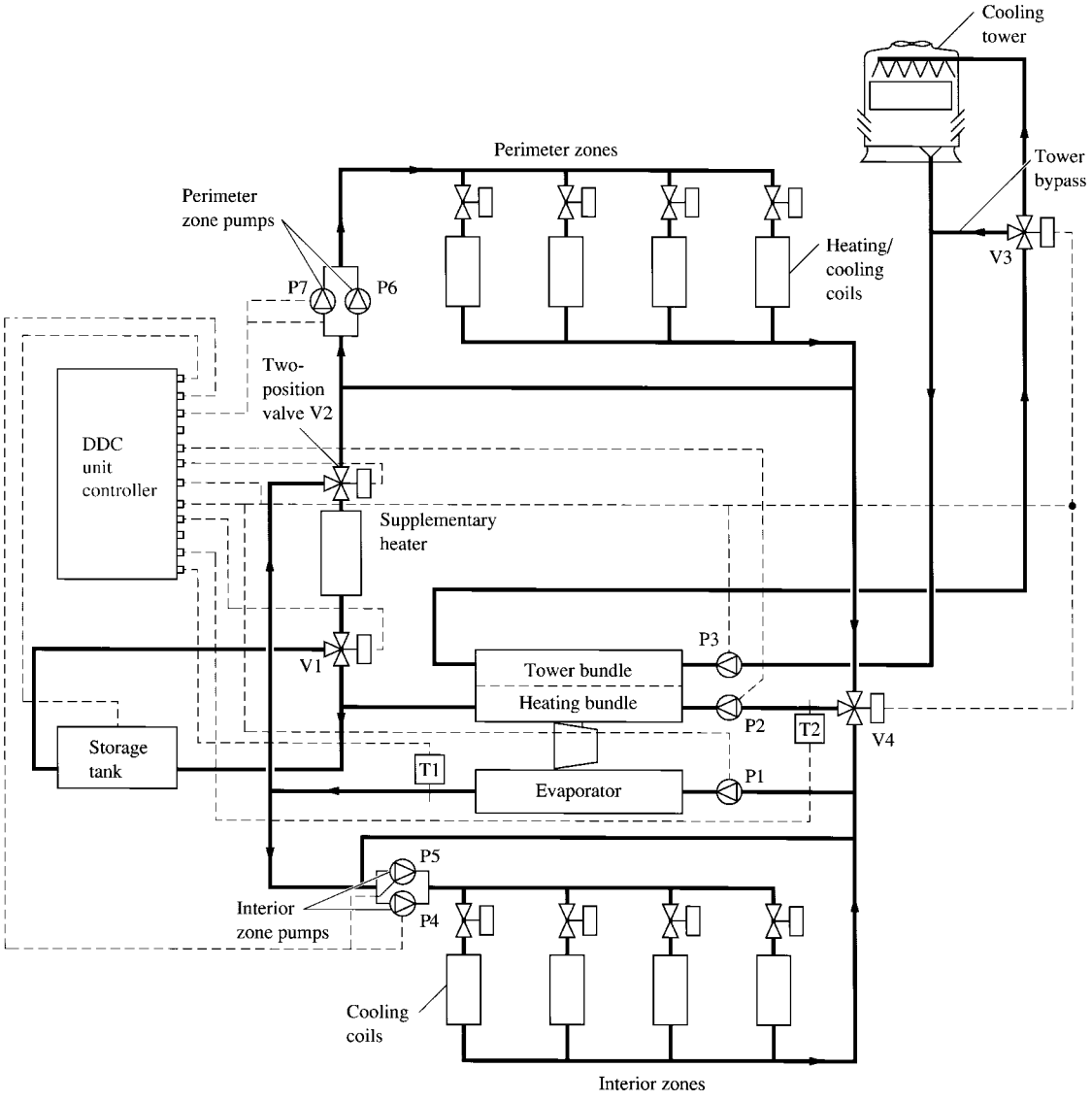
Instead of a double-bundle condenser, an auxiliary condenser plus a standard condenser can be used, as shown in Fig. 13.6. Hot gaseous refrigerant is forced to either the standard or the auxiliary condenser after it is discharged from the compressor. When heating is required in the perimeter zone, the auxiliary condenser pump is energized, and the tower bypass three-way valve bypasses part of or all the condenser water returning to the standard condenser. Because the water returning from the perimeter zone and entering the auxiliary condenser is colder than that entering the standard condenser due to the heating load in the perimeter zone, the auxiliary condenser tube bundle has a lower temperature and attracts the hot gas from the compressor. The auxiliary condenser recovers as much heat as it can allow. The remaining rejected heat goes to the standard condenser and is rejected to the atmosphere through the cooling tower. A balance of heat rejection between these two condensers automatically holds.

A hot water bypass should be included if an auxiliary condenser is used, as shown in Fig. 13.5. Such a bypass ensures that water will not circulate through the auxiliary condenser when the chiller is deenergized. In Fig. 13.5*a*, the hot water pump P2 is energized only when the compressor of the centrifugal chiller is in operation.

Operating Modes

Depending on whether heating is required in the perimeter zone, the operating modes of a centrifugal chiller that is incorporated with a heat recovery system in a building with perimeter and interior

zones can be divided into cooling mode and heating mode operation. Both the cooling and heating mode operations can be further divided into occupied and unoccupied periods, based on whether the conditioned space is occupied. The typical operation of a centrifugal chiller that is incorporated with a heat recovery system is described below.



(a)

FIGURE 13.5 Centrifugal chiller incorporating a heat recovery system: (*a*) schematic diagram; (*b*) double-bundle condenser.

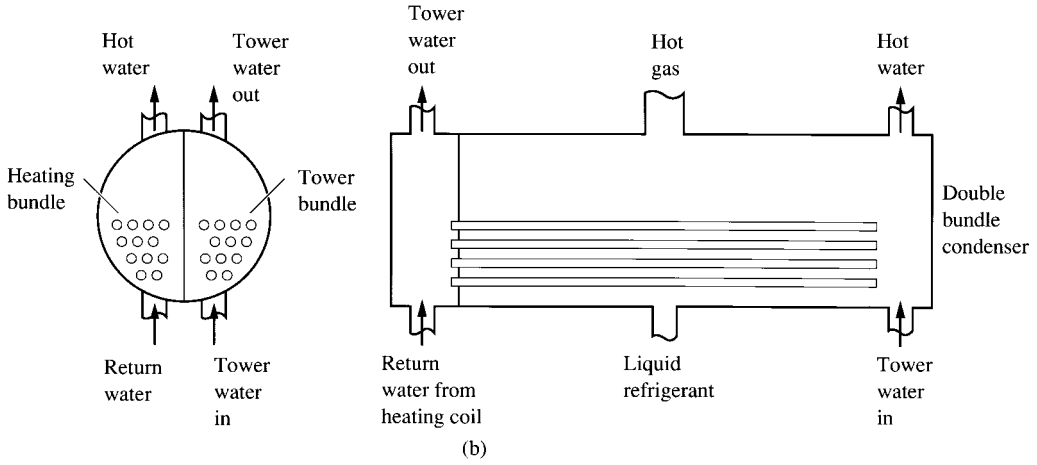


FIGURE 13.5 (*Continued*)

Occupied Cooling Mode. Cooling mode operation takes place when the space temperature in the perimeter zone $T_{r,p} \geq 75°F$ (23.9°C). The occupied cooling mode starts when the chilled water pump P1 is turned on (see Fig. 13.5*a*). Meanwhile, hot water pump P2 remains inactive; tower pump P3 forces all the condenser water to flow through the tower bundle, modulation valve V3, and back to the cooling tower. Chilled water is then supplied to the perimeter zone through zone pump

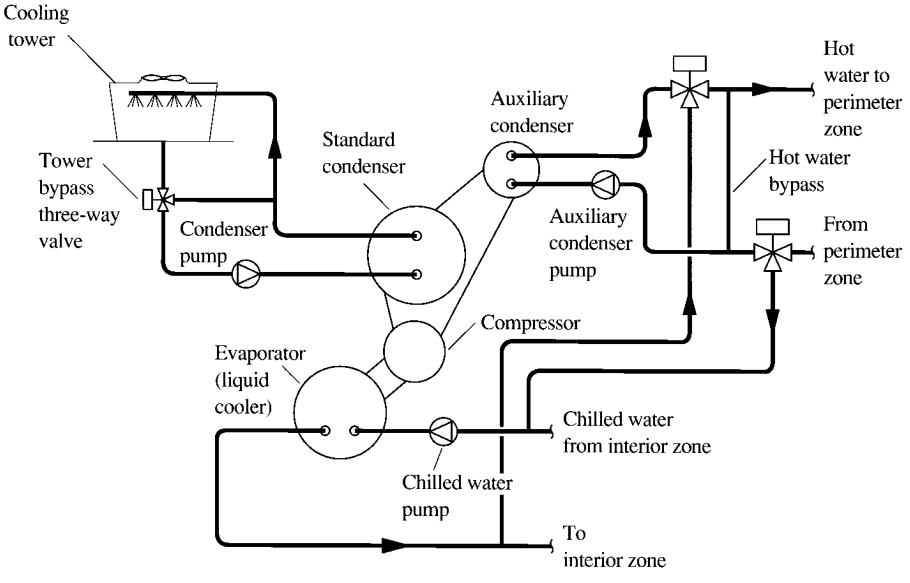


FIGURE 13.6 Heat recovery system using an auxiliary condenser.

P6 or standby pump P7 and to the interior zone through zone pump P4 or standby pump P5. The chilled water supply temperature T_{ws} is usually between 42 and 45°F (5.6 and 7.2°C). Chilled water pump P1 and hot water pump P2 are both constant-speed, constant-flow pumps. Zone pumps P5, P6, P7, and P8 are all variable-speed, variable-flow pumps.

Unoccupied Cooling Mode. When $T_{rp} \geq 75$°F (23.9°C) and the space is unoccupied, as determined by a timer, the DDC unit controller shuts down the entire centrifugal chiller and heat recovery system.

Occupied Heating Mode. When $T_{rp} < 70$°F (21.1°C) and the conditioned space is occupied, the system is under occupied heating mode operation. The centrifugal chiller is energized, and pump P1 supplies chilled water to the interior zone through zone pumps P4 and P5. At the same time, the DDC unit controller actuates the two-position valve V2 to block passage of chilled water to the perimeter zone. Hot water pump P2 is energized so that hot water from the heating bundle in the condenser is forced through the deenergized supplementary heater and supplied to heating coils in the perimeter zone through zone pump P6 or P7 at a temperature T_{sup} between 105 and 110°F (40.6 and 43.3°C). If the rejected heat in the heating bundle is more than that required in the heating coils in the perimeter zone, the return water temperature T_{ret} increases before the inlet of the heating bundle and pump P2. When sensor T2 detects such a rise in temperature, the DDC unit controller modulates valve V1, which permits some hot water to flow to the storage tank. When the storage tank is full of hot water and sensor T2 detects a further increase in water temperature, the modulation valve is positioned so as to allow all the hot water to bypass the storage tank. Tower pump P3 starts, and valve V3 is modulated to control the flow rate of condenser water to the cooling tower.

If the outdoor temperature T_o is so low that heat rejection at the heating bundle is not sufficient to meet the heating requirement at the perimeter zone, then the hot water temperature detected by sensor T2 is less than a preset value. The tower pump is then stopped. Hot water bypasses the storage tank, and the supplementary heater is gradually energized so as to maintain the water temperature sensed by T2 within a predetermined range.

Unoccupied Heating Mode. When $T_o < 70$°F (21.1°C) and the conditioned space is unoccupied, the DDC unit controller commands the following actions:

1. The hot water temperature detected by sensor T2 is reset to a value according to outdoor temperature T_o to maintain a night setback indoor temperature $T_r = 55$°F (12.8°C) and lower.
2. Only hot water pump P2 and zone pump P6 or P7 are in operation; all other equipment ceases to operate.
3. Valve V1 is modulated so that hot water from the storage tank alone is supplied to the heating coils in the perimeter zone.

If T2 senses a water temperature lower than the preset value, the supplementary heater is then gradually energized. The modulating valve V1 allows the hot water to bypass the storage tank and then enter the heaters.

System Characteristics

First, compared with a cooling-only centrifugal chiller, the refrigeration cycle of a centrifugal chiller incorporating a heat recovery system has a greater pressure differential between condensing and evaporating pressure $\Delta p_{e,c}$, in psi (kPa). This is because a cooling-only chiller usually has water leaving the condenser at a temperature at 90 to 95°F (32.2 to 35°C) during summer design conditions, whereas a chiller incorporating heat recovery needs a higher condenser leaving temperature T_{cl} of 105 to 110°F (40.6 to 43.3°C) for the hot water supply in winter heating. Although the evaporating temperature can be slightly higher in winter than in summer, an increase of 10 to 12°F

(5.6 to 6.7°C) in temperature difference between condensing and evaporating temperature is roughly 10 to 15 percent more power input to the compressor.

Second, during occupied heating mode at part-load operation, to maintain a preset T_{ret} and let T_{sup} float results in a lower mean condenser water temperature $T_{m,con}$; instead maintain a preset T_{sup} and let T_{ret} float. A lower mean $T_{m,con}$ reduces the power input to the compressor.

Third, to be more energy-efficient, T_{cl} should be reduced in part-load operation during winter heating. Condenser water entering temperature T_{ce} should be adjusted to provide an optimum hot water supply temperature T_{cl} at part-load operation.

Fourth, an air economizer reduces the heat recovery effect. This result is due to the reduction of the refrigeration load, which also reduces the heat recovery from condensing heat.

Finally, for the same building with equal areas of conditioned space, the heating load is smaller than the cooling load in many locations in the United States. For a refrigeration plant with multiple centrifugal chillers, usually only some of them include heat recovery. The remainder are cooling-only centrifugal chillers, which are more efficient.

13.4 AUTOMATIC BRUSH CLEANING FOR CONDENSERS

Principles and Operation

The condenser water from the cooling tower is usually in an open-circuit system. As described in Sec. 10.3, although many well-maintained cooling towers may have a long-term fouling factor for condenser tube surfaces of 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot °C/W$), the fouling factor for condenser tubes may exceed 0.0005 $h \cdot ft^2 \cdot °F/Btu$ (0.000088 $m^2 \cdot °C/W$) when a significant amount of suspended solids is contained in the condenser water. High fouling factors result in a high condensing pressure and higher kW/ton. ARI Standard 550-88 specifies a field fouling allowance of 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot °C/W$) for condensers.

The purpose of an automatic brush cleaning system is to remove deposits and dirt from the water-side tube surface of a condenser for which the fouling factor of the condenser tubes is higher than 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot °C/W$). Cleaning brushes in a condenser tube of a horizontal, two-pass, and shell-and-tube condensers are illustrated in Fig. 13.7. Special nylon brushes are inserted inside the tube, and catch baskets are attached to each end of the tube.

When condenser water flows through the condenser tubes, the difference in the pressure across the brush, approximately 1 psi, causes the brush to move in the direction of water flow. As the brushes in the condenser move, they remove deposit from the inner tube surfaces. The deposits are carried away by the condenser water. The brushes are then caught by the baskets at one end. The time taken for the brushes to travel the whole length of the tube is about 1 to 2 min.

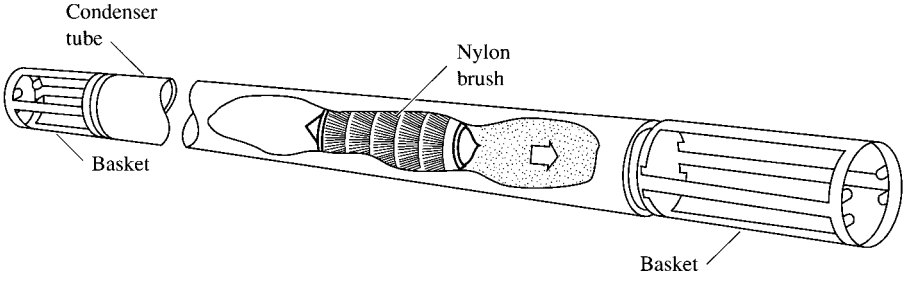


FIGURE 13.7 Cleaning brushes in a condenser tube. (*Source: Water Services of America. Reprinted with permission.*)

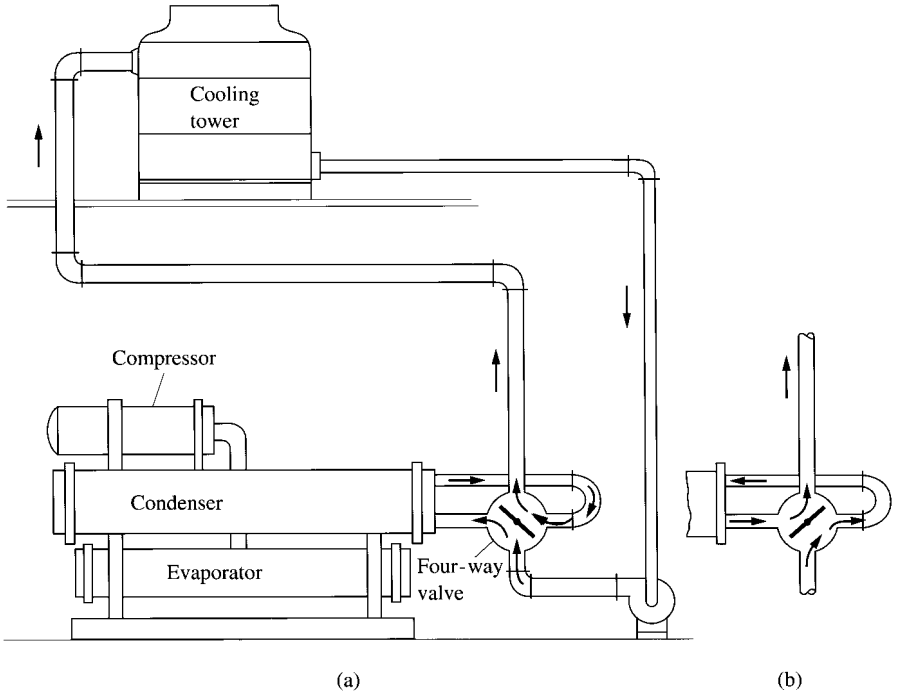


FIGURE 13.8 (*a*) Normal and (*b*) reverse operation of a four-way valve in an automatic brush cleaning system.

A special four-way valve is mounted at the condenser water entrance, as shown in Fig. 13.8*a*. When the four-way valve is in normal operation, the upper half of the brushes are caught by the baskets at the right-hand end of the condenser tube, and the lower half by the left-hand end baskets. During normal operation, the condenser water enters the condenser through the lower entrance so as to provide better subcooling for the liquid refrigerant accumulated at the bottom of the condenser.

After a predetermined time, a signal is given to the control system to unload the compressor by closing the inlet guide vanes or reducing the speed of the compressor. The four-way valve reverses the water direction, which causes the brushes to move in the opposite direction, from right to left for the upper half brushes and from left to right for the bottom half, until they are caught by the baskets at the ends. As soon as the brushes are caught, a signal is given to the control system to reverse the water flow again, so that the four-way valve is in normal operation. The brushes stay in the end baskets during normal operation for about 1 to 4 h until a signal reverses the water flow to repeat the process. The frequency of the cleaning cycle depends on the contaminants contained in the condenser water. The nylon brushes usually have a service life of 4 to 5 years.

Effect of Brush Cleaning System

Automatic brush cleaning systems have been found to accomplish the following results:

- The fouling factor drops to 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot °C/W$) or below. Consequently, energy costs are lower.

- The maintenance cost can be reduced because the interval for cleaning the condenser tubes is extended from 6 to 12 months or even longer.
- Chemicals for water treatment are still required, but the quantity can be reduced appropriately according to the actual operating conditions.

Walker (1994) reported that the Walt Disney World Dolphin Hotel used an automatic brush cleaning system that kept the fouling factor of the condenser tubes around 0.0001 $h \cdot ft^2 \cdot °F/Btu$ (0.000018 $m^2 \cdot °C/W$), so that the investment of the automatic tube-cleaning system was paid back in just slightly less than 3 years.

13.5 CENTRIFUGAL COMPRESSOR PERFORMANCE MAP

The performance of a centrifugal compressor can be illustrated by a compressor performance map or, simply, a compressor map. Like a centrifugal fan, a compressor map includes the volume flow, specific work or power input, efficiency, surge region, and opening of the inlet vanes. Volume flow and system head are expressed by the compressor performance curve or, simply, compressor curve. The centrifugal compressor can be operated either at constant speed or at variable speed.

Surge of Centrifugal Compressor

Surge is defined as the unstable operation of a centrifugal compressor or fan with vibration and noise. Surge occurs when the compressor is unable to develop a discharge pressure that is sufficient to satisfy the requirement at the condenser. During compressor surge, the pressure flow characteristics of the compressor fluctuate up and down along the compressor curve, resulting in pressure and flow pulsation and noise.

Surge first develops at the point where the slope of the pressure lift versus volume flow Δp-$\dot{V}$ curve on the compressor performance curve is zero, as shown in Fig. 13.9. Pressure lift is the total

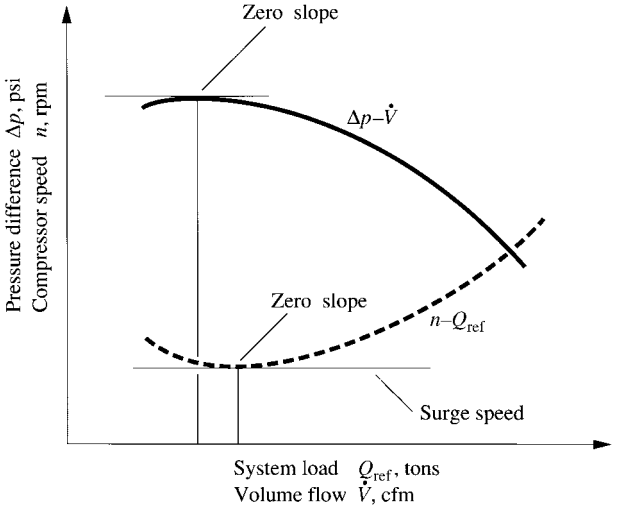


FIGURE 13.9 Compressor surge.

pressure rise of a centrifugal compressor, in psi (kPa). For a variable-speed centrifugal compressor, surge first develops at a point of zero slope along the curve that characterizes compressor speed n versus refrigeration load Q_{ref}, in tons (kW) (the n-Q_{ref} curve). This point is determined by assuming a rotating speed, in rpm, that is lower than the minimum possible speed at a required pressure lift and volume flow of the compressor.

A three-stage centrifugal compressor has a significantly smaller surging region than a two-stage or a single-stage centrifugal compressor. A single-stage centrifugal compressor using inlet vanes to control its capacity has a comparatively smaller surging region than a variable-speed compressor.

Surging may damage the components of the compressor. A centrifugal compressor should never be operated within the surge region on a compressor map.

Centrifugal Compressor Map at Constant Speed

A compressor map of a single-stage centrifugal compressor operated at constant speed is presented in Fig. 13.10*a*. The abscissa is the percentage of design volume flow of refrigerant at compressor suction $100\,\dot{V}_{suc}/\dot{V}_{suc,d}$, or load ratio

$$\text{LR} = \frac{Q_{ref}}{Q_{ref,d}} \tag{13.2}$$

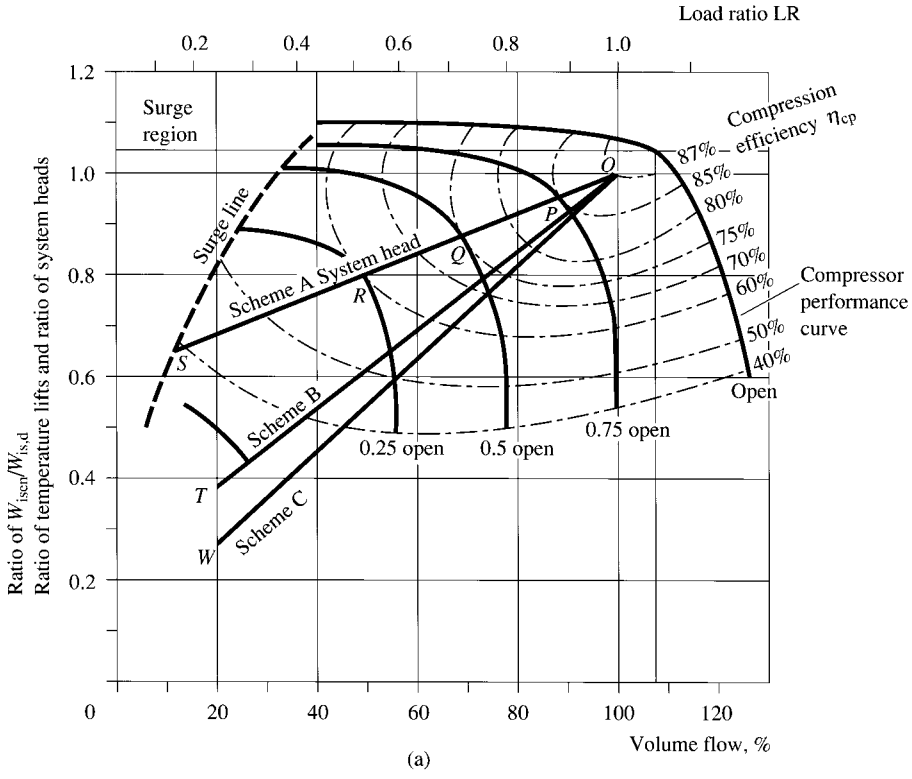


FIGURE 13.10 Compressor map of a single-stage centrifugal compressor: (*a*) constant speed, inlet vane modulation; (*b*) variable speed. (*Adapted with permission from ASHRAE Handbook 1988, Equipment.*)

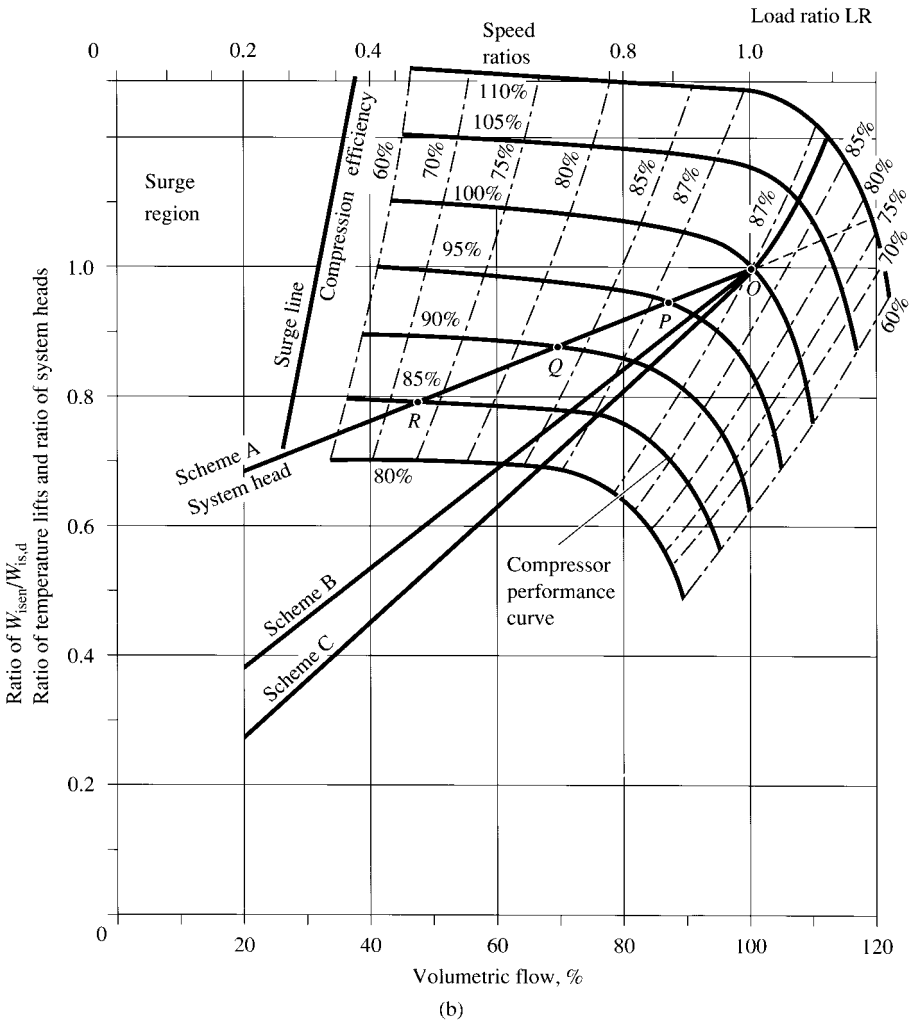


FIGURE 13.10 (*Continued*)

where $Q_{\text{ref},d}$ = refrigeration load at design condition, tons (kW). The ordinate is the ratio of isentropic work W_{isen} to the isentropic work at design condition $W_{\text{is},d}$, or $W_{\text{isen}}/W_{\text{is},d}$. Isentropic work is defined as the work required for isentropic compression per unit mass of refrigerant, as described in Sec. 9.20. For the same compressor operated at the same conditions, the ratio $W_{\text{isen}}/W_{is,d}$ is approximately equal to the ratio of temperature lifts $R_{T,1}$ and the ratio of system heads R_{head}, that is,

$$R_{T,1} = \frac{T_{\text{con}} - T_{\text{ev}}}{T_{\text{con},d} - T_{\text{ev},d}} \tag{13.3}$$

and
$$R_{\text{head}} = \frac{p_{\text{con}} - p_{\text{ev}}}{p_{\text{con},d} - p_{\text{ev},d}} \tag{13.4}$$

where T_{con}, $T_{\text{con},d}$ = condensing temperature and condensing temperature at design condition, respectively °F (°C)
T_{ev}, $T_{\text{ev},d}$ = evaporating temperature and evaporating temperature at design condition, respectively °F (°C)
p_{con}, $p_{\text{con},d}$ = condensing pressure and condensing pressure at design condition, respectively, psig (kPag)
p_{ev}, $p_{\text{ev},d}$ = evaporating pressure and evaporating pressure at design condition, respectively, psig (kPag)

System head, or head, is the total pressure rise of the refrigerant by the compressor, in psi (kPa). In a centrifugal refrigeration system, system head is mainly used to raise p_{ev} to p_{con}.

The relationship between the power input to the compressor P_{in}, in hp, and the isentropic work W_{isen} is

$$P_{\text{in}} = \frac{\dot{V}_{\text{suc}} \rho_{\text{suc}} W_{\text{isen}}}{42.41 \eta_{\text{com}}} \tag{13.5}$$

where $\dot{V}_{\text{suc}}$ = volume flow of suction vapor, cfm
ρ_{suc} = density of suction vapor, lb/ft^3
η_{com} = compressor efficiency

In Fig. 13.10*a*, compression efficiency η_{cp}, as defined in Sec. 9.20, can be calculated as

$$\eta_{\text{cp}} = \frac{\eta_{\text{com}}}{\eta_{\text{mec}}} \tag{13.6}$$

where η_{mec} = mechanical efficiency [see Eq. (9.67)]. Compression efficiency usually drops when the load ratio falls below 1, as shown by the constant η_{cp} contours.

In Fig. 13.10*a*, "open, 0.75 open, . . ., closed" indicates the fraction of opening of the inlet vanes, to be discussed in Sec. 13.6. For each fraction of inlet vanes, there is a corresponding compressor performance curve that relates the load ratio and R_{head}.

In Fig. 13.10*a*, there are three required system head (p_{con}-p_{ev}) curves, illustrated by straight lines *OS*, *OT*, and *OW*. The system head drops when the load ratio falls below 1. In the top left corner of Fig. 13.10*a*, a surge line separates the surging region of this single-stage centrifugal compressor from the normal operating region. For two- or three-stage centrifugal compressors, the surge region is smaller.

Centrifugal Compressor Map at Variable Speed

A compressor map of a single-stage centrifugal compressor operating at variable speed is illustrated in Fig. 13.10*b*. The abscissa, ordinate, and required system head curves are the same as those in the compressor map at constant speed (see Fig. 13.10*a*). The inlet vanes are fully open at both design load and part-load operations. Their main differences are these:

- Compressor curves are curves at various rotating speeds, such as 105 percent, 100 percent, . . ., 80 percent of speed at design condition, instead of curves at various inlet vane openings on a constant-speed map.
- At lower heads, the surge region is enlarged.

13.6 CAPACITY CONTROL OF CENTRIFUGAL CHILLERS

Capacity control of a centrifugal chiller is achieved mainly by modulating the volume flow of the refrigerant at the centrifugal compressor.

Difference between Centrifugal Compressors and Centrifugal Fans

During part-load operation, the following means are used to modulate the capacity of a centrifugal compressor:

- Open the inlet vanes.
- Use an adjustable-frequency ac inverter to vary the compressor rotating speed. Adjustable-frequency variable-speed drive is discussed in Sec. 15.4.
- Combine the opening of inlet vanes and variable-speed control by means of an inverter.
- Vary the speed of the steam- or gas-turbine-driven large centrifugal chillers.

A hot-gas bypass provides no energy saving at reduced system loads as specified in Sec. 11.5.

Although centrifugal compressors and centrifugal fans are both centrifugal turbomachinery, and both centrifugal chillers and centrifugal fan duct systems use inlet vanes and variable-speed drives to provide capacity control during part-load operation, there is an important difference between the centrifugal chiller and centrifugal fan duct system. In a fan duct system, the total pressure of the centrifugal fan is used mainly to overcome the friction and dynamic losses in airflow. This required fan total pressure drops considerably when the air volume flow is reduced. In the centrifugal chiller, on the other hand, the system head is used mainly to lift the evaporating pressure to condensing pressure to produce the refrigerant effect. The pressure drop due to friction and dynamic losses of refrigerant flow is only a small part of the required head. Reduced volume flow of refrigerant at part-load operation has a minor influence on the required system head.

The primary factor that affects the system head at part-load operation is the difference between condensing and evaporating pressures or temperatures, $p_{con} - p_{ev}$ or $T_{con} - T_{ev}$. For a water-cooled condenser using condenser water cooled from a cooling tower, the outdoor wet-bulb temperature directly influences the entering temperature of condenser water and, therefore, the condensing temperature and pressure.

Required System Head at Part-Load Operation

During part-load operation, because the condensing and evaporating surface areas at reduced volume flow of refrigerant are greater than the required area, there is a drop in condensing temperature T_{con} and an increase in evaporating temperature T_{ev}. Most probably, the outdoor wet-bulb temperature also drops at reduced load. Therefore, the profiles for required system head at part-load operation for a water-cooled centrifugal chiller usually fall into the following three schemes:

- *Scheme A*. Constant 85°F (29.4°C) condenser water entering temperature T_{ce} and constant 44°F (6.7°C) chilled water leaving temperature T_{el}: The temperature differential $T_{con} - T_{ev}$ at a load ratio equal to 1 is 66°F (36.7°C). At a load ratio of 0.2, $T_{con} - T_{ev}$ is about 45°F (25°C).
- *Scheme B*. Constant T_{el} and a drop of 2.5°F (1.4°C) in T_{ce} for each 0.1 increment of reduction of load ratio. The quantity $T_{con} - T_{ev}$ at design load is still 66°F (36.7°C), whereas at load ratio = 0.2, $T_{con} - T_{ev}$ is about 26°F (14.4°C).
- *Scheme C*. A reset of T_{el} of 1°F (0.56°C) increase and a drop of 2.5°F (1.4°C) of T_{ce} for each 0.1 increment of reduction of load ratio. Under this scheme, $T_{con} - T_{ev}$ at design load is still 66°F (36.7°C), and at 0.2 load ratio, $T_{con} - T_{ev}$ is about 19°F (10.6°C).

Required system head curves for schemes A, B, and C at various load ratios are shown in Fig. 13.10.

On a compressor map (e.g., Fig. 13.10*a*), the intersections of the compressor performance curve and the required system head curve at that load ratio—points *P*, *Q*, *R*, . . . —are the operating points of the centrifugal chiller. The required system head curve joins the operating points together, to form an operating curve of the centrifugal chiller at various load ratios.

Capacity Control Using Inlet Vanes

Inlet vanes are pivoted movable vanes installed at the inlet of the impeller, as shown in Fig. 13.1. In Fig. 13.1 only one set of inlet vanes is shown. For a three-stage centrigugal chiller with three impellers, actually there are three sets of inlet vanes, one for each impeller, to improve the chiller's performance. These vanes are linked mechanically so that they can turn simultaneously around the axis. When the opening of the inlet vanes at the inlet of the centrifugal compressor is reduced, it throttles the volume flow of vapor refrigerant and imparts a spin or prerotation on the refrigerant flow before it enters the impeller. If the direction of the spin is the same as the direction of rotation of the impeller, the tangential component of the absolute velocity of the fluid elements has a positive value. The head lift thus developed is smaller than if vapor refrigerant enters the impeller with a radial entry. This effect produces a new performance curve at a lower head and volume flow at each position of reduced vane opening.

The compression efficiency η_{com} decreases as the inlet vane opening angle decreases, as the vapor refrigerant at inlet is further deviated from the radial entry and the inlet area is reduced. At high load ratios with larger openings, compression efficiency is only slightly affected. At low load ratios, throttling at the inlet vanes causes considerable reduction of compression efficiency. However, the reduction of shaft power from the decreases in head lift Δp and volume flow of vapor refrigerant $\dot{V}$ compensates for the effect of the drop in η_{com} and shows energy saving from inlet vane capacity modulation.

For a single-stage centrifugal compressor using inlet vanes to modulate its capacity (see Fig. 13.10*a*), the following can be noted:

- If the condenser water entering temperature T_{ce} remains at the design condition, the operating curve hits the surge line when the load ratio drops to about 0.12.
- When the condenser water temperature drops 2.5°F (1.4°C) for every 0.1 increment of reduction of load ratio, at a load ratio of 0.5 the compression efficiency of the compressor then drops below 50 percent.

Capacity Control by Variable Speed

When an adjustable-frequency variable-speed drive is added to a centrifugal chiller, the rotating speed of the centrifugal compressor varies and a family of compressor curves is produced (see Fig. 13.10*b*). A lower rotating speed results in a performance curve of lower head and volume flow of refrigerant.

One can see the following in Fig. 13.10*b*:

- When the load ratio ≥ 0.5, the compression efficiency of a centrifugal compressor using variable-speed control is not less than 75 percent.
- If the condenser water entering temperature T_{ce} remains at the design value, the operating curve hits the surge line at a load ratio of about 0.27.
- If T_{ce} drops and T_{el} is reset according to scheme C, the operating curve intersects the surge line at a load ratio of about 0.18.

When capacity control is to be achieved by varying the compressor rotating speed, a reset of chilled water leaving temperature T_{el} should be used.

Comparison between Inlet Vanes and Variable Speed

Inlet vanes modulation is simple and has low initial cost. At lower load ratios, it functions as a throttling device and produces considerable dynamic and friction losses and, consequently, a lower compression efficiency than a variable-speed compressor. In Fig. 13.10, inlet vanes have a comparatively smaller surge region than in a variable-speed compressor in a single-stage centrifugal compressor when the load ratio falls below 0.3. However, a two- or three-stage centrifugal compressor significantly reduces the surging region.

Hubbard (1997) and Bellenger (1996) showed that the annual energy use of a centrifugal chiller with an adjustable-frequency variable-speed drive compressor is about 10 to 15 percent lower than that of a constant-speed compressor depending on the system characteristics of the chiller. The simple payback period for an adjustable-frequency variable-speed drive ranges from 5 to 10 years. Erth (1987) compared the power consumption of centrifugal chillers, in kW/ton, of constant-speed centrifugal compressors using inlet vanes and variable-speed compressors with or without a chilled water leaving temperature T_{el} reset, as shown in Fig. 13.11. At a load ratio of 0.25, variable-speed control with T_{el} reset has a power consumption of only about one-half its design value.

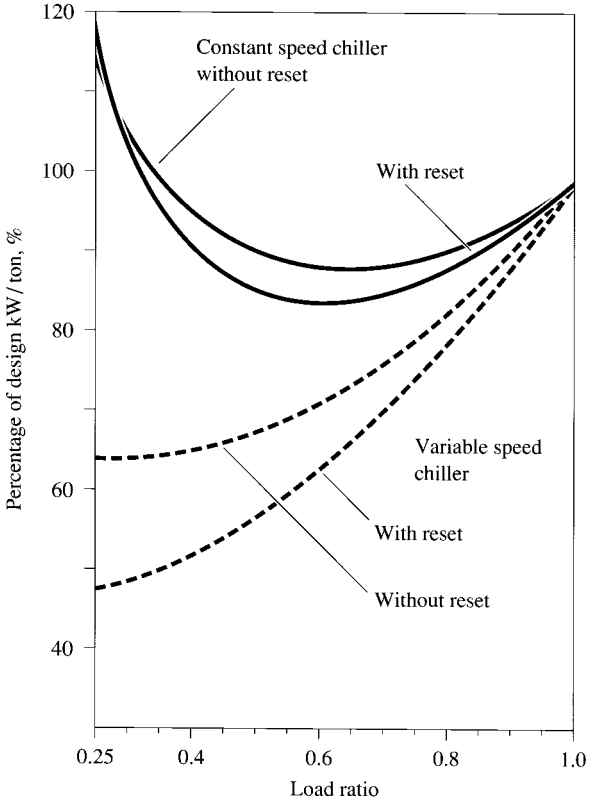


FIGURE 13.11 Comparison between constant-speed and variable-speed centrifugal chiller. (*Source: R. A. Erth, Heating/Piping/Air Conditioning, May 1987. Reprinted with permission.*)

13.7 CENTRIFUGAL CHILLER CONTROLS

Functional Controls and Optimizing Controls

A large modern centrifugal chiller plant usually includes multiple chillers to prevent the entire shutdown of the refrigeration system when one chiller fails to operate. Control of a centrifugal chiller plant, therefore, includes the functional controls and optimizing controls, as well as the unit control of a single chiller and the system control of a multiple-chiller plant. Unit control consists of specific, safety, diagnostic, and optimizing controls. Only a microprocessor-based DDC system can possibly perform all these duties. For functional controls, a DDC unit controller monitors and controls the following:

- Chilled water leaving temperature control
- Air purge
- Safety features, including oil pressure, low-temperature freezing protection at the evaporator, high condensing temperature control, motor overheating, and time delaying
- Status features, including condenser water flow, chilled water flow, electric current, and refrigerant and chilled water temperatures and pressures at key points

Chiller plant optimizing control with multiple centrifugal chillers includes the following:

- Optimize the on/off staging of the multiple chillers.
- Optimize start and stop.
- Control condenser water entering temperature.

Controls of the multiple chillers in a centrifugal chiller plant are discussed in later sections.

Chilled Water Leaving Temperature Control

Continuous modulation control is extensively used for chilled water leaving temperature T_{el} control in centrifugal chillers. A temperature sensor located in the chilled water pipes at the exit of the evaporator is used to sense T_{el}. The DDC unit controller either varies the position of the opening of the inlet vanes or varies the speed of the compressor motor through an adjustable-frequency ac invertor. The volumetric flow of refrigerant then decreases or increases, as does the capacity of the centrifugal chiller. Chilled water leaving temperature control is actually the capacity control of a centrifugal chiller.

The temperature of the chilled water leaving the evaporator T_{el} is often reset to a higher value at reduced load according to either the outdoor temperature T_o or the load ratio. The load ratio can be indicated by the chilled water return temperature. An increase in T_{el} may reduce the dehumidifying capability at the cooling coil, especially at very low load ratios.

A better scheme for T_{el} reset is this: When the outdoor temperature is high, that is, $T_o > 80°F$ (26.7°C), there is no reset. Only when T_o is less than 80°F (26.7°C) is T_o reset for an increase of 0.2 to 0.3°F (0.11 to 0.17°C) for each 1°F (0.56°C) drop in T_o. When T_o falls below 50°F (10°C), T_o remains constant.

For energy-efficient design and operation, as discussed in Sec. 7.11, ASHRAE/IESNA Standard 90.1-1999 specifies that chilled and hot water systems with a design capacity exceeding 300,000 Btu/h (87,900 W) supplying chilled water to comfort air conditioning systems shall include automatically reset chilled water leaving temperature according to building loads or outdoor temperature. A proportional-integral-derivative (PID) control mode is often used for modulation control of the chilled water leaving temperature for better performance.

Condenser Water Temperature Control

Interaction of the centrifugal chiller and the cooling tower must be considered when one optimizes the condenser water temperature entering the condenser T_{ce} so as to minimize power consumption. The control strategy is to minimize the total power consumption of the chiller and tower fans.

According to Cascia (1988), the condenser water temperature should be floated; i.e., it should rise and fall with the outdoor wet-bulb temperature in agreement with the following requirements:

- Temperature T_{ce} must be maintained above the lower limit specified by the centrifugal chiller manufacturer, such as 60°F (15.6°C), in order to maintain a necessary head pressure for proper operation. If the lower limit is reached, the tower fan must be turned off first; then the tower bypass valve should be modulated to maintain T_{ce} equal to or above the specified limit.
- If T_{ce} reaches the high limit, possibly more tower fans should be added so that T_{ce} does not exceed the high limit.

A temperature sensor should be used to sense T_{ce}. A DDC unit or system controller actuates the cooling tower fans and the cooling tower bypass valve to maintain the predetermined T_{ce}.

Safety Controls

Electric Demand Limit Control. Electric demand or electric current limit control is used to prevent the electric current from exceeding a predetermined value. A motor current is measured at the current, transformers, and a proportional signal is sent to the DDC unit controller. If the current exceeds a predetermined limit, the controller directs the inlet vanes to close or reduces the rotating speed of the compressor through the adjustable-frequency ac inverter. The limit could be from 40 to 100 percent of the electric demand.

Motor Protection. The compressor motor assembly is the most expensive part of the centrifugal chiller. For a typical control that protects a motor against high temperatures, if the resistance temperature detector in the motor stator winding senses a temperature under 165°F (73.9°C), the DDC unit controller introduces a time delay, normally of 4 min, after the chiller switch is turned on. If the sensed temperature exceeds 165°F (73.9°C), a start delay of 15 min is required. If the stator winding temperature exceeds a dangerous upper limit, such as 265°F (129.4°C), the DDC unit controller then prohibits the operation of the motor because of its high temperature.

There are also protection controls against current overload, temporary power loss, low voltage, and phase unbalance.

Low-Temperature and High-Pressure Cutouts. Refrigerant temperature at the evaporator is detected by a sensor. This information is sent to the DDC unit controller so that if the predetermined low-temperature cutout is approached, inlet vanes hold or close to provide freeze protection for the evaporator.

A pressure sensor detects the condensing pressure of the refrigerant. The DDC unit controller outputs override the inlet vane position instruction to close the inlet vanes if the condensing pressure reaches the predetermined limit.

High Bearing Temperature. If the bearing temperature detected by a temperature sensor exceeds a preset value, the chiller is stopped immediately.

Short-Cycling Protection. Usually a time delay, typically of 30 min, is needed to restart a centrifugal chiller after it has been turned off to prevent short cycling.

Surge Protection. A surge detection device is used to send a signal to the DDC unit controller. A head relief scheme is implemented by lowering the condenser water temperature. If the surge continues beyond a predetermined time period, the compressor is shut down.

Loss of Flow. If a loss of flow in evaporator or condenser is detected by the flow switch, the chiller will be shut down accordingly.

Air Purge. An automatic purge operation includes the operation of an air purge unit within a specific time interval, for instance, 5 min of air purge every 2 h. Both condensable gas and condensable vapor, mainly water vapor, leaking into the centrifugal chiller raises the purge drum pressure. If an increase in drum pressure is detected by a pressure sensor, the period of operation of the air purge unit is lenthened.

Fault Detection and Diagnostics

As discussed in Sec. 5.15, fault detection and diagnostics aid in detecting a problem, solving a problem, and getting the chiller on-line quickly in case there is a problem. For a typical diagnostic display module, more than 100 diagnostic checks and displays are available. Diagnostic modules are also able to display the run test to verify the thermistor, potentiometer, oil pump, air purge unit, and compressor before the chiller is started.

Operator Interface

All the inputs (such as switches and set points), output displays (such as operating mode indicators, key parameter monitors, and status lights), and diagnostic displays should be visible at a glance on the control display panel. They should be easy to understand and to use.

13.8 OPERATING CHARACTERISTICS OF CENTRIFUGAL CHILLERS

Sequence of Operation of a Typical Centrifugal Chiller

Many centrifugal chillers in commercial buildings are shut down at night in summer. Usually, one of the chillers is operated to serve areas needing 24-h service, such as computer areas. The temperature of chilled water in the shut-down system rises during the shutdown period. Chillers are often started earlier than air-handling units (AHUs) or terminals to pull down the chilled water temperature so that its T_{el} is maintained within acceptable limits.

When the switch of a centrifugal chiller is turned on, the sequence of operations to start and run a centrifugal chiller is as follows:

1. The sensed chilled water leaving temperature T_{el} is higher than the set point, indicating a demand for refrigeration.
2. The interlocked chilled water pump switch starts the chilled water pump, and the chilled water flow is sensed by the chilled water flow sensor.
3. A signal is issued to start the condenser water pump and the tower fans; the condenser water flow is sensed by the flow switch.
4. A signal is sent to start the oil pump, and the oil pressure is sensed by the pressure sensor.
5. The short-cycling protection timer is checked to see whether the time period after the previous shutdown is greater than the predetermined limit.
6. A command is sent to close the inlet vanes and is confirmed.

7. After the oil pressure has been established for 10 to 20 s, a start command is sent to the compressor motor starter.
8. A successful start is followed by a display of "run normal." Failure to perform a successful start shuts down the compressor, and the diagnostic code is displayed for correcting the fault. After correction and reset, the start sequence may be repeated to attempt a successful start again.
9. During "run normal" operation, the current limit, the condenser high-pressure limit, and the evaporator low-temperature cutout are monitored to see whether these limits have been approached. If so, the relevant controls hold or close the inlet vanes so that these limits are not exceeded. When a surge is detected, the head relief relay is energized. If surge continues for 15 min, the compressor will shut down.
10. When the DDC unit controller originates a stop signal based on the timer's schedule, a "preparing to shut down" message appears as the inlet guide vanes close or the variable-speed compressor unloads. The compressor and condenser pump motor starter are deenergized. The oil pump will continue to run for about 10 min. Finally, the chilled water pump for this chiller is deenergized. The chiller can be manually stopped at any time by pushing the Stop key.

System Balance at Full Load

When a centrifugal chiller is operating at full design load under the standard rating conditions, that is, $T_{el} = 44°F$ (6.7°C), and chilled water volume flow $\dot{V}_{ev} = 2.4$ gpm/ton (0.043 L/s·kW), $T_{ce} = 85°F$ (29.4°C) and condenser water volume flow $\dot{V}_{con} = 3$ gpm/ton (0.054 L/s·kW).

If the capacity of the selected evaporator is greater than the compressor at design conditions, more liquid refrigerant is vaporized in the evaporator than the compressor can extract. The evaporating pressure gradually rises until the increase in the mass flow rate due to a higher density of suction vapor extracted by the compressor is just equal to the mass flow rate of refrigerant vaporized in the evaporator. A new system balance forms between the evaporator and compressor at a higher evaporating temperature T_{ev}.

On the other hand, if the capacity of the selected compressor is greater than the evaporator, more evaporized refrigerant is extracted than the liquid refrigerant vaporized in the evaporator. The evaporating pressure and temperature gradually drop until the mass flow rate of suction vapor of the centrifugal compressor is just equal to the mass flow rate of the vaporized refrigerant in the evaporator.

If the capacity of the selected condenser is greater than the compressor at design conditions, more hot gas can be condensed to liquid form in the condenser than the compressor can supply. The result is a drop in condensing pressure p_{con} and temperature T_{con}. Because the condenser water entering temperature remains unchanged, a drop in T_{con} results in a lower log-mean temperature and a reduction in the condensing capacity. A new system balance is then formed between the mass flow rate of hot gas condensed in the condenser and that discharged from the centrifugal compressor at a lower T_{con}.

On the other hand, if the capacity of the selected centrifugal compressor is greater than the condenser, more hot gas is discharged to the condenser by the compressor than can be condensed in the condenser. The result is a rise in condensing pressure p_{con} and temperature T_{con}. A rise in T_{con} means a higher log-mean temperature and a greater condenser capacity, as well as a greater head lift, which reduces the discharged volume flow rate of the hot gas from the centrifugal compressor. Both effects are helpful to make a new system balance between the compressor and the condenser.

13.9 PART-LOAD OPERATION OF CENTRIFUGAL CHILLER

Part-Load Operating Characteristics

If we assume that a centrifugal chiller is operated initially at full design load and is in a condition of system balance, then chilled water leaving the evaporator is controlled at a temperature of 44°F (6.7°C), and the return temperature T_{ee} is 60°F (15.6°C).

If there is a reduction of coil load in the AHUs or in the terminals, the chilled water temperature entering the evaporator T_{ee} falls. Because the rate of heat transfer and the amount of refrigerant evaporated are only slightly affected, T_{el} is lower than 44°F (6.7°C), the set point.

When the temperature sensor at the exit of the evaporator senses a drop in T_{el} and signals the DDC unit controller, the positioner of the inlet vanes is actuated to close the opening. The centrifugal compressor is now operating at less than the design head, and the mass flow rate of the refrigerant extracted by the compressor is reduced from $\dot{m}_r$ to $\dot{m}_{r'}$. At part load, both the evaporating pressure p_{ev} and temperature T_{ev} tend to increase. Such an effect further reduces ΔT_m, the boiling heat-transfer coefficient h_b, and the overall heat-transfer coefficient U_o. The increase in mass flow rate due to a smaller head lift at part-load operation is offset by the closing of the inlet vanes of the compressor. The rate of heat transfer and the rate of evaporation are, therefore, decreased until the mass flow rate of evaporated refrigerant is equal to the mass flow rate of the refrigerant extracted by the centrifugal compressor.

Because the rate of condensation of refrigerant in the condenser is greater than the reduced mass flow rate of hot gas discharged from the compressor $\dot{m}_{r'}$, the condensing pressure p_{con} and temperature T_{con} tend to drop until the rate of condensation is equal to $\dot{m}_{r'}$. Consequently, there is a reduction of total heat rejection in the condenser.

A higher evaporating pressure and a lower condensing pressure result in a small pressure difference p_{con}-p_{ev}, head lift, as well as a lower hydrostatic liquid column in the condenser. As a result, the liquid refrigerant flow through the float valve or orifice plates is reduced.

When the compressor operates with the inlet vanes closed at a small angle, i.e., at a load ratio between 0.8 and 1, the compression efficiency decreases only slightly. However, because a lower head and a lower volume flow have a great effect on the work input and power consumption of the compressor, the result is a reduction in the power input. The energy performance index, in kW/ton, of the compressor is reduced less than power input because of the reduction of Q_{ref}.

The reduction in the mass flow rate of the refrigerant that flows through the centrifugal compressor and the orifice plates and the reduction of the rates of evaporation and condensation create a new system balance during part-load operation.

For a constant-speed centrifugal chiller, power consumption, in kW/ton, rises when the load ratio drops below 0.55 (see Fig. 13.11). This increase occurs because the fall in compression efficiency exceeds the savings in power consumption at a lower head and flow during low load ratios when inlet vanes are closed to a smaller opening.

Evaporating and Condensing Temperatures at Part-Load Operation

If other conditions remain the same, the greater the heat-transfer surface area A_o in the evaporator, the higher the evaporating temperature T_{ev}. Also, the greater A_o in the condenser, the lower the condensing temperature T_{con}. Consequently, greater A_o results in a lower temperature lift between T_{con} and T_{ev}.

From Eq. (10.9), if the thermal resistance of metal is ignored, the rate of heat transfer at the shell-and-tube evaporator and condenser can be calculated as

$$Q = U_o A_o \Delta T_m \tag{13.7}$$

$$U_o = \frac{1}{1/(\eta_f h_o) + R_f(A_o/A_i) + (A_o/A_i)(1/h_i)} \tag{13.8}$$

where η_f = fin efficiency; for integrated fins on copper tubes, its value is between 0.9 and 0.95. From Eq. (10.12), the boiling coefficient is given as

$$h_b = C\left(\frac{Q_{ev}}{A_o}\right)^n \tag{13.9}$$

Meanwhile, from Eq. (10.18), the condensing coefficient can be calculated as

$$h_{con} = C_{con}\left(\frac{1}{Q_{rej}/A_o}\right)^{1/3} \tag{13.10}$$

The log-mean temperature difference across the evaporating and condensing surfaces can be expressed as

$$\Delta T_m = \frac{T_{ee} - T_{\text{ev}} - (T_{el} - T_{\text{ev}})}{\ln[(T_{ee} - T_{\text{ev}})/(T_{el} - T_{\text{ev}})]} = \frac{T_{\text{con}} - T_{cl} - (T_{\text{con}} - T_{ce})}{\ln[(T_{\text{con}} - T_{cl})/(T_{\text{con}} - T_{ce})]} \tag{13.11}$$

Then the evaporating temperature T_{ev} can be evaluated as

$$T_{\text{ev}} = \frac{e^B T_{el} - T_{ee}}{e^B - 1} \tag{13.12}$$

where

$$B = \frac{T_{ee} - T_{el}}{Q_{\text{ev}}/(A_{\text{ev}} U_{oe})} \tag{13.13}$$

Similarly, the condensing temperature:

$$T_{\text{con}} = \frac{e^C T_{cl} - T_{ce}}{e^C - 1} \tag{13.14}$$

where

$$C = \frac{T_{cl} - T_{ce}}{Q_{\text{rej}}/(A_{\text{con}} U_{oc})} \tag{13.15}$$

where $A_{\text{ev}}, A_{\text{con}}$ = evaporating and condensing surface area, ft^2 (m^2)
U_{oe}, U_{oc} = overall heat-transfer coefficient based on evaporating and condensing surface area, Btu/h·ft^2·°F (W/m^2·°C)

Example 13.1. A centrifugal chiller using HCFC-123 as the refrigerant has the following operating parameters for its shell-and-tube liquid cooler at design conditions:

Chilled water entering temperature T_{ee}	54°F (12.2°C)
Chilled water leaving temperature T_{el}	44°F (6.7°C)
Evaporating temperature T_{ev}	36°F (2.2°C)
Ratio of outer to inner surfaces A_o/A_i	3.5
Fouling factor on water side	0.00025 h·ft^2·°F/Btu (0.000044 m^2·°C/W)
Water-side heat-transfer coefficient h_i	1700 Btu/h·ft^2·°F (9653 W/m^2·°C)

1. If the boiling coefficient of HCFC-123 can be calculated by $h_b = 2.5\ (Q_{\text{ev}}/A_o)^{0.7}$ and $\eta_f = 1$, calculate the outer evaporating surface area required in the shell-and-tube evaporator for 1 ton (3.5 kW) of refrigeration capacity.

2. Calculate the evaporating temperature for the same evaporator at a load ratio of 0.5.

Solution

1. From Eqs. (13.8), and (13.9) and the given data, the overall heat-transfer coefficient at design condition is

$$U_o = \frac{1}{1/h_b + R_f A_o/A_i + (A_o/A_i)(1/h_i)} = \frac{1}{1/[2.5(Q_{\text{ev}}/A_o)^{0.7}] + 0.00025 \times 3.5 + 3.5/1700}$$

from Eq. (13.11), the log-mean temperature difference at the evaporator is

$$\Delta T_m = \frac{T_{ee} - T_{\text{ev}} - (T_{el} - T_{\text{ev}})}{\ln[(T_{ee} - T_{\text{ev}})/(T_{el} - T_{\text{ev}})]}$$
$$= \frac{54 - 36 - (44 - 36)}{\ln[(54 - 36)/(44 - 36)]} = 12.33°\text{F}\ (6.85°\text{C})$$

Then, from Eq. (13.7),

$$Q_{\text{ev}}/A_o = U_o \Delta T_m = \frac{12.33[2.5(Q_{\text{ev}}/A_o)^{0.7}]}{1 + 0.00734(Q_{\text{ev}}/A_o)^{0.7}}$$

Rearranging, we find

$$0.00734\frac{Q_{\text{ev}}}{A_o} + \left(\frac{Q_{\text{ev}}}{A_o}\right)^{0.3} = 30.83$$

Solving by iteration, we have

$$\frac{Q_{\text{ev}}}{A_o} \approx 2750\ \text{Btu/h} \cdot \text{ft}^2$$

For $Q_{\text{ev}} = 1$ ton $= 12{,}000$ Btu/h, then,

$$A_o = \frac{12{,}000}{2750} = 4.36\ \text{ft}^2\ (0.41\text{m}^2)$$

That is, for each ton of refrigeration, there should be 4.36 ft^2 (0.41 m^2) of outer surface area A_o for the tubes at the evaporator.

2. At 50 percent part-load operation, $Q_{\text{ev}} = 0.5 \times 12{,}000 = 6000$ Btu/h. Then

$$\frac{Q_{\text{ev}}}{A_o} = \frac{6000}{4.36} = 1375\ \text{Btu/h} \cdot \text{ft}^2$$

and

$$h_b = 2.5\left(\frac{Q_{\text{ev}}}{A_o}\right)^{0.7} = 2.5(1375)^{0.7} = 393\ \text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

Also, $U_o = \dfrac{1}{1/393 + 0.00025 \times 3.5 + 3.5/1700} = 182.6\ \text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}\ (1037\ \text{W/m}^2 \cdot \text{C})$

From Eq. (13.7), then, at 50 percent part-load operation

$$\Delta T_m = \frac{Q_{\text{ev}}}{U_o A_o} = \frac{6000}{182.6 \times 4.36} = 7.54°\text{F}\ (4.2°\text{C})$$

Since the chilled water flow rate at the plant loop is constant, then at 50 percent part-load operation $T_{ee} - T_{el} = 0.5(54 - 44) = 5°\text{F}$. Also, from Eq. (13.13),

$$B = \frac{T_{ee} - T_{el}}{Q_{\text{ev}}/(U_o A_o)} = \frac{5}{6000/(182.6 \times 4.36)} = 0.663$$

Therefore, from Eq. (13.12) the evaporating temperature at 50 percent part-load operation is

$$T_{ev} = \frac{e^B T_{el} - T_{ee}}{e^B - 1}$$

$$= \frac{e^{0.663} \times 44 - 49}{e^{0.663} - 1} = 38.69°\text{F}\ (3.72°\text{C})$$

For part-load operation at a load ratio of 0.5, T_{ev} rises $38.69 - 36 = 2.69°\text{F}$ (1.5°C).

Temperature Lift at Part-Load Operation for Centrifugal Chillers

Figure 13.12 is a diagram of temperature lift at various load ratios for a centrifugal chiller using HCFC-123 as refrigerant. The chilled water temperature leaving the chiller is maintained at 44°F (6.7°C) at all load ratios, and the outdoor wet-bulb temperature also remains constant at 75°F (23.9°C) when the load ratio varies from 0 to 1. Fouling factors for both evaporator and condenser are 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot C/W$).

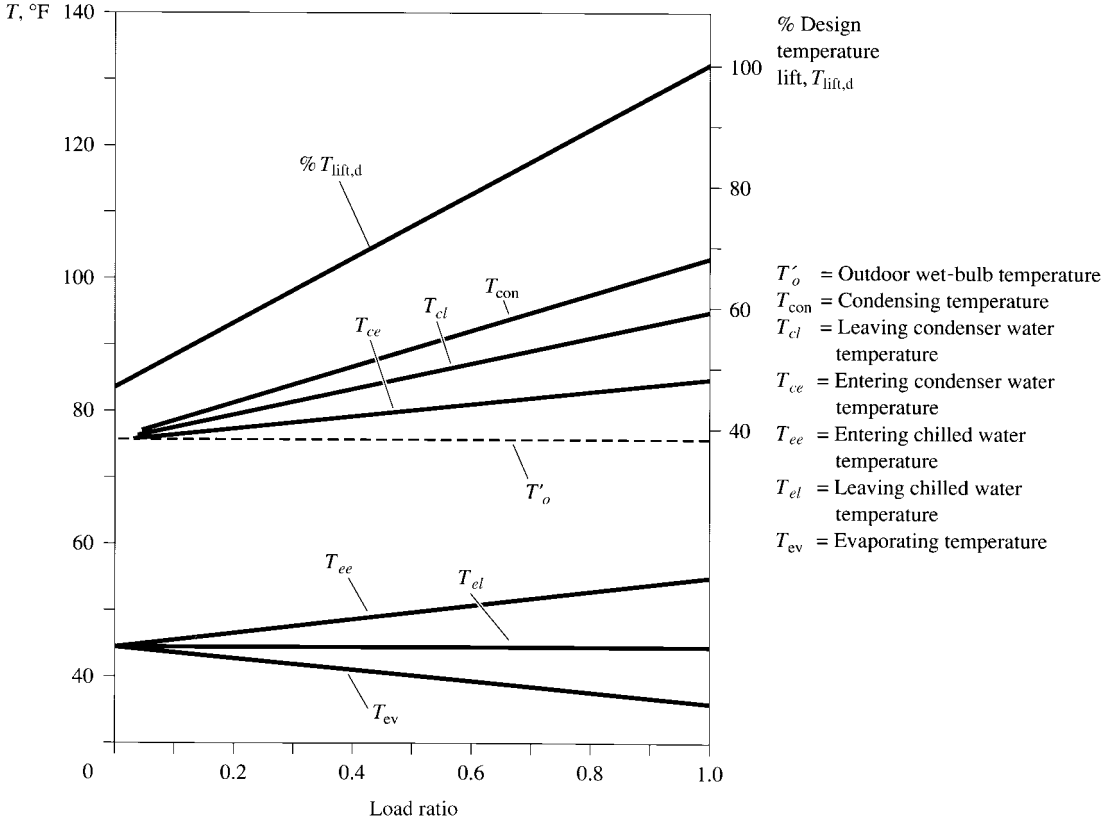


FIGURE 13.12 Temperature lift at various load ratios of a centrifugal chiller.

Under design conditions, if the cooling tower approach is 9°F (5°C), the evaporating temperature T_{ev} is 8°F (4.4°C) lower than T_{el}, the condensing temperature T_{con} is 8°F (4.4°C) higher than T_{cl}, and T_{cl} is 10°F (5.6°C) higher than T_{ce}. Therefore, the condensing temperature is

$$T_{con} = 75 + 9 + 10 + 8 = 102°F\ (38.9°C)$$

and the evaporating temperature is

$$T_{ev} = 44 - 8 = 36°F\ (2.2°C)$$

The temperature lift at design condition is, therefore,

$$\Delta T_{lift,d} = T_{con} - T_{ev} = 102 - 36 = 66°F\ (36.7°C)$$

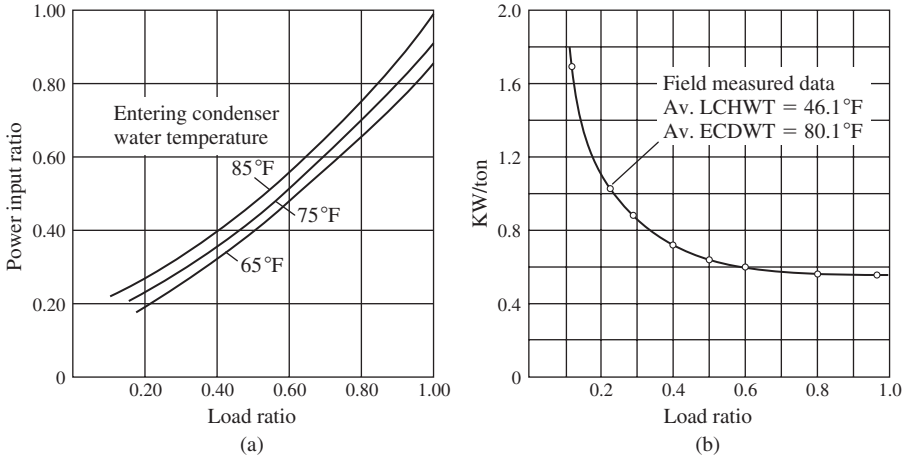
If the load ratio = 0, theoretically T_{con} drops to 75°F (23.9°C), and T_{ev} rises to 44°F (6.7°C). Therefore, the temperature lift ΔT_{lift} drops to

$$\Delta T_{lift} = 75 - 44 = 31°F\ (17.2°C)$$

This drop is about 31/66 = 0.47, or 47 percent, of the design temperature lift $\Delta T_{lift,d}$.

The drop in temperature lift is caused by the following:

- A decrease in condenser water entering temperature T_{ce} because of the smaller value of approach in the cooling tower at reduced load ratios
- A drop in condenser water temperature rise (entering and leaving temperature difference) because of the reduction in total heat rejection
- A decrease in the difference between the condensing temperature and the condenser water leaving temperature $T_{con} - T_{cl}$
- A decrease in the difference between the chilled water leaving temperature and the evaporating temperature $T_{el} - T_{ev}$.



Av. LCHWT = average leaving chilled water temperature
Av. ECDWT = average entering condenser water temperature

FIGURE 13.13 Relationship (*a*) between power input ratio and load ratio and (*b*) between kW/ton and load ratio of a water-cooled centrifugal chiller.

The actual percentage of design power input may be higher or slightly lower than the load ratios, depending on

- Whether the outdoor wet-bulb temperature is constant or drops at part-load operation
- whether there is a T_{el} reset
- the decrease of compression efficiency at part-load operation

Figure 13.13*a* is a plot of the relationship between the power input ratio $P_{in}/P_{in,d}$ and load ratio $Q_{ref}/Q_{ref,d}$ at various condenser water entering temperatures of 65, 75, and 85°F (18.3, 23.9, and 29.4°C). Here, P_{in} and $P_{in,d}$ represent the power input and design power input, in hp (kW). Likewise, Q_{ref} and $Q_{ref,d}$ are the refrigeration load and the design refrigeration load, in Btu/h (W). The lower the load ratio, and the lower the condenser water entering temperature T_{ce}, the lower the power input ratio. For a specific load ratio, power input ratio $P_{in}/P_{in,d}$ is alway less than the load ratio $Q_{ref}/Q_{ref,d}$ except when $T_{ce} \geq 85$°F (29.4°C) and $Q_{ref}/Q_{ref,d} \leq 0.4$.

Figure 13.13*b* is a plot of the chiller efficiency curve, a curve that relates the energy use in kW/ton of the chiller and the load ratio of a water-cooled centrifugal chiller based on Austin (1991). The optimum loading point is at 58 percent load at ARI condenser water conditions, 80 percent load at constant 85°F (29.4°C) condenser water entering temperature. ARI condenser water conditions are based on a 2.5°F (1.4°C) condenser water temperature drop for each 10 percent load decrease starting with 85°F (29.4°C) at 100 percent load. Lower condenser water temperature needs a higher optimum load point.

13.10 FREE REFRIGERATION

Principle of Operation

Free refrigeration means the production of a refrigeration effect without the operation of a compressor. This effect can be achieved because refrigerant tends to migrate to the area of lowest temperature in the system when the compressor ceases to operate.

Air-free refrigeration operation for a single-stage, water-cooled centrifugal chiller is illustrated in Fig. 13.14. When the free refrigeration is turned on, the compressor ceases to operate, the inlet vanes are fully opened, and the valve in the passage that connects the condenser and evaporator is opened.

When the temperature of condenser water from the cooling tower is lower than the temperature of the chilled water leaving the evaporator, the higher saturated pressure of the refrigerant in the evaporator forces the vaporized refrigerant to migrate to the condenser, where the saturated pressure of the refrigerant is lower. The absorption of latent heat of vaporization from the chilled water in the evaporator's tube bundle causes the temperature of chilled water to drop. The condensation of vaporized refrigerant into liquid results in the rejection of the latent heat of condensation to the condenser water through the tube bundle in the condenser, raising the temperature of condenser water a few degrees. Liquid refrigerant drains to the evaporator as a result of gravity. Free refrigeration is thus achieved.

Automatic free refrigeration control can be provided by the DDC system. When the condenser water temperature is a specified number of degrees lower than the required chilled water temperature, the DDC unit controller starts the free refrigeration cycle. The shutoff valve is opened, and an interlocked circuit shuts the compressor off. The free refrigeration cycle lasts as long as the condenser water from the cooling tower is a few degrees higher than the temperature of the chilled water leaving the evaporator, and as long as the free refrigeration capacity meets the refrigeration load required.

For a two-stage centrifugal chiller with a flash cooler, or a three-stage chiller with a two-stage flash cooler, there are more connecting passages and shutoff valves. The principle of operation of free refrigeration is still the same. Once the refrigeration required is greater than the free

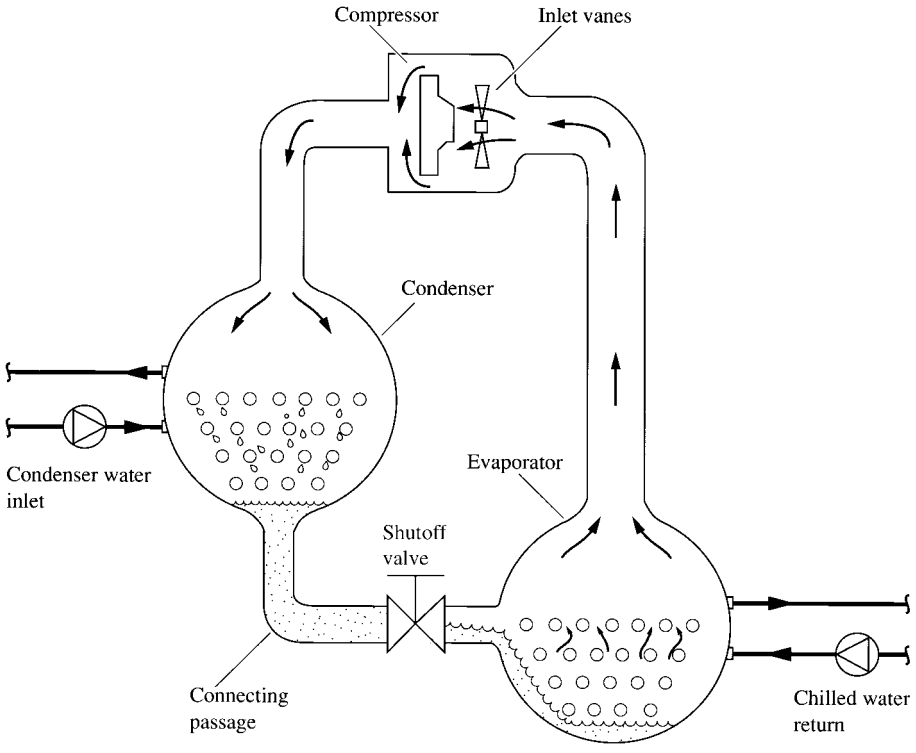


FIGURE 13.14 Free refrigeration operation for a single-stage, water-cooled centrifugal chiller.

refrigeration capacity, as determined from the temperature of the chilled water returned to the evaporator, the DDC unit controller closes the vapor and liquid shutoff valves and starts the compressor. The vaporized refrigerant from the evaporator is extracted by the compressor, discharged to the condenser, and condensed to liquid refrigerant as in normal operation.

Free Refrigeration Capacity

Free refrigeration capacity depends on the following parameters:

- The temperature difference between the chilled water leaving the evaporator and the condenser water entering the condenser.
- The size of the evaporator.
- Condenser water flow.
- Fouling factors at the evaporator and the condenser. An allowable fouling factor is usually 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m \cdot C/W$). A higher fouling factor means a lower free refrigeration capacity.

Free refrigeration capacity is always expressed as a percentage of the base capacity of the centrifugal chiller. For a standard-size evaporator in a typical centrifugal chiller that is running in a free refrigeration cycle, if the chilled water leaves the evaporator at 48°F (8.9°C) and the condenser

water enters the condenser $T_{en,c}$ at 35°F (1.7°C), then the free refrigeration capacity $Q_{ref,f}$ can be 45 percent of the base capacity. When $T_{en,c}$ is 40°F (4.4°C), $Q_{ref,f}$ is 25 percent; and when $T_{en,c}$ is 44°F (6.7°C), $Q_{ref,f}$ may be zero.

Free refrigeration saves energy input to the compressor and can be applied to locations where the condenser water temperature can be at least 5°F (2.8°C) lower than the temperature of chilled water leaving the evaporator. A free refrigeration arrangement is usually optional.

13.11 OPERATION OF MULTIPLE-CHILLER PLANT

Multiple-Chiller Plant

A large centrifugal refrigeration system is often located in a multiple-chiller plant in which two or more centrifugal chillers are installed. Multiple chillers improve the system reliability. On the contrary, too many smaller chillers often lower the efficiency and increase the initial cost.

A multiple-chiller water-cooled plant also comprises cooling towers, chilled water pumps, condenser water pumps, piping, accessories, and controls. Among these, centrifugal compressors, chilled and condenser water pumps, and cooling tower fans are energy users.

According to Bellenger and Becker (1996) and Knebel (1997), the electric power consumption of the chilled water and condenser water pumps varies between 0.1 and 0.2 kW/ton, and the cooling tower fans vary from 0.04 to 0.08 kW/ton depending on the efficiency of pumps and fans as well as the head of the pumps.

Parallel and Series Piping

Multiple chillers can be piped in parallel, as shown in Fig. 13.15*a*. Each chiller usually has its own chilled water pump. If a butterfly valve is installed for each chiller-pump combination, both the cooling capacity and the chilled water flow are turned on and off in sequence and are controlled by the DDC system controller. The flexibility and reliability of operation of parallel chillers are excellent. Chillers can be withdrawn from operation for maintenance without affecting the others. For a

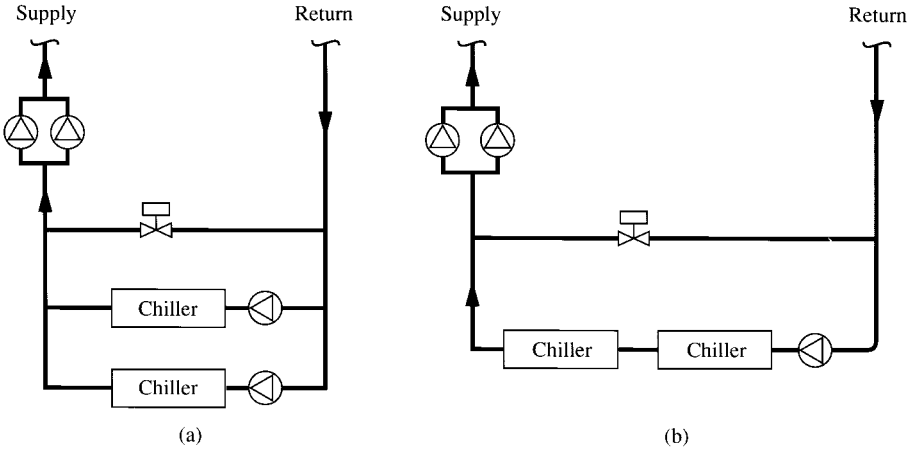


FIGURE 13.15 (*a*) Parallel chillers and (*b*) series chillers.

plant with two or more parallel piped chillers, the chiller that is turned on first is called the leading chiller, and the other chillers are called lag chillers.

When multiple chillers are piped in series, as shown in Fig. 13.15*b*, there are usually only two chillers in series, and they have the following characteristics:

- Chilled water coils are sized for a relatively higher temperature rise and lower water flow rate.
- The pressure drop of chilled water through two series-connected evaporators is additive.
- Control is more complicated in part-load operation.

In practice, multiple-chiller plants are often piped in parallel.

Chiller Staging

In a multiple-chiller plant using a plant-building loop combination, the principle of staging the chillers, on or off, is based on these considerations: (1) The chilled water flow rate in the plant loop is always slightly higher than the building loop; (2) staging on the chiller(s) must meet the requirements of the increased system load; (3) power input to the chillers and chilled water pumps should be minimum.

Cascia (1988) recommended the following for staging the chillers on or off:

- Calculate the required refrigeration load.
- Determine the load ratio of each chiller and the chiller plant.
- Calculate the part-load power consumption for each chiller, and the combination of the chiller and water pumps including chilled water and condenser water pumps. Select the combination with the minimum power consumption.

Austin (1991) recommended in a multiple-chiller plant to start a lag chiller when the leading chiller exceeds its optimum load point. The leading chiller is always the most energy-efficient chiller and the last one to turn off. The last chiller to be turned on is the least energy-efficient one and the first chiller to turn off.

For a multiple-chiller plant:

- A higher compressor efficiency always means a lower kW/ton energy use.
- It is always energy-efficient to operate a single chiller with its chiller pump, instead of two chillers of the same size and their chiller pumps, if a single chiller can meet the required refrigeration load. The saving of the power input to the chilled water pump lowers the energy use.
- If all the multiple chillers are equal in size, have the same chilled water flow rate, and have the same full-load cooling capacity, then when two or more chillers are turned on, each must be loaded approximately equally. This is because the chilled water entering and leaving temperature differential $T_{ee} - T_{el}$ is the same, and each turned-on chiller must have the same refrigeration load ratio R_{load}, which can be calculated as:

$$R_{\text{load}} = \frac{T_{ee} - T_{el}}{T_{ee,d} - T_{el,d}} \tag{13.16}$$

where $T_{ee,d}, T_{el,d}$ = temperature of chilled water entering and leaving the evaporator at design conditions, °F (°C). For a multiple-chiller plant, it is attractive to install chillers of unequal size to meet required low loads when only one chiller is turned on. From Eq. (13.16), when two or more chillers are loaded, the load ratio of unequal-size chillers is still approximately the same. It is important to note that all unequal-size chilled water pumps connected in parallel must have the same head.

Chiller Minimum Performance

ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency requirements for water-cooled electrically operated centrifugal chillers as follows:

Size	Minimum efficiency	Efficiency as of 10/29/2001
<150 tons	3.8 COP, 3.9 IPLV	5.0 COP, 5.0 IPLV
≥ 150 tons and <300 tons	4.2 COP, 4.5 IPLV	5.55 COP, 5.55 IPLV
≥ 300 tons	5.2 COP, 5.3 IPLV	6.1 COP, 6.1 IPLV

Minimum efficiency requirements do not apply for chillers in low-temperature applications where design leaving fluid temperature is $\leq 40°F$ (4.4°C).

Design Considerations

Power Consumption. The following are effective means to reduce the power consumption, in kW/ton (COP), as well as to enhance the minimum performance of centrifugal chillers:

- Raise the temperature of the chilled water leaving the evaporator T_{el} by using a T_{el} reset at reduced loads under the condition that space relative humidity does not exceed 65 percent.
- Lower the condenser water entering temperature T_{ce} as low as the chiller manufacturer permits.
- If the fouling factor on the water side of the condensing surface exceeds 0.00025 $h \cdot ft^2 \cdot °F/Btu$ (0.000044 $m^2 \cdot °C/W$), reduce the fouling factor by using a brush cleaning system or some other means, especially in location where the outdoor air is heavily contaminated.
- Increase the surface area of the evaporator and condenser if they are cost-effective.

Many manufacturers of centrifugal chillers now offer different sizes of evaporators and condensers to match a specific size of compressor. The designer can thus select the centrifugal chiller that meets the designated minimum performance.

Equipment Sizing. Although a single chiller is less costly than multiple chillers, there is a danger of total loss of service when the chiller fails to operate or needs maintenance. Multiple chillers are usually the best choice for a large chiller plant. A plant installed with three chillers ensures a 66 percent of capacity even if one of the chillers fails.

If continuity of service is critical, a standby chiller that has a capacity equal to the largest chiller in the refrigeration plant should be provided. A smaller chiller used during light-load conditions, especially in after-hours operation, in addition to several equally large chillers is often beneficial. The refrigeration load always increases as building usage grows. The possibility of future growth should be considered.

Plant Location and Layout. Generally, the central plant should be located near the system load as well as near to the sources of utilities. For a high-rise building, the ideal location is often the hub of the building, near shopping and public areas. If space at the hub is not available, the chiller plant may be located in the basement, on an intermediate floor, or on the top floor, depending on the type of building and the architectural design.

Equipment should be laid out in an orderly arrangement for efficient piping. Vertical and lateral clearances must be maintained. Equipment may be 10 to 12 ft (3 to 3.6 m) in height. Provision of access platforms and ladders to valves and piping should be considered in the design stage. A code-specified clearance of 3 to 5 ft (0.9 to 1.5 m) from electrical panels and devices must be maintained.

Maintenance and service facilities should also be planned in the design stage. These include removal and replacement of components, stored space, washrooms, and janitorial facilities.

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CHAPTER 14
REFRIGERATION SYSTEMS: ABSORPTION

14.1 ABSORPTION SYSTEMS 14.1
Type of Absorption System 14.1
Historical Development 14.2
Cost Analysis 14.2
Applications 14.3
14.2 PROPERTIES OF AQUEOUS LITHIUM-BROMIDE SOLUTION 14.3
Mass Balance in Solution 14.3
Vapor Pressure 14.3
Equilibrium Chart 14.4
Enthalpy-Concentration Diagram 14.5
14.3 DOUBLE-EFFECT, DIRECT-FIRED ABSORPTION CHILLERS 14.6
System Description 14.6
Air Purge Unit 14.8
Series Flow, Parallel Flow, and Reverse Parallel Flow 14.8
Flow of Solution and Refrigerant 14.9
14.4 PERFORMANCE OF DOUBLE-EFFECT, DIRECT-FIRED ABSORPTION CHILLER 14.11
Mass Flow Rate of Refrigerant and Solution 14.11
Thermal Analysis 14.12
Coefficient of Performance 14.14
14.5 ABSORPTION CHILLER CONTROLS 14.16
Capacity Control and Part-Load Operation 14.16
Crystallization and Controls 14.17
Cooling Water Temperature Control 14.17
Safety and Interlocking Controls 14.18
Monitoring and Diagnostics 14.18
14.6 OPERATING CHARACTERISTICS AND DESIGN CONSIDERATIONS 14.18
Difference between Absorption and Centrifugal Chillers 14.18
Evaporating Temperature 14.19
Cooling Water Entering Temperature 14.19
Heat Removed from Absorber and Condenser 14.19
Condensing Temperature 14.19
Corrosion Control 14.20
Rated Condition of Absorption Chiller 14.20
Minimum Performance 14.20
14.7 ABSORPTION CHILLER-HEATERS 14.20
Heating Cycle 14.21
Actual Performance 14.22
14.8 ABSORPTION HEAT PUMPS 14.22
Functions of Absorption Heat Pump 14.22
Case Study: Series-Connected Absorption Heat Pump 14.22
14.9 ABSORPTION HEAT TRANSFORMER 14.24
System Description 14.24
Operating Characteristics 14.24
Coefficient of Performance 14.26
REFERENCES 14.26

14.1 ABSORPTION SYSTEMS

Type of Absorption System

Absorption systems use heat energy to produce refrigeration or heating and sometimes to elevate the temperature of the waste heat. Aqueous lithium bromide (LiBr) is often used to absorb the refrigerant, the water vapor, and provides a higher coefficient of performance.

Current absorption systems can be divided into the following categories:

- Absorption chillers use heat energy to provide refrigeration.
- Absorption chiller-heaters use heat energy to provide cooling or heating.

- Absorption heat pump extracts heat energy from the evaporator through the absorber, adds to the heat input in the generator, and releases them both to the hot water in the condenser for heating.
- Absorption heat transformers elevate the temperature of the waste heat to a value higher than any other input fluid stream supplied to the absorption heat transformer.

Historical Development

In the 1950s and 1960s, both centrifugal chillers driven by electric motors and absorption chillers using steam as heat input to provide summer cooling were widely used in central refrigeration plants. Steam was widely used because excess steam was available in summer in many central plants that used steam to provide winter heating, and because energy costs were of little concern.

After the energy crisis in 1973, the price of natural gas and oil used to fuel steam boilers drastically increased. The earliest single-stage, indirect-fired steam absorption chillers had a coefficient of performance (COP) of only 0.6 to 0.7. They required more energy and could not compete with electric centrifugal chillers. Many absorption chillers were replaced by centrifugal chillers in the late 1970s and 1980s.

Because of the high investment required to build new power plants, electric utility companies added high-demand charges and raised cost-per-unit charges during peak usage periods. In recent years, double-effect, direct-fired absorption chillers have been developed in both Japan and the United States with a COP approximately equal to 1.

Cost Analysis

A cost analysis is often required. Aumann (1996) recommended that in addition to the chiller itself, the energy and the initial costs of the auxiliaries, such as condenser pumps, cooling tower, and tower fans, be included in the cost analysis because of the higher heat rejection in the absorption chiller. The auxiliary energy costs can be 30 percent higher for absorption chillers than for electric.

Sun (1991) compared two designs for a project with a 1200-ton (4220-kW) design refrigeration load. One design is three 400-ton (1406-kW) electric centrifugal chillers, and the other is two electric centrifugal chillers plus one absorption chiller.

The cost of gas for this project is \$0.46 per therm in summer, from April through November, and \$0.50 per therm in winter. One therm is equal to 100,000 Btu/h (29,300 W) of heat energy input. The cost of electricity is \$0.126 per kWh during on-peak periods and \$0.0433 per kW/h for the off-peak periods. The on-peak period is the 8-h weekday daytime determined by the electric utility. The additional electricity demand charge is \$3.60 per kW.

The centrifugal chiller uses 0.7 kW/ton (5.02 COP), and the additional energy use for the auxiliaries for the centrifugal chiller is about 0.1 kW/ton higher than that for the absorption chiller. The double-effect, direct-fired absorption chiller uses 0.12 therm/ton. The comparison cost (in dollars) of these two alternatives for the third 400-ton (1406-kW) chiller operated 8 h/day, 5 days/wk, 8 months for summer and 4 months for winter is as follows:

	Absorption chiller	Centrifugal chiller
Initial cost		
Chiller	200,000	80,000
Cooling tower	53,000	48,000
Demand	2126	11,491
Energy		
Electricity		
Summer	7200	36,700
Winter	3598	18,337

Gas		
Summer	22, 969	
Winter	12,474	
Total energy cost before tax	$48,417	$66,528
Total energy cost after tax	$54,469	$74,844

The absorption chiller has an annual energy cost saving of $20,375 and a simple payback period of about 6 years.

Applications

According to the *Air Conditioning, Heating and Refrigeration News*, April 14, 1997, the shipments of new absorption chillers from the manufacturers in the United States in 1996 numbered 579, whereas the shipments of centrifugal and screw chillers in 1996 numbered about 9200. Absorption chillers had a share of about 5 percent of the new large chiller market in the United States in the mid-1900s.

Absorption chillers have the advantage of using gas and are therefore not affected by the high electric demand charge and high unit rate at on-peak hours. Most absorption chillers use water as refrigerant, and its ozone depletion potential is zero. Absorption chillers are advantageous to combine with electric chillers so that electric chillers will undertake the base load and the absorption chillers handle the load at on-peak hours. Although absorption chillers have a higher initial cost than centrifugal compressors, in many locations in the United States where the cost ratio of electricity to natural gas is favorable, installation of absorption chiller for use during on-peak hours or even normal operating hours is sometimes economically beneficial. Absorption chillers may have a simple payback of several years.

14.2 PROPERTIES OF AQUEOUS LITHIUM-BROMIDE SOLUTION

Mass Balance in Solution

The composition of a solution is generally expressed by the mass fraction of its components. In a solution containing lithium bromide (LiBr) and water, X is used to indicate the mass fraction of lithium bromide, i.e.,

$$X = \frac{m_l}{m_l + m_w} \tag{14.1}$$

where m_l = mass of lithium bromide in solution, lb (kg)
m_w = mass of water in solution, lb (kg)

The mass fraction of water in solution is $1 - X$. If the mass fraction of LiBr in a solution is expressed as a percentage, it is known as the concentration of LiBr.

Vapor Pressure

In general, the total pressure of a solution is equal to the sum of the vapor pressures of the solute and of the solvent. However, in the case of a lithium bromide–water solution, the vapor pressure of pure LiBr can be ignored because its value is much smaller than that of water.

When LiBr is dissolved in water, the boiling point of the solution at a given pressure is raised. However, if the temperature of the solution remains constant, the dissolved LiBr reduces the vapor pressure of the solution.

When a solution is saturated, equilibrium is established. The number of molecules across the interface from liquid to vapor per unit time is equal to the number of molecules from vapor into liquid. If the number of liquid molecules per unit volume is reduced due to the presence of a solute then the number of vapor molecules per unit volume is also reduced. Consequently, the vapor pressure of the solution is decreased.

Equilibrium Chart

The properties of an aqueous lithium bromide solution, including vapor pressure, temperature, and the mass fraction at equilibrium, may be illustrated on an equilibrium chart based on the Dühring plot, as shown in Fig. 14.1. The ordinate of the equilibrium chart is the saturated vapor pressure of water in log-scale millimeters of mercury absolute (mm Hg abs) and the corresponding saturated temperature (°F). The scale is plotted on an inclined line. The abscissa of the chart is the temperature of the solution (°F). Mass fraction or concentration lines are inclined lines and are not parallel to each other.

At the bottom of the concentration lines, there is a crystallization line or saturation line. If the temperature of a solution of constant mass fraction of LiBr drops below this line—or if the mass fraction of LiBr of a solution of constant temperature is higher than the saturated condition—the part of LiBr salt exceeding the saturated condition tends to form solid crystals.

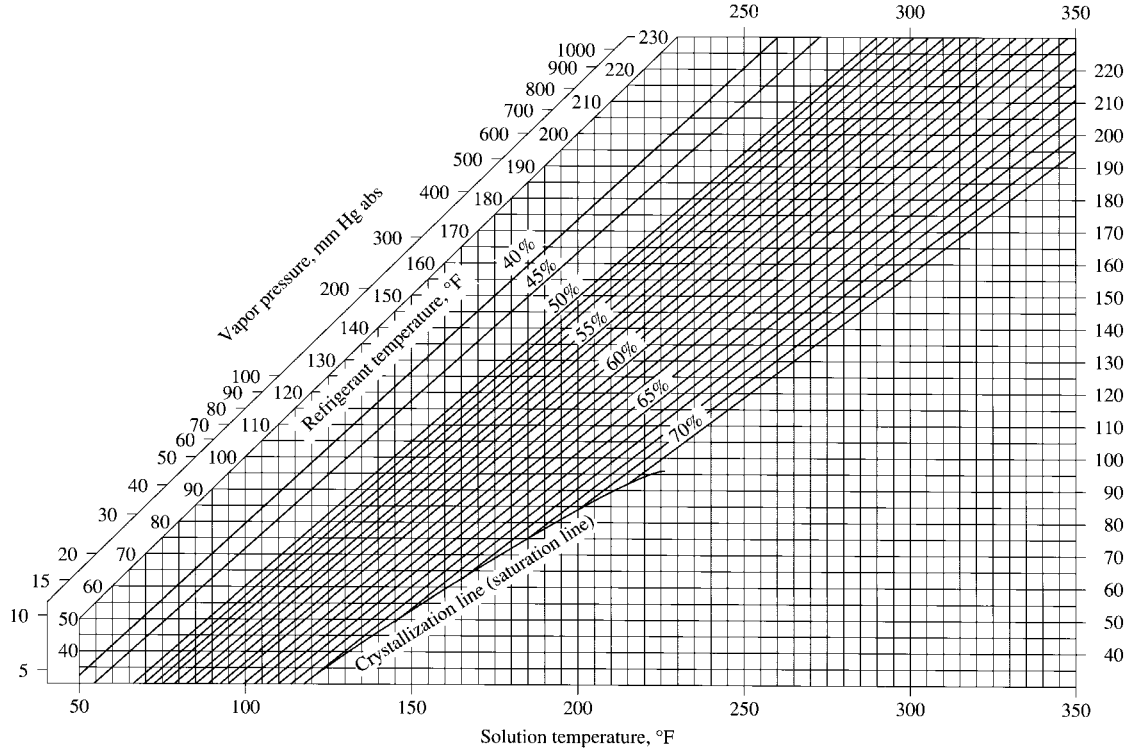


FIGURE 14.1 Equilibrium chart for aqueous lithium-bromide (LiBr) solution. (*Source: Carrier Corporation. Reprinted with permission.*)

Enthalpy-Concentration Diagram

When water is mixed with anhydrous lithium bromide at the same temperature to form a solution adiabatically, there is a significant increase in the temperature of the solution. If the mixing process is to be an isothermal process, i.e., if the temperature of the process is to be kept constant, then heat must be removed from the solution. Such a heat transfer per unit mass of solution is called the *integral heat of the solution* Δh_i, or *heat of absorption*, in Btu/lb (kJ/kg). Based on the common rule of thermodynamics, Δh_i is negative.

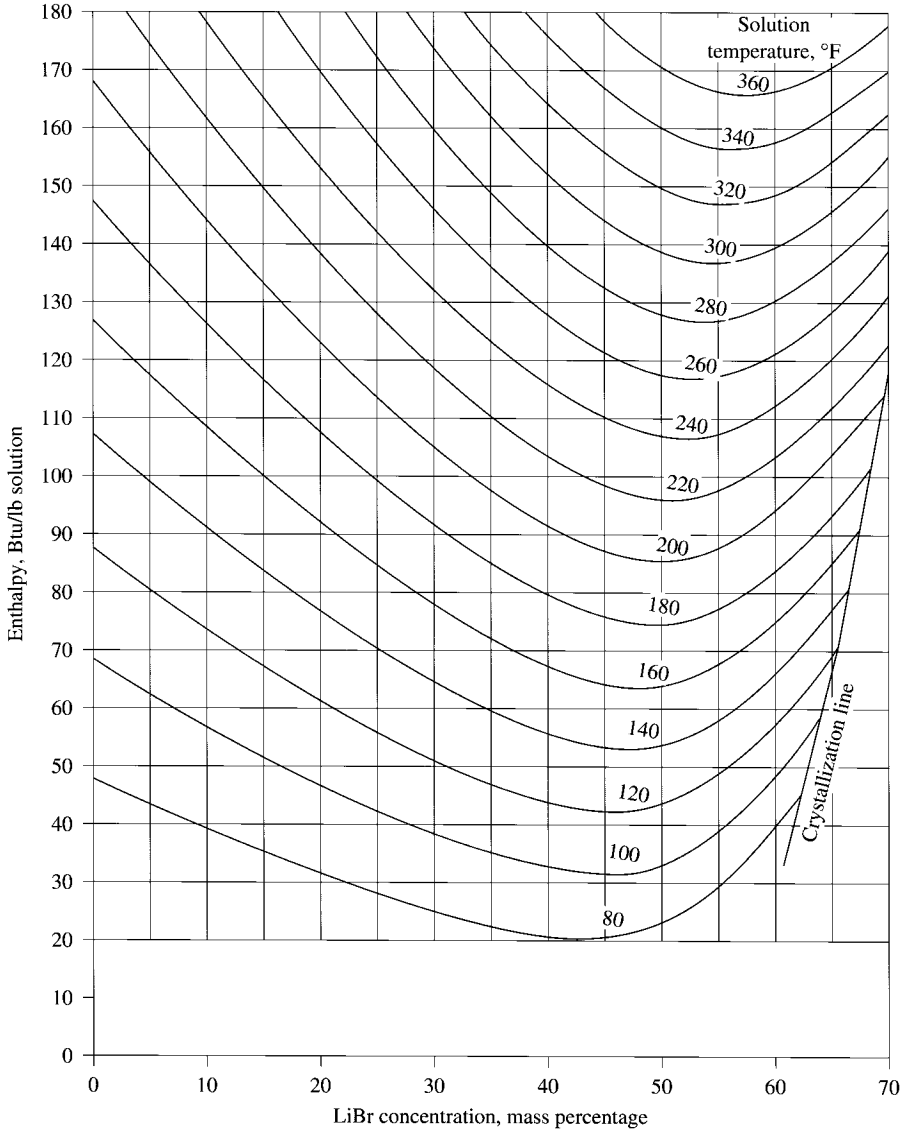


FIGURE 14.2 Enthalpy-concentration diagram for aqueous LiBr solution. (*Source: ASHRAE Handbook 1989, Fundamentals*, *Reprinted with permission.*)

If an aqueous LiBr solution is formed by an isothermal process, its specific enthalpy h, in Btu/lb (kJ/kg), may be expressed as

$$h = (1 - X)h_w + Xh_l + \Delta h_i \tag{14.2}$$

where h_w, h_l = specific enthalpy of water and LiBr, respectively, Btu/lb (kJ/kg). Figure 14.2 shows the enthalpy-concentration h-X diagram of an aqueous LiBr solution. The ordinate of the diagram is the specific enthalpy of the solution h, in Btu/lb, (kJ/kg), and the abscissa is the percent concentration of the aqueous LiBr solution. The curved lines represent the specific enthalpy of the solution at various concentrations of LiBr mixed with water at a constant temperature. At the right-hand side of the diagram, there is also a crystallization line. To the right of this line, excess solid LiBr may crystallize.

14.3 DOUBLE-EFFECT, DIRECT-FIRED ABSORPTION CHILLERS

System Description

In an absorption chiller, water is the refrigerant, and aqueous lithium bromide (LiBr) is the absorbent. Double-effect, direct-fired absorption chillers are most commonly used today because of their higher coefficient of performance. Double-effect means that there are two generators. Direct-fired means that the heat input is supplied to a directly fired generator. Figure 14.3 is a schematic diagram of a double-effect, direct-fired absorption chiller.

A double-effect, direct-fired absorption chiller consists of 12 main components: an evaporator, absorber, high-temperature generator, low-temperature generator, condenser, vapor-liquid separator, high-temperature heat exchanger, low-temperature heat exchanger, high-temperature solution pump, low-temperature solution pump, refrigerant pump, and refrigerant throttling device.

Evaporator and Refrigerant Pump. An evaporator is composed of a tube bundle, an outer shell, spraying nozzles, and a water trough. The chilled water flows inside the tubes. A refrigerant pump sprays the liquid refrigerant through the spray nozzles over the outer surface of the tube bundle to increase the rate of evaporation. A water trough is located at the bottom of the evaporator to maintain a certain liquid level for circulation. A high vacuum, typically at a saturated vapor pressure of 6.8 mm Hg abs. (906 Pa abs.), is maintained to provide an evaporating temperature of about 42°F (5.6°C).

Absorber and Solution Pumps. Absorbers are water-cooled tube bundles with cooling water flowing inside them. In the absorber, the vaporized refrigerant, the water vapor, is extracted and absorbed by the concentrated LiBr solution sprayed from the nozzles so that at high vacuum, 5 mm Hg abs. (666 Pa abs.) can be maintained in the absorber. The low-temperature solution pump raises the pressure of dilute solution from 5 mm Hg abs. (666 Pa abs.) to about 45 to 60 mm Hg abs. (6000 to 8000 Pa abs.) in the low-temperature generator; and the high-temperature solution pump raises the pressure of the partly concentrated solution from about 50 to 350 to 450 mm Hg abs. (6.7 to 46.7 to 60 kPa abs.). This LiBr solution pressure is sufficient to overcome the pressure drop in the heat exchangers, pipes, and pipe fittings.

Heat Exchangers. Heat exchangers are sometimes called economizers. Shell-and-tube or plate-and-frame heat exchangers are most commonly used for greater effectiveness. There are two heat exchangers: a low-temperature heat exchanger and a high-temperature heat exchanger. In the low-temperature heat exchanger, heat is transferred from the hot solution mixture discharged from the heat exchangers to the cold diluted solution pumped from the absorber. In the high-temperature heat exchanger, heat is transferred from the hot concentrated solution discharged from the high-

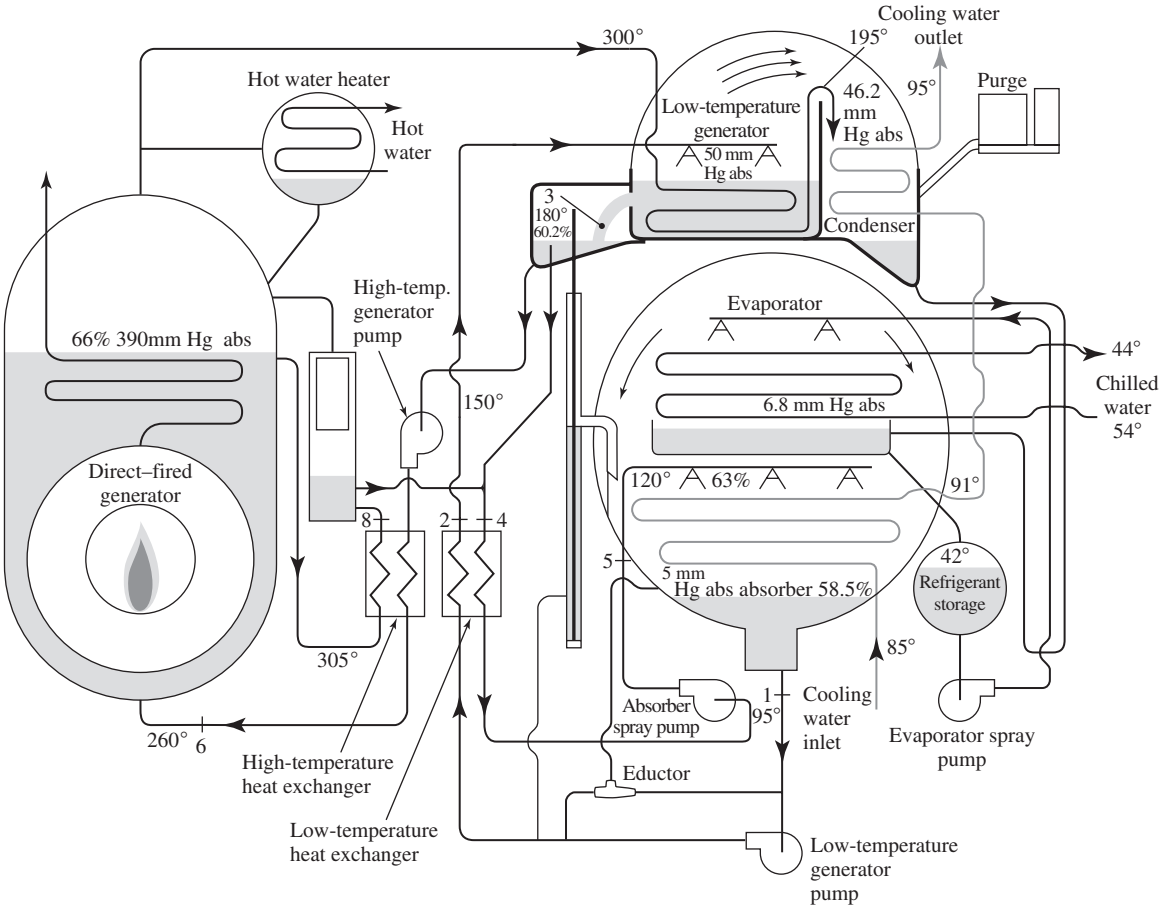


FIGURE 14.3 A double-effect, direct-fired, reverse parallel flow absorption chiller. (*Source: ASHRAE Handbook 1998, Refrigeration. Reprinted with permission.*)

temperature generator to the partly concentrated solution from the low-temperature generator. Both heat exchangers conserve heat energy input and increase the coefficient of performance of the absorption chiller.

Generators. In the high-temperature generator, also called a direct-fired generator, heat is supplied from the direct-fired burner to the diluted solution, and then the water is vaporized from the solution. Vapor refrigerant flows to the low-temperature generator and condenses to liquid there. The latent heat released during condensation is then used to heat the diluted solution. Water is again vaporized at the low-temperature generator. Many types of fuel can be used in the direct-fired burner, such as natural gas, oil, or propane. Natural gas is the most widely used. In the high-temperature generator, there is a vapor-liquid separator in which water vapor is boiled off from the partly concentrated solution.

Instead of heat input from a direct-fired burner, absorption chillers can use other heat sources such as industrial waste heat or exhaust heat from engines or gas turbines.

Condenser. A condenser is a water-cooled tube bundle in which cooling water flows. In the condenser, water is often supplied at a temperature of 80 to 90°F (26.7 to 32.2°C). Water vapor boiled off from the low-temperature generator is condensed to liquid in the condenser, and the liquid water condensed inside the tubes of the low-temperature generator is cooled to a lower temperature in the condenser.

Throttling Devices. Orifices, traps, and valves are often used as throttling devices to reduce the pressure of refrigerant and solution. An orifice plate on the liquid refrigerant line acts as an expansion valve between the condenser and evaporator.

A single-effect absorption chiller has only one generator, while a double-effect absorption chiller uses part of the latent heat of condensation of the vapor refrigerant in the low-temperature generator to enhance the COP. If steam is used as the input heat source in the indirect-fired absorption chiller, the combined efficiency of the boiler, loss in steam transport, and the heat transfer at the high-temperature generator is significantly lower than that of the direct-fired furnace.

Air Purge Unit

Because the pressures in the evaporator, absorber, and condenser are far below atmospheric pressure, air and other noncondensable gases leak into an LiBr absorption refrigeration system. An air purge unit should be installed to remove these noncondensable gases and to maintain the required pressure, temperature, and concentration in the absorption chiller.

Figure 14.4 shows a typical purge unit in an LiBr absorption refrigeration system. Such a unit usually consists of a purge chamber, a pickup tube, a vacuum pump, a solenoid valve, and a manual shutoff valve. When noncondensable gases leak into the system during operation, they tend to migrate to the absorber where the pressure is lowest. Through the pickup tube, noncondensable gases and a certain amount of water vapor flow to the purge chamber from the absorber. Water vapor is absorbed by the LiBr solution sprayed over the tube, and the heat of absorption is removed by the cooling water inside a bank of tubes. The solution absorbed with water vapor is returned to the absorber through a liquid trap. Noncondensable gases are evacuated from the chamber periodically by a vacuum pump which raises the gas pressure and discharges the gases to the outdoor atmosphere.

In addition to the air purge unit, palladium cells are required to continuously remove the small amount of hydrogen gas produced during corrosion because thin membranes of heated palladium are only permeable by the hydrogen gas.

Series Flow, Parallel Flow, and Reverse Parallel Flow

In a double-effect, direct-fired absorption chiller, because there are two generators, the flow of the solution from the absorber to generators can be in series flow, parallel flow, or reverse parallel flow.

Series Flow. The solution from the absorber flows in sequence through the low-temperature heat exchanger, high-temperature heat exchanger, high-temperature generator, high-temperature heat exchanger, low-temperature generator, and low-temperature heat exchanger and returns to the absorber.

Parallel Flow. The solution from the absorber is pumped both to the high-temperature generator through the high-temperature heat exchanger and to the low-temperature generator through the low-temperature heat exchanger separately, and then to the low-temperature generator. The concentrated solution from both generators flows through corresponding heat exchangers. They are recombined and returned to the absorber.

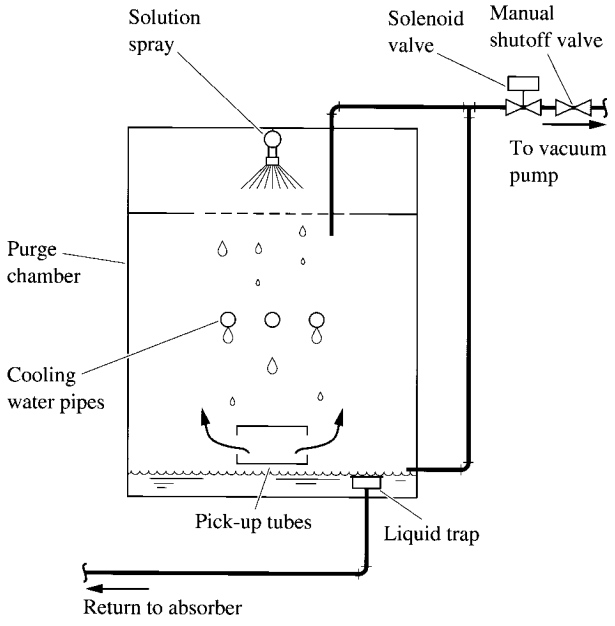


FIGURE 14.4 Purge unit in a lithium bromide absorption chiller. (*Source: ASHRAE Handbook 1998, Refrigeration. Reprinted with permission.*)

Reverse Parallel Flow. As shown in Fig. 14.3, the solution from the absorber is pumped through a low-temperature heat exchanger and flows into a low-temperature generator. After that, the solution splits into two streams. One flows in sequence through the high-temperature heat exchanger, high-temperature generator, and high-temperature heat exchanger again, and then combines with the other stream discharged from the low-temperature generator. The combined stream flows through the low-temperature heat exchanger and returns to the absorber. A reverse parallel flow system is one of the most energy-efficient operating cycles today. It also has a more compact design.

Flow of Solution and Refrigerant

For a double-effect, direct-fired, reverse parallel flow absorption chiller, the absorption cycle shows the properties of the solution at various concentration, temperatures, and pressures, as seen in Fig. 14.5. Sometimes the refrigerant, water vapor, is combined with the solution; and sometimes the refrigerant and solution are separated.

Three kinds of LiBr solution are used during operation: concentrated solution, partly concentrated solution, and diluted solution. Concentrated solution has a higher concentration of LiBr and is created after all the refrigerant has been boiled off from the solution. Concentrated solution is used to absorb the water vaporized in the evaporator. The solution that leaves the low-temperature generator is partly concentrated. Diluted solution has a lower concentration of LiBr and is the combination of aqueous LiBr solution and absorbed water vapor.

An absorption refrigeration system can also be divided into three regions according to the pressure of the refrigerant and solution. The high-pressure region includes the high-temperature generator; the intermediate-pressure region comprises the low-temperature generator and condensers; and the low-pressure region includes the absorber and evaporator. Because the evaporator and absorber as well as the low-temperature generator and the condenser are both connected by vapor passages

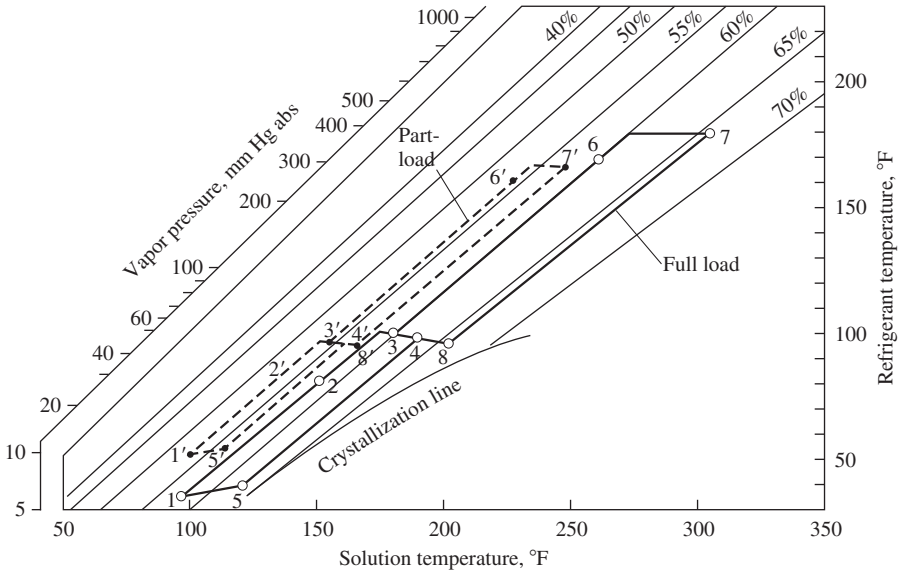


FIGURE 14.5 Absorption cycle of a double-effect, direct-fired, reverse parallel flow absorption chiller.

of adequate size, the pressure difference between the evaporator and absorber as well as between the low-temperature generator and condenser is small.

In a typical double-effect, direct-fired, reverse parallel flow absorption chiller operating at design load, as shown in Fig. 14.3, water usually evaporates at about 42°F (5.6°C) and at a saturated pressure of 6.8 mm Hg abs. (906 Pa abs.) in the evaporator, point 1. Chilled water enters the evaporator inside the tube bundle at 54°F (12.2°C) and leaves at 44°F (6.7°C). Heat is transferred from the chilled water to the vaporized refrigerant at an amount equal to the latent heat of vaporization.

Vaporized water in the evaporator is then extracted to the absorber because of its lower vapor pressure and is absorbed by the concentrated LiBr solution. In the absorber, concentrated solution is supplied and sprayed at about 120°F (48.9°C) with a concentration of 63 percent, point 5. It is cooled by the cooling water flowing inside the tube bundle at an entering temperature of 85°F (29.4°C). As the water vapor from the evaporator is absorbed, the solution is diluted to a concentration of about 58.5% and its temperature drops to 95°F (35°C). The vapor pressure of the solution is then about 5 mm Hg abs. (666 Pa abs.), which is lower than the evaporating pressure of 6.8 mm Hg abs. (906 Pa abs.). The heat of absorption is removed by the cooling water. Diluted solution is then pumped to the low-temperature heat exchanger and the low-temperature generator by a low-temperature solution pump.

In the low-temperature heat exchanger, diluted solution enters at 95°F (35°C) and leaves at about 150°F (65.6°C). After the diluted solution enters the low-temperature generator, because of the release of the latent heat of condensation from the condensed water vapor that flows inside the tube bundle of the low-temperature generator, the diluted solution is first heated to a temperature of 180°F (82.2°C), and the water vapor then boils off at a pressure of about 48.5 mm Hg abs. (6.5 kPa abs.), point 3. As the water vapor is boiled off, the LiBr solution becomes partly concentrated with a concentration of 60.2%.

At the outlet of the low-temperature generator, partly concentrated LiBr solution is divided into two streams: One flows through the high-temperature heat exchanger and then enters the high-

temperature generator through a high-temperature solution pump. The other stream combines with the concentrated solution from the high-temperature heat exchanger and enters the low-temperature heat exchanger, point 4.

In the high-temperature heat exchanger, partly concentrated solution enters at 180°F (82.2°C) and leaves at about 260°F (126.7°C). After that, partly concentrated solution flows into the high-temperature generator. In the high-temperature generator, the solution is heated to 305°F (152°C), and the water vapor boils off at a pressure of about 390 Hg mm abs. (52 kPa abs.), point 7. As the water vapor is boiled off, the LiBr solution is concentrated to 66% in the high-temperature generator. The concentrated solution returns to the high-temperature heat exchanger and cools to about 203°F (95°C) as it leaves the heat exchanger.

The other stream of partly concentrated LiBr solution combines with the concentrated solution from the high-temperature heat exchanger at 203°F (95°C), and the mixture enters the low-temperature heat exchanger at 192°F (89°C) and a concentration of 63.0%. After it flows through the low-temperature heat exchanger, the temperature of the solution drops to about 120°F (49°C). The solution then enters the absorber, sprays over the cooling water tube bundle, and absorbs the water vapor from the evaporator.

As the water vapor that boils off from the LiBr solution in the high-temperature generator at 300°F (149°C) flows through the tubes submerged in the diluted solution in the low-temperature generator, the water vapor condenses to liquid form. Latent heat of condensation released from the liquid refrigerant heats the diluted solution from 150 to 180°F (65 to 82°C) and boils off the water vapor from the diluted solution at a saturated pressure of about 50 mm Hg abs. (6.7 kPa abs.). Finally, in the low-temperature generator, the partly concentrated solution reaches a concentration of 60% and splits into two streams when it leaves at the outlet.

The condensed liquid refrigerant that flows inside the tubes in the low-temperature generator enters the trough of the condenser at a temperature of around 195°F (90°C). Water vapor boiled off in the low-temperature generator is extracted by the condenser's low saturated pressure of 46.2 mm Hg abs. (6.2 kPa abs.) through the top passage and condenses to liquid as the cooling water removes the latent heat of condensation at a temperature of about 98°F (37°C). In the condenser, condensed liquid refrigerant from the tubes is cooled to a temperature of 98°F (37°C) and combined with the liquid refrigerant condensed in the condenser. Liquid refrigerant is forced through an orifice, throttled to a pressure of about 6.8 mm Hg abs. (906 Pa abs.), and returned to the evaporator.

14.4 PERFORMANCE OF DOUBLE-EFFECT, DIRECT-FIRED ABSORPTION CHILLER

Mass Flow Rate of Refrigerant and Solution

For a throttling process, the enthalpy of water entering the evaporator h_{le} is equal to the enthalpy of saturated liquid leaving the condenser h_{lc}, both in Btu/lb (kJ/kg). The mass flow rate of refrigerant required for each 1 ton or 12,000 Btu/h (kW) of refrigeration Q_{ref} produced in the absorption chiller $\dot{m}_r$, in lb/h·ton$_{\text{ref}}$ (kg/s·kW$_{\text{ref}}$), is calculated as

$$\dot{m}_r = \frac{Q_{\text{ref}}}{h_{ve} - h_{lc}} = \frac{12{,}000}{h_{ve} - h_{lc}} \tag{14.3}$$

where h_{ve} = enthalpy of saturated water vapor leaving evaporator, Btu/lb (kJ/kg). According to the mass balance of LiBr in the diluted and concentrated solution,

$$\begin{aligned} X_1\dot{m}_{s1} &= X_5 m_{s5} = X_3(\dot{m}_{s7} + \dot{m}_{s9}) \\ \dot{m}_{s1} &= \dot{m}_{s5} + \dot{m}_r = \dot{m}_{s5} + \dot{m}_{rl} + \dot{m}_{rh} \end{aligned} \tag{14.4}$$

Also,

$$\dot{m}_{s5} = \frac{X_1 \dot{m}_r}{X_5 - X_1}$$

$$C_{\text{cir}} = \frac{X_1}{X_5 - X_1} \tag{14.5}$$

where X_1, X_5 = mass fraction of LiBr in diluted and concentrated solution leaving and entering absorber
$\dot{m}_{s1}$ = mass flow rate of diluted solution per ton of refrigeration, $\text{lb/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kg/s} \cdot \text{kW}_{\text{ref}}$)
$\dot{m}_{s5}$ = mass flow rate of concentrated solution entering absorber per ton of refrigeration, $\text{lb/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kg/s} \cdot \text{kW}_{\text{ref}}$)
$\dot{m}_{s7}, \dot{m}_{s9}$ = mass flow rate of partly concentrated solution per ton of refrigeration at points 7 and 9, respectively, $\text{lb/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kg/s} \cdot \text{kW}_{\text{ref}}$)
$\dot{m}_{rl}, \dot{m}_{rh}$ = mass flow rate of refrigerant boiled off from low-temperature generator and high-temperature generator per ton of refrigeration, $\text{lb/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kg/s} \cdot \text{kW}_{\text{ref}}$)
C_{cir} = circulating factor

Thermal Analysis

To help determine the status of the solution in the absorption cycle of a double-effect, direct-fired, reverse parallel flow absorption chiller as shown in Figs. 14.3 and 14.5, the state points of the cycle are defined as follows:

1 = Mean condition of the diluted solution leaving the absorber
2 = Dilute solution leaving the low-temperature heat exchanger
3 = Partly concentrated solution leaving the low-temperature generator
4 = The condition of the mixture of partly concentrated and concentrated solutions entering the low-temperature heat exchanger
5 = Concentrated solution entering the absorber
6 = Partly concentrated solution leaving high-temperature heat exchanger
7 = Concentrated solution leaving high-temperature generator
8 = Concentrated solution leaving high-temperature heat exchanger
9 = Partly concentrated solution leaving low-temperature generator

When the absorption chiller operates at steady-state condition, the heat balance analyses for the main system components, in $\text{Btu/h} \cdot \text{ton}$, are shown below:
In the evaporator, for 1 ton of refrigeration load (kW) Q_{rl}, in $\text{Btu/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kW/kW}_{\text{ref}}$),

Heat of leaving water vapor	−	heat of entering liquid	=	refrigeration load
$\dot{m}_r h_{ve}$	−	$\dot{m}_r h_{lc}$	=	$\dot{m}_r(h_{ve} - h_{lc}) = Q_{rl}$

(14.6)

In the absorber, the heat of entering water vapor is equal to the heat of leaving water vapor in the evaporator:

Heat of entering water vapor	+	heat of entering solution	−	heat of leaving solution	=	heat to be removed from absorber
$\dot{m}_r h_{ve}$	+	$\dot{m}_{s5} h_5$	−	$\dot{m}_{s1} h_1$	=	Q_{ab}

(14.7)

where h_1, h_5 = enthalpy of solution leaving and entering absorber, Btu/lb (kJ/kg)
Q_{ab} = heat to be removed from absorber, $\text{Btu/h} \cdot \text{ton}_{\text{ref}}$ ($\text{kW/kW}_{\text{ref}}$)

In the high-temperature generator,

Heat of leaving water vapor	+	heat of leaving solution	−	heat of entering solution	=	heat to be supplied to high-temperature generator

$$\dot{m}_{rh}h_{vhg} + \dot{m}_{s7}h_7 - \dot{m}_{s6}h_6 = Q_{hg} \tag{14.8}$$

where $\dot{m}_{rh}$ = mass flow rate of water vapor boiled off in high-temperature generator per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
h_{vhg} = enthalpy of superheated water vapor leaving high-temperature generator, Btu/lb (kJ/kg)
$\dot{m}_{s7}$ = mass flow rate of concentrated solution leaving high-temperature generator per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
$\dot{m}_{s6}$ = mass flow rate of partly concentrated solution entering high-temperature generator per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
h_6, h_7 = enthalpy of solution entering and leaving high-temperature generator, Btu/lb (kJ/kg)
Q_{hg} = heat input to high-temperature generator per ton of refrigeration, $\text{Btu/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kW/kW}_{\text{ref}}$)

In the low-temperature generator,

Heat of entering water vapor	−	heat of leaving water	=	heat of leaving solution	−	heat of entering solution	+	heat to water vapor boiled off in low-temperature generator

$$\dot{m}_{rhg}(h_{vhg} - h_{llg}) = (\dot{m}_{s6} + \dot{m}_{s9})h_3 - \dot{m}_{s2}h_2 + \dot{m}_{rlg}h_{fg} \tag{14.9}$$

where h_{llg} = enthalpy of liquid water leaving low-temperature generator, Btu/lb (kJ/kg)
$\dot{m}_{s6}$ = mass flow rate of partly concentrated solution leaving low-temperature generator to combine with concentrated solution per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
$\dot{m}_{s9}$ = mass flow rate of partly concentrated solution leaving low-temperature generator toward high-temperature heat exchanger per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
$\dot{m}_{s2}$ = mass flow rate of diluted solution entering low-temperature generator per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)
h_2, h_3 = enthalpy of solution entering and leaving low-temperature generator, Btu/lb (kJ/kg)
h_{fg} = latent heat of vaporization in low-temperature generator, Btu/lb (kJ/kg)
$\dot{m}_{rlg}$ = mass flow rate of water vapor boiled off in low-temperature generator per ton of refrigeration, $\text{lb/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kg/s}\cdot\text{kW}_{\text{ref}}$)

And

$$\begin{aligned} \dot{m}_r &= \dot{m}_{rlg} + \dot{m}_{rhg} \\ \dot{m}_{s3} &= \dot{m}_{s6} + \dot{m}_{s9} \end{aligned} \tag{14.10}$$

In the condenser,

Heat of entering liquid water	+	heat of entering water vapor	−	heat of leaving liquid water	=	heat to be removed from condenser

$$\dot{m}_{rhg}h_{lhg} + \dot{m}_{rlg}h_{vlg} - \dot{m}_r h_{lc} = Q_{\text{con}} \tag{14.11}$$

where h_{vlg} = enthalpy of water vapor leaving low-temperature generator, Btu/lb (kJ/kg)
Q_{con} = heat to be removed from condenser, $\text{Btu/h}\cdot\text{ton}_{\text{ref}}$ ($\text{kW/kW}_{\text{ref}}$)

In the high-temperature heat exchanger, if the difference of specific heat of solution is ignored, the effectiveness ε_h is given as

$$\varepsilon_h = \frac{\dot{m}_{s7}(h_7 - h_8)}{\dot{m}_{s7}(h_7 - h_3)} = \frac{h_7 - h_8}{h_7 - h_3} \tag{14.12}$$

Similarly, the effectiveness of the low-temperature heat exchanger is

$$\varepsilon_l = \frac{\dot{m}_{s5}(h_4 - h_5)}{\dot{m}_{s5}(h_4 - h_1)} = \frac{h_4 - h_5}{h_4 - h_1} \tag{14.13}$$

where h_4, h_8 = enthalpy of mixture of concentrated and partly concentrated solutions entering low-temperature heat exchanger and enthalpy of concentrated solution leaving high-temperature heat exchanger, respectively, Btu/lb (kJ/kg)

Because the pump power is usually small compared to the heat supplied to the generator, it is often ignored in thermal analyses.

If the heat loss from the system components is ignored, then heat input to the absorption system must be equal to the heat removed from the system, in Btu/h·ton$_{ref}$ (kW/kW$_{ref}$), that is,

$$Q_{hg} + Q_{rl} = Q_{ab} + Q_{con} \tag{14.14}$$

Coefficient of Performance

If the pump power of the solution and refrigerant pumps is excluded, the coefficient of performance of a direct-fired absorption chiller COP$_c$ can be calculated as

$$\text{COP}_c = \frac{\text{refrigeration output}}{\text{HHV of fuel consumption}} = \frac{12{,}000\eta_{com}}{Q_{hg}} \tag{14.15}$$

where Q_{hg} = heat input to high-temperature generator per ton of refrigeration, Btu/h·ton$_{ref}$ (kW/kW$_{ref}$)
η_{com} = combustion efficiency of direct-fired generator

There are two kinds of heat input of fuel: the higher heating value (HHV) and lower heating value (LHV). The HHV is obtained when all the combustion products are cooled to the temperature existing before combustion and the water vapor formed during combustion is condensed. The LHV subtracts the latent heat of vaporization from the HHV. The LHV is about 90 percent of the HHV. For any given COP$_c$ value, it must be checked whether the heat input is based on HHV or LHV.

For double-effect, direct-fired, reverse parallel flow absorption chillers, the COP (HHV also including the combustion efficiency) of current products varies between 0.92 and 1.03.

Example 14.1. A typical double-effect, direct-fired, reverse parallel flow absorption chiller, described in Sec. 14.3 is operated under the following conditions.

The enthalpy of the solution at various points is as follows:

Point	Temperature,°F	Concentration, %	Enthalpy, Btu/lb
1	95	58.5	42
2	150	58.5	67.5
3	180	60.2	85
4	192	63.0	97
5	120	63.0	55.5
6	260	60.2	122.8
7	305	66.0	150
8	203	66.0	107.5

The mass flow rates of partly concentrated solution to the high-temperature heat exchanger and to combine with the concentrated solution from the high-temperature heat exchanger are equal. Finally, the amounts of water vapor boiled off from the high- and low-temperature generators are equal.

Calculate the effectiveness of the heat exchangers and the coefficient of performance of this absorption chiller.

Solution. From a psychrometric table, at a condensing temperature of 98°F, the enthalpy of saturated liquid water leaving the condenser $h_{lc} = 66.03$ Btu/lb, and the enthalpy of saturated water vapor leaving the evaporator h_{ve} at an evaporating temperature of 42°F is 1079.5 Btu/lb. From Eq. (14.3), the mass flow rate of refrigerant per ton of refrigeration is therefore

$$\dot{m}_r = \frac{12{,}000}{h_{ve} - h_{lc}} = \frac{12{,}000}{1079.5 - 66.03}$$
$$= 11.84 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

As described in the previous section, the concentration of dilute solution from absorber is 58.5%, and the mean concentration of solution entering the absorber is 63%. From Eq. (14.5), the mass flow rate of concentrated solution for each ton of refrigeration produced is therefore

$$\dot{m}_{s5} = \frac{X_1}{X_5 - X_1}\dot{m}_r = \frac{0.585}{0.63 - 0585}(11.84)$$
$$= 153.92 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

From Eq. (14.4), the mass flow rate of diluted solution per ton of refrigeration is

$$\dot{m}_{s1} = \dot{m}_{s5} + \dot{m}_r = 153.92 + 11.84 = 165.76 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

From Eq. (14.13), the effectiveness of the low-temperature heat exchanger can be calculated as

$$\varepsilon_l = \frac{h_4 - h_5}{h_4 - h_1} = \frac{97 - 55.5}{97 - 42} = 0.75$$

From Eq. (14.12), the effectiveness of the high-temperature heat exchanger can be calculated as

$$\varepsilon_h = \frac{h_7 - h_8}{h_7 - h_3} = \frac{150 - 107.5}{150 - 85} = 0.65$$

The mass flow rate of water vapor boiled off from the high-temperature generator is

$$\dot{m}_{rh} = 0.5\dot{m}_r = 0.5(11.84) = 5.92 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

From the psychrometric table, the enthalpy of the saturated water vapor at 300°F is h_{vhg} = 1180.04 Btu/lb. Also the mass flow rate of partly concentrated solution at point 6 can be calculated as

$$\dot{m}_{s6} = 0.5(\dot{m}_{s1} - 0.5\dot{m}_r) = 0.5[165.76 - 0.5(11.84)] = 79.92 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

And the mass flow rate of concentrated solution at point 7 can be calculated as

$$\dot{m}_{s7} = \dot{m}_{s6} - 0.5\dot{m}_r = 79.92 - 0.5(11.84) = 74.0 \text{ lb/h} \cdot \text{ton}_{\text{ref}}$$

From Eq. (14.8), the heat input to the high-temperature generator is then calculated as

$$Q_{hg} = \dot{m}_{rh}h_{vhg} + \dot{m}_{s7}h_7 - \dot{m}_{s6}h_6 = 5.92(1180) + 74(150) - 79.92(122.8)$$

$$= 6985.6 + 11{,}100 - 9814.2 = 8271.4 \text{ Btu/h} \cdot \text{ton}_{\text{ref}}$$

If the combustion efficiency of the direct-fired generator is 85 percent, from Eq. (14.15), the coefficient of performance of this double-effect, direct-fired, reverse parallel flow absorption chiller (HHV) can be calculated as

$$\text{COP}_c = \frac{12{,}000\eta_{\text{com}}}{Q_{hg}} = \frac{12{,}000(0.85)}{8271.4} = 1.23$$

If the heat losses from various system components of the absorption chiller are taken into consideration, then the calculated COP_c will be nearer to the value given by the manufacturer's catalogs, about 0.92 to 1.0.

14.5 ABSORPTION CHILLER CONTROLS

Capacity Control and Part-Load Operation

A double-effect, direct-fired, reverse parallel flow absorption chiller adjusts its capacity during part-load operation by modulation of the heat input to the high-temperature generator through the direct-fired burner, or other waste heat supply. A sensor is often located at the outlet of the evaporator to monitor the temperature of the chilled water leaving the evaporator T_{el}. As the system refrigeration load falls, T_{el} decreases accordingly. Once a drop of T_{el} below a predetermined set point is sensed, the DDC unit controller reduces the gas flow or other fuel supply to the direct-fired burner, and less water vapor is boiled off from the solution in the generator. Between 30 and 100 percent of the system design refrigeration load, the burner fired rate is modulated. Below 30 percent of design load, the burner is cycled on and off, and all refrigerant and solution pumps remain on so that the system refrigeration load is allowed to drop to 10 percent minimum of the design load. The refrigerant pump is cycled at minimum cooling.

Consequently, the concentration of the solution entering the absorber drops, less water vapor is extracted to the absorber, and, therefore, the rate of evaporation and refrigeration effect in the evaporator are both reduced until they are balanced with the reduction of refrigeration load so that T_{el} is maintained within acceptable limits. Meanwhile, because less water vapor has been extracted to the absorber from the evaporator, both evaporating pressure p_{ev} and evaporating temperature T_{ev} increase.

Although less water vapor is boiled off in the generators, the rate of heat transfer at the condensing surface and the amount of water vapor to be condensed to liquid water in the condenser remain the same. The condensing pressure p_{con} and temperature T_{con} decrease until a new balance is formed.

If T_{el} rises above a limit, on the other hand, more heat is provided to the generator, and the concentration of solution and the refrigeration capacity increase and T_{el} again falls within preset limits. Of course, the increase in solution concentration should not exceed the saturation limit.

The left-hand side of Fig. 14.5 shows the part-load absorption cycle of the double-effect, direct-fired, reverse parallel flow absorption chiller. During part-load operation, the following conditions exist:

- The mass flow rate of refrigerant is directly proportional to the load ratio.
- The evaporating pressure p_{ev} and temperature T_{ev} rise.
- The condensing pressure p_{con} and temperature T_{con} will drop.
- The boiled-off temperatures in the high- and low-temperature generators will decrease.
- Heat input to the high-temperature generator is reduced.
- The drop in temperature of cooling water at a lower outdoor wet-bulb temperature lowers the condensing temperature T_{con} and, therefore, the heat input.

During part-load operation, the evaporating and condensing temperatures; the concentrations of diluted, partly concentrated, and concentrated solutions; and the other operating parameters can be determined from the equations based on the heat balance analyses described in Sec.14.4.

Crystallization and Controls

In an absorption chiller using aqueous LiBr as the absorbent, the LiBr in a solution of constant concentration starts to crystallize when the solution temperature drops below the crystallization line on the equilibrium chart. The crystals formed are pure LiBr salt. Crystallization does not harm the absorption chiller, but it eventually decreases the concentration of the remaining solution. It is also a symptom of malfunction, and the cause of crystallization must be determined before normal operation can resume.

Absorption chillers are now designed to operate in the region away from the crystallization line shown in Fig. 14.5. There are also devices available that prevent crystallization and dissolve crystals if crystallization occurs. Crystallization is no longer the serious problem for currently designed absorption chillers that it was before the energy crisis.

Causes of crystallization include the following:

- Air leaks into the system raise the evaporating temperature and the chilled water leaving temperature T_{el}. A higher T_{el} increases the heat input and the solution concentration to the point of crystallization.
- When the system is operated at full load, if the temperature of the cooling water is too low, the diluted solution temperature may fall low enough to reduce the temperature of the concentrated solution to the point of crystallization.
- If the electric power is interrupted, the system ceases to operate. The temperature of the concentrated solution in the heat exchanger starts to drop and may fall below the crystallization line.

Manufacturers have developed several devices to minimize the possibility of crystallization. One such device uses a bypass valve to permit refrigerant to flow to the concentrated solution line when conditions that can cause crystallization are detected by the sensor. A recently developed microprocessor-based DDC unit controller uses measured temperatures and pressures at key points to calculate the concentration of the solution, to prevent crystallization.

Cooling Water Temperature Control

Earlier absorption chiller designs required a close control of the cooling water temperature to prevent crystallization. Current absorption chiller designs include control devices to prevent crystallization, so many manufacturers now allow the temperature of the cooling water to fall to lower value at part-load operation. A three-way mixing bypass valve should be installed to mix

bypass recirculating water from the condenser and to maintain the temperature of cooling water entering the absorber at a temperature specified by the manufacturer. Cooling tower fans can be cycled to supplement the bypass control.

Safety and Interlocking Controls

Low-Temperature Cutout. If the temperature of the refrigerant in the evaporator falls below a preset value, the DDC unit controller shuts down the absorption chiller to protect the evaporator from freezing. As soon as the refrigerant temperature rises above the limit, DDC unit controller starts the chiller again.

Chilled Water Flow Switch. When the mass flow rate of chilled water falls below a limit, a pressure-sensitive or flow-sensitive sensor alerts the DDC unit controller, which stops the absorption chiller.

Cooling Water Flow Switch. When a loss in cooling water supply is detected by the pressure- or flow-sensitive sensor, the DDC unit controller shuts down the absorption chiller. The chiller is started again only when the cooling water supply is reestablished.

High-Pressure Relief Valve. A high-pressure relief valve or similar device is often installed on the shell of the high-temperature generator to prevent the maximum pressure in the system from exceeding a preset value.

Direct-Fired Safety Controls. Similar to that described in Secs. 8.2 and 8.3, a direct-fired absorption chiller needs controls of high-pressure and low-pressure switches, flame ignition, and monitoring for its burner and generator.

Interlocked Controls. Absorption chiller should be interlocked with chilled water pumps, cooling-water pumps, and cooling tower fans so that the absorption chiller starts only when these pieces of equipment are in normal operation.

Optimizing On/Off Control. If multiple absorption chillers are installed in a chiller plant, as in a plant with multiple centrifugal chillers, optimizing control can be used to turn chillers on and off.

Monitoring and Diagnostics

Monitoring lights show the operating status of key parameters. Also fault detection and diagnostic displays determine the operating problem and necessary corrections.

14.6 OPERATING CHARACTERISTICS AND DESIGN CONSIDERATIONS

Difference between Absorption and Centrifugal Chillers

In a vapor compression refrigeration system, compressor power input is primarily affected by head ($p_{con} - p_{ev}$) or temperature lift ($T_{con} - T_{ev}$) of the refrigeration cycle. In an absorption chiller, $T_{con} - T_{ev}$ is also a factor that influences the heat input to the high-temperature generator, but its influence is significantly smaller than that of centrifugal chillers.

For the double-effect, direct-fired, reverse parallel flow absorption chiller described in Example 14.1, the heat input to high-temperature generator Q_{hg} is distributed as follows:

Heat input Q_{hg}	8271.4 Btu/h·ton$_{ref}$	100.0 percent
Latent heat of vaporization h_{fg}	6985.6 Btu/h·ton$_{ref}$	84.5 percent
Heat of diluted solution	1285.8 Btu/h·ton$_{ref}$	15.5 percent

Latent heat of vaporization is the primary factor that affects the heat input in an absorption chiller, and temperature difference $T_{con} - T_{ev}$ has only a minor effect on h_{fg}. This is the primary difference between an absorption chiller and a centrifugal chiller.

Evaporating Temperature

As in the centrifugal chiller, the evaporating temperature T_{ev} and pressure p_{ev} in an absorption chiller depend mainly on the chilled water temperature leaving the evaporator T_{el}. The difference $T_{el} - T_{ev}$ in current absorption chiller design is between 2 and 5°F (1.1 and 2.8°C). A smaller $T_{el} - T_{ev}$ means a higher COP and a large heat-transfer surface area in the evaporator. It is actually a compromise between energy cost and investment. For $T_{el} = 44$°F (6.7°C) and $T_{el} - T_{ev} = 2$°F (1.1°C), the evaporating pressure is around 6.8 mm Hg abs. (0.268 in. Hg abs. or 906 Pa abs.).

The vapor pressure of the concentrated solution during absorption should be slightly less than the evaporating pressure p_{ev}, so that water vapor vaporized in the evaporator can be extracted to the absorber. Both T_{ev} and T_{el} affect the cooling capacity of the absorption chiller as well as the heat input to the high-temperature generator.

Cooling Water Entering Temperature

The temperature of cooling water entering the absorber T_{ca}, in°F (°C), has the following effects on the performance of an absorption chiller:

- Lower T_{ca} means a higher cooling capacity.
- Lower T_{ca} results in a lower T_{con}.
- Lower T_{ca} means a lower heat input to the high-temperature generator per ton of refrigeration produced and, therefore, a high COP$_c$.
- At high load ratios, too low of a T_{ca} may cause crystallization.

Manufacturer recommendations should be followed. For example, one manufacturer recommends that a minimum of $T_{ca} = 72$°F (22.2°C) be maintained at all load conditions.

Heat Removed from Absorber and Condenser

The total amount of heat to be removed from the absorber and condenser in a typical double-effect, direct-fired, reverse parallel flow absorption chiller is about 1.5 times the heat rejected from the condenser in a centrifugal chiller. The heat to be removed in the absorber is about 60 percent of the total heat removal. Heat removed from the condenser is about 40 percent of total heat removal.

Usually, a cooling water temperature rise of 10 to 15°F (5.6 to 8.3°C) is used. For a cooling water temperature rise of 10°F (5.6°C) when the temperature entering the absorber is 85°F (29.4°C), the temperature of cooling water is about 91°F (32.8°C) entering the condenser and 95°F (35°C) leaving the condenser.

Condensing Temperature

Condensing temperature T_{con} depends mainly on the temperature of cooling water available and the heat-transfer surface area. If cooling water is at 85°F (29.4°C), allowing for a temperature increase

of 6°F (3.3°C) in the absorber and 4°F (2.2°C) in the condenser, and the difference between the condensing temperature and cooling water temperature leaving the condenser $T_{con} - T_{cl} = 3$°F (1.7°C), then condensing temperature T_{con} can be calculated as

$$T_{con} = 85 + 6 + 4 + 3 = 98\text{°F } (36.7\text{°C})$$

Condensing pressure p_{con} should be slightly lower than the boiled-off vapor pressure in the low-temperature generator in order to extract the vapor and be condensed in the condenser.

Corrosion Control

Many heat exchangers are typically of the shell-and-tube type. Shells are often made of mild steel or stainless steel, and tube bundles are usually made of copper or copper-nickel alloy. Lithium bromide is very corrosive. It attacks steel, copper, and copper alloys in the presence of air at temperatures above 300°F (149°C). Corrosion inhibitor should be used to protect the internal components against corrosive attacks, as specified by the manufacturer.

Rated Condition of Absorption Chiller

When one selects a double-effect, direct-fired, reverse parallel flow, water-cooled absorption chiller, the chiller's cooling capacity must meet the required system refrigeration load. According to ARI Standard 560, absorption chillers are rated under the following conditions:

Leaving chilled water temperature	44°F (6.7°C)
Chilled water temperature rise	10°F (5.6°C)
Chilled water flow rate	2.4 gpm/ton (0.043 L/s·kW)
Entering cooling water temperature	85°F (29.4°C)
Cooling water flow rate	4.4 gpm/ton (0.079 L/s·kW)
Fouling factor	0.00025 h·ft^2·°F/Btu (0.000044 m^2·C/W)

At design load, under operating conditions other than the standard rating conditions, both the cooling capacity and the heat input should be modified.

Minimum Performance

ASHRAE/IESNA Standard 90.1-1999 mandates the minimum efficiency requirements for absorption chillers at all capacities at rated conditions as follows:

Type	Minimum efficiency	Efficiency as of 10/29/2001
Air-cooled, single effect	0.48 COP	0.60 COP
Water-cooled, single effect	0.60 COP	0.70 COP
Double-effect, indirect-fired	0.95 COP, 1.00 IPLV	1.00 COP, 1.05 IPLV
Double-effect, direct-fired	0.95 COP, 1.00 IPLV	1.00 COP, 1.00 IPLV

14.7 ABSORPTION CHILLER-HEATERS

An absorption chiller-heater has nearly the same equipment and piping arrangement as those in an absorption chiller except for some additional valves and switches. The heating cycle operation of an

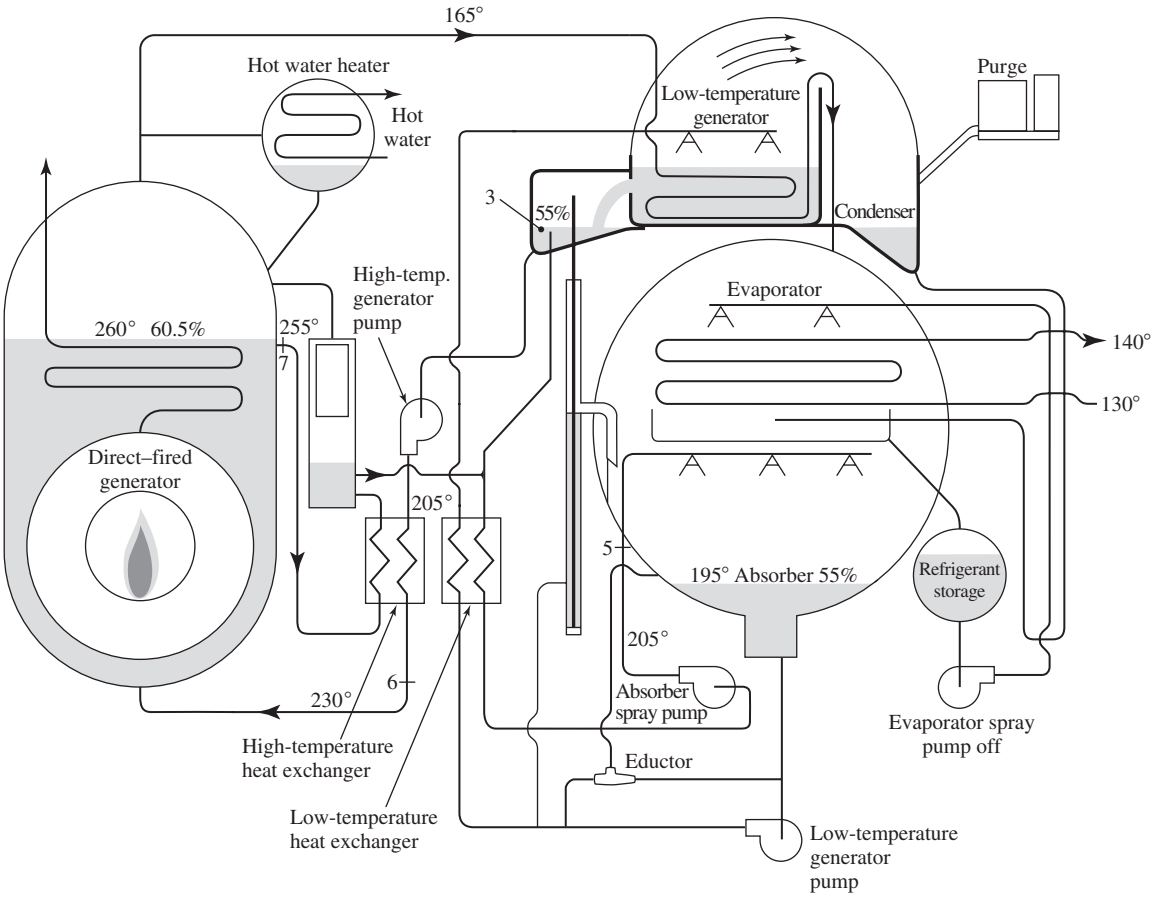


FIGURE 14.6 An absorption chiller-heater operated at heating cycle. (*Source: ASHRAE Handbook 1998, Refrigeration. Reprinted with permission.*)

absorption chiller is shown in Fig. 14.6. The evaporator is now acting as a condenser to provide hot water heating to an air conditioning space with a system efficiency comparable to that of a hot water or warm air heating system.

Heating Cycle

During the heating cycle operation in a chiller-heater, the diluted LiBr solution at a temperature of 195°F (90.6°C) and a concentration of 55% is extracted by the low-temperature solution pump and forced to the low-temperature generator through the low-temperature heat exchanger. In the low-temperature generator, the concentration of the solution remains the same, whereas the temperature of the diluted solution increases to 205°F (96°C). The solution is extracted again by a high-temperature solution pump and forced to the high-temperature generator through a high-temperature heat exchanger. In the high-temperature generator, the solution is heated by a direct-fired

burner. The water vapor boils off from the solution at a temperature of 165°F (74°C), flows inside the tube through the low-temperature generator, and enters the evaporator. The water vapor is then condensed to water because of the removal of the latent heat of condensation by the hot water tube bundle in which hot water is heated from 130 to 140°F (54 to 60°C). Hot water is supplied to the conditioned space for heating at a temperature of 140°F (60°C) by a circulating pump.

The concentrated solution at 255°F (124°C) and a concentration of 60.5% flows back from the high-temperature generator through high-temperature and low-temperature heat exchangers and enters the absorber at 205°F (96°C). In the absorber, the solution absorbs the condensed water from the evaporator and becomes a diluted solution of 55% concentration at 195°F (90°C). The diluted solution is extracted by the low-temperature solution pump and repeats the heating cycle again.

Actual Performance

According to the data listed in one leading manufacturer's catalog in the United States, the thermal efficiency E_t, in percent, of a chiller-heater during heating cycle operation is about 85 percent, including the combustion efficiency of the direct-fired burner.

14.8 ABSORPTION HEAT PUMPS

Functions of Absorption Heat Pump

An absorption heat pump extracts heat from a low-temperature heat source, such as waste heat or surface water, and delivers its heat output at a higher temperature for winter heating or other applications at a coefficient of performance greater than 1. In Japan and Sweden, absorption heat pumps have been installed in industrial and district heating plants using industrial waste heat to provide hot water, typically at 165°F (74°C), for winter heating or other purposes at a COP between 1.2 and 1.7. Absorption heat pumps can be used either for winter heating or for cooling in summer and heating in winter.

The coefficient of performance for cooling COP_c for an absorption heat pump can be calculated as described in Eq. (14.15). The coefficient of performance for heating for a double-effect absorption heat pump COP_h can be calculated as

$$\text{COP}_h = \frac{Q_{\text{ab}} + Q_{\text{con}}}{Q_{hg}} \tag{14.16}$$

where Q_{ab} = heat removed from absorber, Btu/h (kW)
Q_{con} = heat removed from condenser, Btu/h (kW)
Q_{hg} = heat input to high-temperature generator, Btu/h (kW)

Several absorbents, or working fluids, other than aqueous LiBr solution are being developed; however, $\text{LiBr}/\text{H}_2\text{O}$ is still the most widely used solution in absorption heat pumps.

Case Study: Series-Connected Absorption Heat Pump

Figure 14.7 is a schematic diagram of a 50-MW series-connected absorption heat pump installed in an incineration plant in Uppsala City, Sweden, as described by Astrand (1988). This series-connected absorption heat pump consists of two single-stage absorption heat pumps, each with an evaporator, absorber, generator, condenser, heat exchanger, and solution pump. Liquid water, as the refrigerant, evaporates in the evaporator. Water vapor is extracted by the concentrated solution in the absorber. The heat of absorption transferred to the hot water in the absorber is then used for

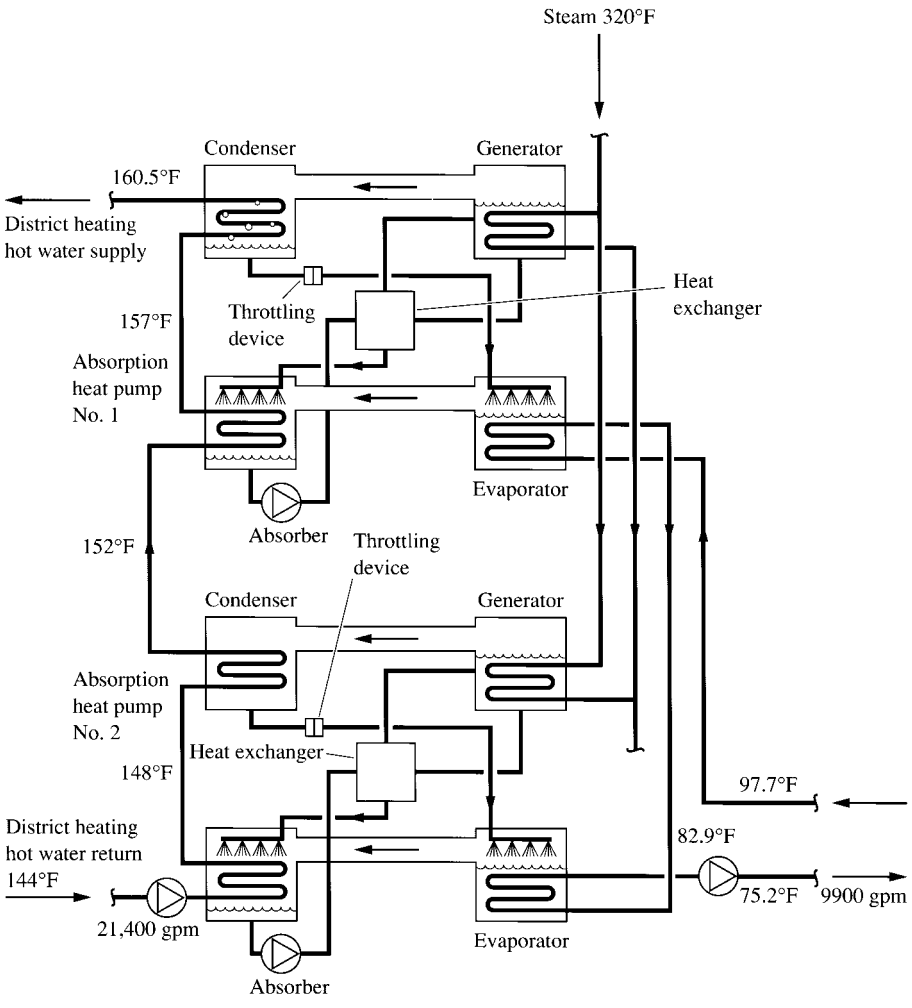


FIGURE 14.7 Schematic diagram of series-connected absorption heat pump.

district heating. The dilute solution is pumped from the absorber to the generator through the heat exchanger. In the generator, steam from the incineration plant boils off the water vapor from the dilute solution. The boiled-off water vapor is extracted to the condenser and condensed to liquid form. Latent heat of condensation is again rejected to the district heating hot water.

Concentrated solution from the generator flows to the absorber through the heat exchanger. Condensed liquid water enters the evaporator via a throttling orifice and is sprayed over the tube bundle in which flue gas cooling water flows from the incineration plant. After absorbing the latent heat of vaporization from the flue gas cooling water, liquid water evaporates into water vapor in the evaporator.

In this series-connected absorption heat pump, absorption heat pump no. 1 is operated at higher temperatures, and heat pump no. 2 is operated at lower temperatures. During operation in 1987, the return hot water from district heating was heated from 144 to 152°F (62.2 to 66.7°C) in the

absorber and condenser of the no. 2 absorption heat pump, and from 152 to 160.5°F (66.7 to 71.4°C) in the absorber and condenser of the no. 1 absorption heat pump. In the evaporator, heat was extracted from the low-temperature heat source, the flue gas cooling water, which entered the absorption heat pump at a temperature of 97.7°F (36.5°C) and left at a temperature of 75.2°F (24°C). The high-temperature heat source, steam, was supplied at 320°F (160°C) at a flow rate of 66,000 lb/h (29,940 kg/h) from the incineration plant. The average COP_{hp} for this series-connected absorption heat pump from July 15 through August 31, 1987, was 1.61.

14.9 ABSORPTION HEAT TRANSFORMER

Large amounts of low-temperature waste heat are released daily from many industrial plants. Absorption heat transformers are systems that operate in a cycle opposite to that of absorption chillers, to boost the temperature of input waste heat 40 to 100°F (22 to 56°C) higher for industrial applications.

Most absorption heat transformers today are single-stage systems using lithium bromide as the working fluid for industries in Japan and Europe. Two-stage absorption heat transformers for greater temperature boosting and use with working fluids other than LiBr are being developed.

System Description

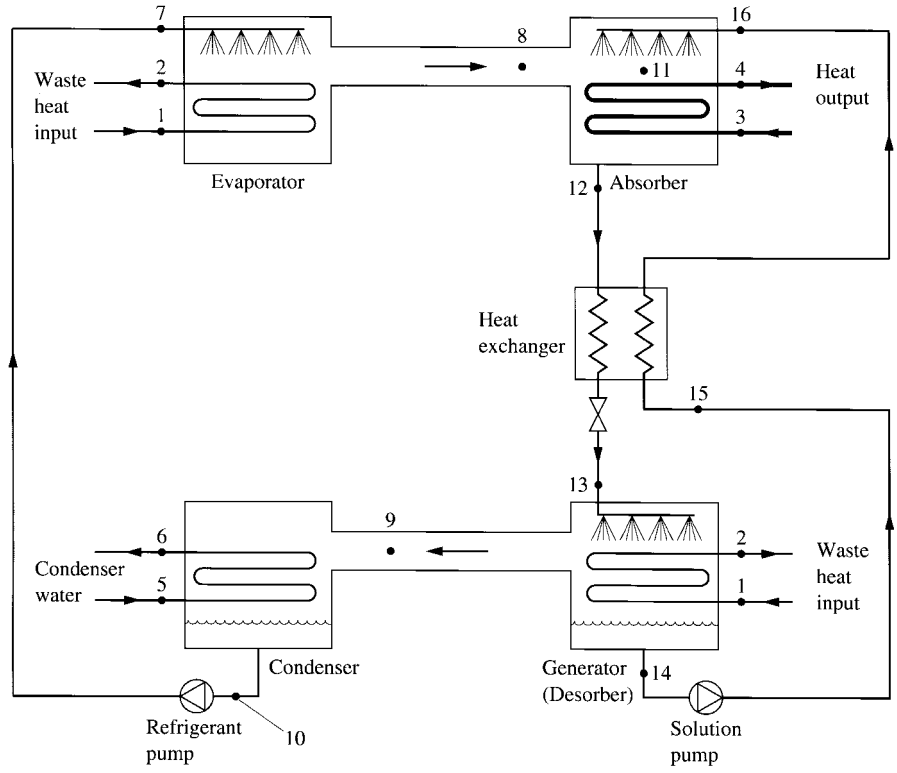
An absorption heat transformer consists of an evaporator, absorber, generator, condenser, heat exchanger, solution pump, refrigerant pump, piping, and accessories. Figure 14.8 shows the schematics and the absorption cycle on the equilibrium chart of a typical single-stage absorption heat transformer using LiBr as the absorbent. In a heat transformer, because the heat is supplied from the waste heat fluid stream entering at low temperature T_1 and leaving at lower temperature T_2, water is vaporized from the evaporator and extracted by the absorbent sprayed in the absorber at point 11. Heat of absorption is then released from the absorbent to the hot water stream entering at a temperature T_3 and raised to a temperature T_4. After absorbing the water vapor, the diluted solution at point 12 enters the heat exchanger, which is also called the recuperator. Diluted solution leaves the heat exchanger at a lower temperature, flows through a regulating valve, and enters the generator, or desorber, at point 13. Because of the heat input from the waste heat fluid streams, water vapor boils off from the diluted solution and is condensed in the condenser. The concentrated solution leaves the generator at point 14 and is pumped to a higher pressure equal to the pressure in the absorber. Liquid water in the condenser is pumped to the evaporator for evaporation again.

Operating Characteristics

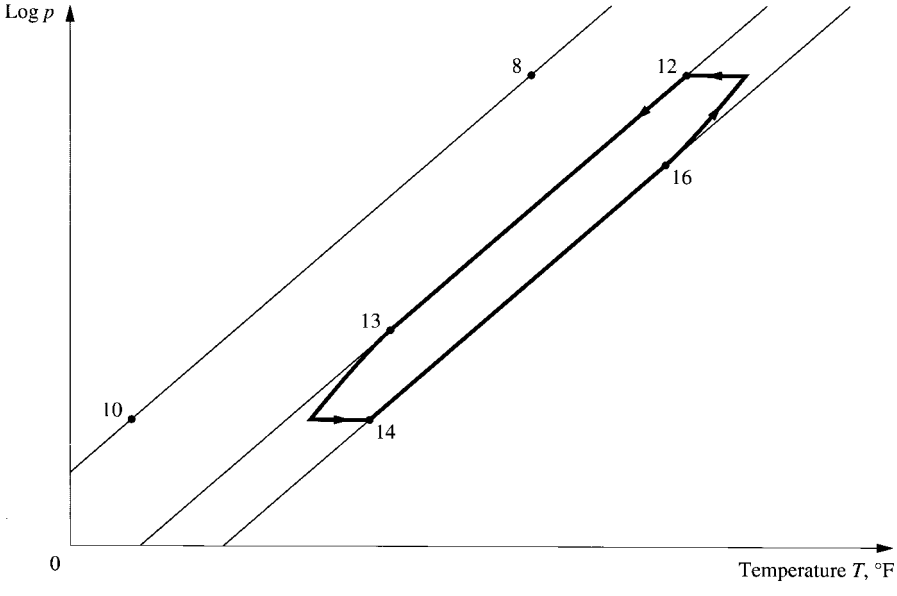
The absorption heat transformer operates at two pressure levels: high pressure, including the evaporator and absorber; and low pressure, including the generator and condenser. There are three temperature levels of input and output fluid streams:

- The fluid stream carrying the heat output from the absorber is at the highest temperature level.
- The heat source (the waste heat input to the evaporator and generator) is at the intermediate temperature level.
- The cooling water in the condenser is at the lowest temperature level.

The purpose of an absorption heat transformer is to boost the temperature of the input waste heat fluid stream, and the function of an absorption heat pump is to attain a higher COP from the lower-temperature heat source.



(a)



(b)

FIGURE 14.8 Single-stage absorption heat transformer: (*a*) schematic diagram; (*b*) absorption cycle.

Coefficient of Performance

The coefficient of performance of an absorption heat transformer $COP_{h,t}$ can be calculated as

$$COP_{h,t} = \frac{Q_{ab}}{Q_{ev} + Q_g} \tag{14.17}$$

where Q_{ev}, Q_g = waste heat input to evaporator and generator, Btu/h (kW). Mainly $COP_{h,t}$ is affected by the amount of temperature boosting. The higher the temperature boosting, the lower the $COP_{h,t}$. Usually $COP_{h,t}$ varies between 0.4 and 0.5.

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CHAPTER 15
AIR SYSTEMS: COMPONENTS—FANS, COILS, FILTERS, AND HUMIDIFIERS

15.1 FAN FUNDAMENTALS 15.2
- Functions and Types of Fans 15.2
- Fan Capacity or Volume Flow Rate 15.4
- Fan Pressure 15.4
- Fan Power and Fan Efficiency 15.4
- Air Temperature Increase through Fan 15.5
- Fan Performance Curves 15.5
- Influence of Elevation and Temperature 15.6

15.2 CENTRIFUGAL FANS 15.7
- Total Pressure Increase at Fan Impeller 15.7
- Backward-Curved Fans 15.8
- Radial-Bladed Fans 15.10
- Forward-Curved Fans 15.11
- Tubular or In-line Fans 15.12
- Unhoused Plug/Plenum Fans 15.12
- Centrifugal Roof Ventilators 15.14

15.3 AXIAL FANS 15.14
- Types of Axial Fans 15.14
- Velocity Triangles 15.16
- Performance Curves 15.18

15.4 FAN CAPACITY MODULATION 15.20
- Fan Speed Modulation with Adjustable-Frequency, Variable-Speed Drives 15.20
- Inlet Vanes Modulation 15.21
- Inlet Cone Modulation 15.23
- Blade Pitch Modulation 15.24

15.5 FAN SURGE AND STALL 15.24
- Fan Surge 15.24
- Fan Stall 15.24

15.6 FAN CONSTRUCTION AND ARRANGEMENTS 15.25
- Size and Class Standards 15.25
- Fan Width and Inlets 15.26
- Drive Arrangements and Direction of Discharge 15.26
- High-Temperature Fans 15.27
- Spark-Resistant Construction 15.28
- Safety Devices 15.28

15.7 FAN SELECTION 15.29
- Conditions Clarified and Factors Considered 15.29
- Estimated Fan Sound Power Level 15.30
- Comparison between Various Types of Fans 15.31
- Case Study—Selection of Fans 15.32

15.8 COIL CHARACTERISTICS 15.32
- Types of Coils 15.33
- Fins 15.33
- Contact Conductance 15.37
- Water Circuits 15.39

15.9 SENSIBLE COOLING AND SENSIBLE HEATING COILS—DRY COILS 15.39
- Heat Transfer in a Sensible Cooling Process 15.39
- Surface Heat-Transfer Coefficients 15.41
- Fin Efficiency 15.42
- Dry Coil Effectiveness 15.42
- Heating Coils 15.44
- Fluid Velocity and Pressure Drop 15.44
- Dry Coil at Part-Load Operation 15.44
- Coil Construction Parameters 15.45

15.10 WATER COOLING COILS—DRY-WET COILS 15.48
- Dry-Wet Boundary 15.48
- Dry Part 15.49
- Wet Part 15.50
- Dry-Wet Coil at Part-Load Operation 15.50
- Selection of a Dry-Wet Coil 15.50

15.11 COIL ACCESSORIES, CONDENSATE CONTROL, AND FREEZE-UP PROTECTION 15.56
- Coil Accessories 15.56
- Coil Cleanliness 15.57
- Condensate Collection and Drain System 15.57
- Coil Freeze Protection 15.58

15.12 AIR CLEANING, RATING, AND ASSESSMENTS 15.60
- Air Filtration and Industrial Air Cleaning 15.60
- Atmospheric Dust and Space Air Contaminants 15.61
- Rating and Assessments 15.61

15.13 TEST METHODS FOR AIR FILTERS 15.62
- Test Methods 15.62
- Test Unit 15.64

15.14 AIR FILTERS 15.65
- Filtration Mechanism 15.65
- Classification of Air Filters 15.65
- Coarse Air Filters 15.65
- Low-Efficiency Air Filters 15.66
- Medium-Efficiency Air Filters 15.67
- High-Efficiency Air Filters 15.68

Ultrahigh-Efficiency Filters 15.68
15.15 ELECTRONIC AIR CLEANERS 15.69
15.16 ACTIVATED CARBON ADSORBERS 15.70
15.17 SELECTION OF AIR FILTERS 15.71
15.18 HUMIDIFICATION AND HUMIDIFIERS 15.72
Space Relative Humidity 15.72
Humidifiers 15.72
Humidifying Load 15.72
Types of Humidifier 15.73
15.19 STEAM AND HEATING ELEMENT HUMIDIFIERS 15.73
Steam Grid Humidifiers 15.73
Steam Humidifiers with Separators 15.74
Heating Element Humidifiers 15.75
Characteristics and Requirements of Steam and Heating Element Humidifiers 15.76
15.20 ATOMIZING AND WETTED ELEMENT HUMIDIFIERS 15.76
Humidification Process 15.76
Ultrasonic Humidifiers 15.77
Case Study: White Plains Ultrasonic Humidification Project 15.77
Centrifugal Atomizing Humidifiers 15.77
Pneumatic Atomizing Humidifiers 15.78
Wetted Element Humidifiers 15.78
15.21 AIR WASHERS 15.79
Construction of an Air Washer 15.79
Functions of an Air Washer 15.80
Performance of an Air Washer 15.80
Bypass Control 15.81
Single-Stage or Multistage 15.81
15.22 CHARACTERISTICS OF ATOMIZING AND WETTED ELEMENT HUMIDIFIERS 15.82
15.23 SELECTION OF HUMIDIFIERS AND DESIGN 15.83
REFERENCES 15.85

15.1 FAN FUNDAMENTALS

Functions and Types of Fans

A fan is the prime mover of an air system or ventilation system. It moves the air and provides continuous airflow so that the conditioned air, space air, exhaust air, or outdoor air can be transported from one location to another through air ducts or other air passages.

A fan is also a turbomachine in which air is usually compressed at a compression ratio R_{com} not greater than 1.07. The *compression ratio*, dimensionless, is defined as

$$R_{com} = \frac{p_{dis}}{p_{suc}} \tag{15.1}$$

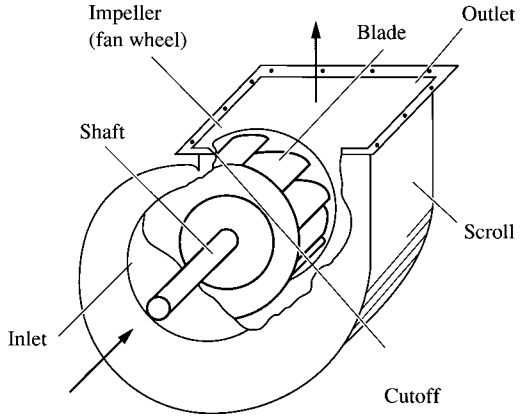
where p_{dis} = discharge pressure at outlet of compressor or fan, $lb_f/in.^2$ abs. or psia (kPa abs.)
p_{suc} = suction pressure at inlet of compressor or fan, psia (kPa abs.)

A blower is usually an enclosed multiblade rotor that compresses air to a higher discharge pressure. There is no clear distinction between a fan and a blower. Traditionally, blowers do not discharge air at low pressure as some fans do.

A fan is driven by a motor directly (direct drive) or via belt and pulleys (belt drive). Some large industrial fans in power plants are driven by steam or gas turbines.

Two types of fans are widely used in air conditioning and ventilation systems: centrifugal fans and axial fans (Fig. 15.1). Fans can be mounted individually as ventilating equipment to provide outdoor air or air movement inside a building. They can also transport air containing dust particles or material from one place to another via air duct systems. In air conditioning systems, fans are often installed in air-handling units, packaged units, or other air conditioning equipment.

In both centrifugal fans and axial fans, the increase of air static pressure is created by the conversion of velocity pressure to static pressure. In centrifugal fans, air is radially discharged from the impeller, also known as the fan wheel—air turns 90° from its inlet to its outlet. In an axial fan, the



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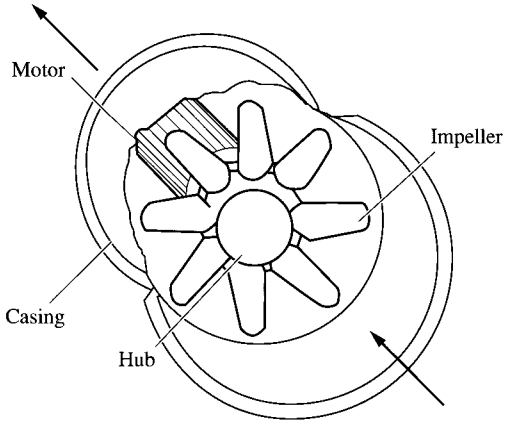

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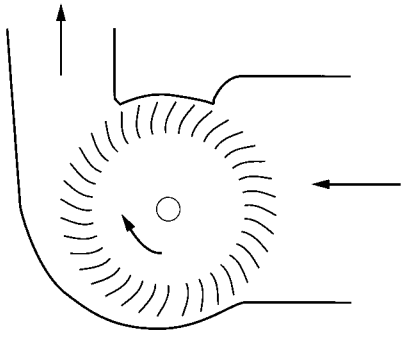


(c)

FIGURE 15.1 Types of fans: (*a*) centrifugal; (*b*) axial; (*c*) cross-flow.

direction of airflow is parallel to the axle of the fan. Additional differences in fan characteristics between these two types of fans are discussed in the following sections.

Crossflow fans force the airflow by means of a long rotor with many vanes, as shown in Fig. 15.1*c*. This type of fan has limited application in small air conditioning equipment because it is less efficient and has a higher sound level than centrifugal fans.

Fan Capacity or Volume Flow Rate

Fan capacity or the fan volume flow rate $\dot{V}_f$, in cfm (m^3/s), is defined as the rate of volume flow measured at the inlet of the fan, corresponding to a specific fan total pressure. It is usually determined by the product of the duct velocity and the area of the duct connected to the fan inlet, according to the test standard Air Movement and Control Association (AMCA) Standard 210-85 and ASHRAE Standard 51-1985.

Fan volume flow rate is independent of air density ρ. However, fan total pressure is affected by air density. Therefore, the fan volume flow rate is normally rated at standard air conditions, i.e., dry air at an atmospheric pressure of 14.696 psia (101,325 Pa abs.), a temperature of 70°F (21.1°C), and a density of 0.075 lb/ft^3 (1.2 kg/m^3).

Fan Pressure

Fan total pressure Δp_{tf}, expressed in inches of height of water column (inches WC or Pa), is the total pressure rise of a fan, i.e., the pressure difference between the total pressure at the fan outlet p_{to} and the total pressure at the fan inlet p_{ti}, both in in. WG (Pag), or

$$\Delta p_{tf} = p_{to} - p_{ti} \tag{15.2}$$

Fan velocity pressure p_{vf}, in in. WC (Pa), is the pressure calculated according to the mean velocity at the fan outlet v_o, in fpm (m/s). If air density $\rho = 0.075$ lb/ft^3 (1.2 kg/m^3), it can be calculated as

$$p_{vf} = p_{vo} = \frac{\rho v_o^2}{2g_c} = \left(\frac{v_o}{4005}\right)^2 = \left(\frac{\dot{V}_o}{4005A_o}\right)^2 \tag{15.3}$$

where p_{vo} = velocity pressure at fan outlet, in. WC (Pa)
g_c = dimensional constant, 32.2 $lb_m \cdot ft/lb_f \cdot s^2$ ($kg \cdot m/N \cdot s^2$)
$\dot{V}_o$ = volume flow rate at fan outlet, cfm (m^3/s)
A_o = cross-sectional area of fan outlet, ft^2 (m^2)

Fan static pressure Δp_{sf}, in in. WC (Pa), is the difference between the fan total pressure and fan velocity pressure, or

$$\Delta p_{sf} = \Delta p_{tf} - p_{vf} = p_{to} - p_{ti} - p_{vo} = p_{so} - p_{ti} \tag{15.4}$$

where p_{so} = static pressure at fan outlet, in. WG (Pa).

Fan Power and Fan Efficiency

Air power P_{air}, in hp (W), is the work done in moving the air along a conduit against a fan total pressure Δp_{tf}, in in. WC (Pa), at a fan volume flow rate of $\dot{V}_f$ in cfm (m^3/s). Because 1 hp = 33,000 $ft \cdot lb_f/min$ and 1 in. WC = 5.192 lb_f/ft^2,

$$P_{\text{air}} = \frac{\Delta p_{tf} \times 5.192 \times \dot{V}_f}{33{,}000} = \frac{\Delta p_{tf} \dot{V}_f}{6356} \tag{15.5}$$

The fan power input on the fan shaft, often called the brake horsepower P_f, can be calculated as

$$P_f = \frac{\Delta p_{tf} \dot{V}_f}{C\eta_t} = \frac{\Delta p_{tf} \dot{V}_f}{6356\eta_t} \tag{15. 6}$$

When Δp_{tf} is expressed in Pa, fan volume flow rate in m^3/s, fan total efficiency η_t is the ratio of air power P_{air} to fan power input P_f on the fan shaft, dimensionless, then constant $C = 1$, and fan power input P_f is expressed in W. If Δp_{tf} is expressed in in. WC, fan volume flow rate is in cfm, fan total efficiency η_t is a dimensionless ratio, then constant $C = 6356$ and fan power input on the shaft P is in hp.

From Eq. (15.6), fan total efficiency can also be calculated as

$$\eta_t = \frac{\Delta p_{tf} \dot{V}_f}{6356 P_f} \tag{15.7}$$

Fan total efficiency is a combined index of aerodynamic, volumetric, and mechanical efficiencies of a fan. Fan static efficiency η_s is defined as the ratio of the product of the fan static pressure Δp_{sf}, in in. WC, and the fan volume flow rate to the fan power input, i.e.,

$$\eta_s = \frac{\Delta p_{sf} \dot{V}_f}{6356 P_f} \tag{15.8}$$

Air Temperature Increase through Fan

If air density $\rho_a = 0.075$ lb/ft^3, the specific heat of air $c_{pa} = 0.243$ Btu/lb·°F, and 1 hp = 42.41 Btu/min, the relationship between fan power input and the air temperature increase when it flows through the fan ΔT_f, in °F, is given as

$$P_f = \frac{\dot{V}_f \rho_a c_{pa} \Delta T_f}{42.41} \tag{15.9}$$

Combining Eqs. (15.6) and (15.9), then, gives

$$\Delta T_f = \frac{0.00667 \Delta p_{tf}}{\rho_a c_{pa} \eta_t} = \frac{0.37 \Delta p_{tf}}{\eta_t} \tag{15.10}$$

This air temperature rise in a fan when air flows through it is caused by the compression process and energy losses that occur inside the fan. When air flows through the air duct, duct fittings, and equipment, the duct friction loss and dynamic losses cause a temperature increase as mechanical energy is converted to heat energy. However, this temperature increase in the air duct is offset by a temperature drop caused by the expansion of air due to the reduction of static pressure along the airflow. Therefore, it is more convenient to assume that the air temperature rise occurs because of the friction and dynamic losses along the airflow only when air is flowing through the fan.

Fan Performance Curves

Fan characteristics can be described by certain interrelated parameters such as volume flow rate, pressure, power, and efficiency. These characteristics are depicted graphically by fan performance curves in Fig. 15.2.

Fan characteristic curves usually set the volume flow rate $\dot{V}$, in cfm (m^3/s), as the abscissa; and fan total pressure Δp_{tf}, fan static pressure Δp_{sf}, fan power input P_f, or fan total efficiency η_t as the ordinate. In Fig. 15.2, there are three pressure-volume flow characteristic curves:

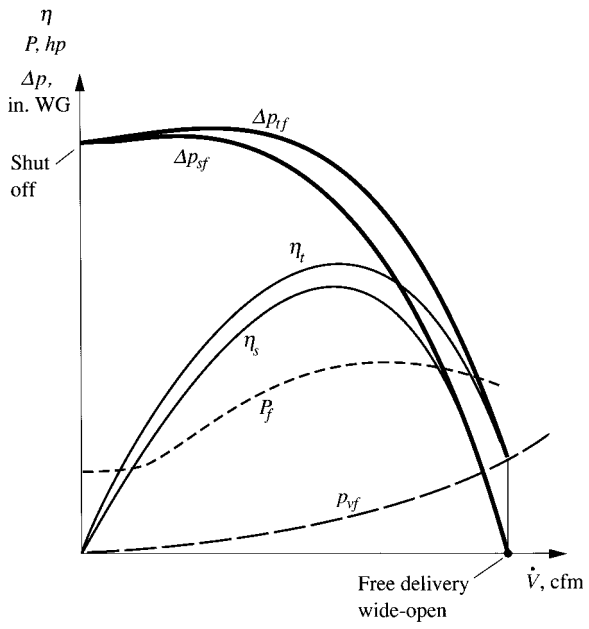


FIGURE 15.2 Fan performance curves.

1. Total pressure-volume flow Δp_t-$\dot{V}$
2. Static pressure-volume flow Δp_s-$\dot{V}$
3. Velocity pressure-volume flow p_v-$\dot{V}$

The point where the Δp_t-$\dot{V}$ or Δp_s-$\dot{V}$ curve intersects with ordinate is called the shutoff point. At this point, the block tight or completely shut-off volume flow rate $\dot{V}_{\text{block}} = 0$. Moreover, static pressure is equal to total pressure at this point.

The volume flow rate at the point of intersection either between the Δp_t-$\dot{V}$ and p_v-$\dot{V}$ curves or between the Δp_s-$\dot{V}$ curve and the x axis is called *free delivery*, or the *wide-open volume flow rate*. At the point of free delivery, $\Delta p_s = 0$, $\Delta p_t = p_v$, and fan static efficiency $\eta_s = 0$.

In Fig. 15.2, the other curves shown are the fan total efficiency η_t-$\dot{V}$ curve, the fan static efficiency η_s-$\dot{V}$ curve, and the fan power input P-$\dot{V}$ curve. The efficiency curves show that at the shutoff point, both η_t and η_s are zero, and at free delivery η_s is again zero. It is important that the fan be operated near maximum efficiency as much as possible. Because of friction and dynamic losses, fan power input at the shutoff point is not equal to zero.

Influence of Elevation and Temperature

Both elevation and temperature have an influence on air density ρ_a, in lb/ft^3 (kg/m^3), so they affect the fan total pressure. The fan total pressure Δp_{tf}, in in. WC (Pa), required at high elevations and temperatures with corrections can be calculated as

$$\Delta p_{tf} = K_T K_{\text{el}} \Delta p_{t,s} \tag{15.11}$$

where $\Delta p_{t,s}$ = fan total pressure at standard air condition, in. WC (Pa). In Eq. (15.11), K_{el} is an elevation factor and can be evaluated as

$$K_{el} = \frac{p_{at}}{29.92} \tag{15.12}$$

Here p_{at} represents the atmospheric pressure at high elevation, in in. Hg. Between sea level and an elevation of 5000 ft above sea level, p_{at} can be roughly estimated as

$$p_{at} = 29.92 - 0.001H_{el} \tag{15.13}$$

where H_{el} = elevation above sea level, ft. Temperature factor K_T can be calculated as

$$K_T = \left(\frac{530}{T_a + 460}\right)^{0.825} \tag{15.14}$$

where T_a = air temperature, °F (°C).

15.2 CENTRIFUGAL FANS

Total Pressure Increase at Fan Impeller

When air flows through the impeller of a centrifugal fan, its total pressure rise is closely related to the peripheral velocities of the impeller as well as the tangential component of the peripheral velocities entering and leaving the impeller. Figure 15.3 shows the velocity triangles at the blade inlet and outlet of the impeller of a centrifugal fan. Let u represent the peripheral velocity of the impeller, w the relative velocity in a direction tangential to the blade profile, and v the absolute velocity of the fluid elements. The angle between the relative velocity w and the peripheral velocity u is called the blade angle β. The tangential component of the absolute velocity is indicated by v_u and the radial component by v_m. Subscripts 1 and 2 denote the velocities at the inlet and outlet of the blade.

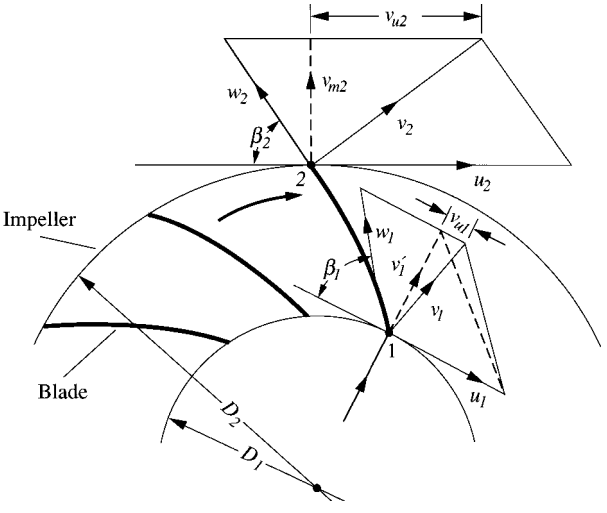


FIGURE 15.3 Velocity triangles at the blade inlet and outlet of a centrifugal fan.

Air enters the impeller at the blade inlet with an absolute velocity of v_1. As the fluid elements flow through the blade passage, they accelerate. Moreover, a guiding pressure is exerted on the fluid elements by the blades because of the rotation of the impeller. Air then leaves the blade outlet with an absolute velocity v_2.

If the energy losses due to the friction and eddies are ignored, and if the velocities are expressed in fpm, then the total pressure rise, or fan total pressure Δp_t, in in. WC, of the centrifugal fan when air flows through the impeller is given as

$$\Delta p_t = \frac{\rho(v_{u2}u_2 - v_{u1}u_1)}{60 \times 60 \times 5.192 g_c}$$

$$= 1.66 \times 10^{-6} \rho(v_{u2}u_2 - v_{u1}u_1) \tag{15.15}$$

where ρ = air density, lb/ft^3
g_c = dimensional constant, 32.2 lb$_m$ · ft/lb$_f$ · s^2 (kg · m/N · s^2)

If the impeller is deliberately designed at a specific blade inlet angle β_1 and a radial component v_{m1} so that the tangential component $v_{u1} = 0$, then the fan total pressure becomes

$$\Delta p_t = 1.66 \times 10^{-6} \rho v_{u2} u_2 \tag{15.16}$$

From Eqs. (15.15) and (15.16), the fan total pressure of a centrifugal fan depends mainly on the peripheral velocity u_2 and the tangential component v_{u2} and, therefore, the configuration of the blades. The peripheral velocity at the blade outlet u_2, often called the tip speed, in fpm (m/min), can be calculated as

$$u_2 = \pi D_2 n \tag{15.17}$$

where D_2 = outside diameter of fan impeller, ft (m)
n = rotational speed of impeller, rpm

For the same type of centrifugal fan, the greater the diameter, the greater the volume flow rate and fan total pressure. For a centrifugal fan of a specific type and size, at each impeller speed, there will be a set of different Δp_t-$\dot{V}$, P-$\dot{V}$, and η_t-$\dot{V}$ curves. The higher the speed, the greater $\dot{V}$, Δp_t, and P.

Another important parameter that affects the characteristics of centrifugal fans is the blast area A_{blast}, in ft^2 (m^2), as shown in Fig. 15.4. The blast area is the cross-sectional area just above the cutoff point where discharge air is prevented from recirculating. It is always smaller than the outlet area of the centrifugal fan A_o, in ft^2 (m^2). Blast area actually determines the maximum velocity discharged from the fan outlet.

Based on the shape of the blade and the direction of air discharged from the impeller, centrifugal fans can be categorized as backward-curved, radial-bladed, forward-curved, tubular or in-line, unhoused or cabinet fans, and roof ventilators.

Backward-Curved Fans

In a backward-curved or backward-inclined centrifugal fan, the blade tip inclines away from the direction of rotation of the impeller. The β_2 angle of a backward-curved centrifugal fan is less than 90°. Figure 15.3 shows the velocity triangles of a backward-curved centrifugal fan. The impeller of a backward-curved centrifugal fan usually consists of 8 to 16 blades. For greater efficiency, the shape of the blades is often streamlined to provide minimum flow separation and, therefore, minimum energy losses. Backward-curved centrifugal fans with such blades are called airfoil fans, as distinguished from fans with sheet-metal blades. The blades in a backward-curved fan are always longer than those of a forward-curved fan. A volute or scroll casing is used. This shape converts some of the velocity pressure to static pressure at the fan outlet.

From the velocity triangles at the blade outlet, the tangential component $v_{u2} = u_2 - v_{m2}\cot\beta_2$ and the radial component $v_{m2} = \dot{V}/(\pi D_2 b_2)$. Here b_2 represents the width of the impeller at the blade tip.

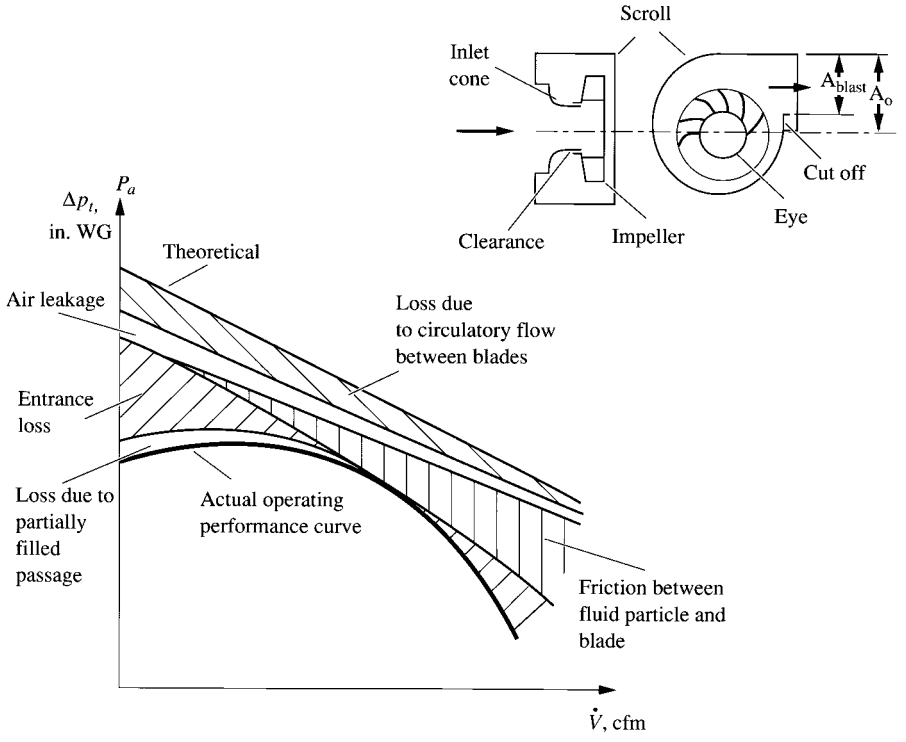


FIGURE 15.4 Operating characteristics for a backward-curved centrifugal fan.

The total pressure developed is

$$\Delta p_t = C_1 \rho v_{u2} u_2 = C_1 \rho \left(u_2^2 - \frac{u_2 \dot{V} \cot \beta_2}{\pi D_2 b_2} \right) \tag{15.18}$$

where C_1 = constant equal to 1.66×10^{-6}. Because $\beta_2 < 90°$, $\cot \beta_2$ is a positive value, and the theoretical Δp_t-$\dot{V}$ curve shown in Eq. (15.18) is a straight line that declines at an increasing volume flow rate, as shown in Fig. 15.4.

In actual operation, however, when the air flows through the centrifugal fan, it encounters the following energy losses:

- Circulatory flow between the blades
- Air leakage at the inlet
- Friction between fluid particles and the blade
- Energy loss at the entrance
- Partially filled passage

These energy losses change the shape of the Δp_t-$\dot{V}$ curve. The actual performance curve of a backward-curved centrifugal fan is a concave curve declining toward the right-hand side, as shown in Fig. 15.4. It can be seen that the maximum total pressure is slightly higher than that at the shutoff condition. After reaching the maximum Δp_t, it drops sharply as the volume flow rate increases.

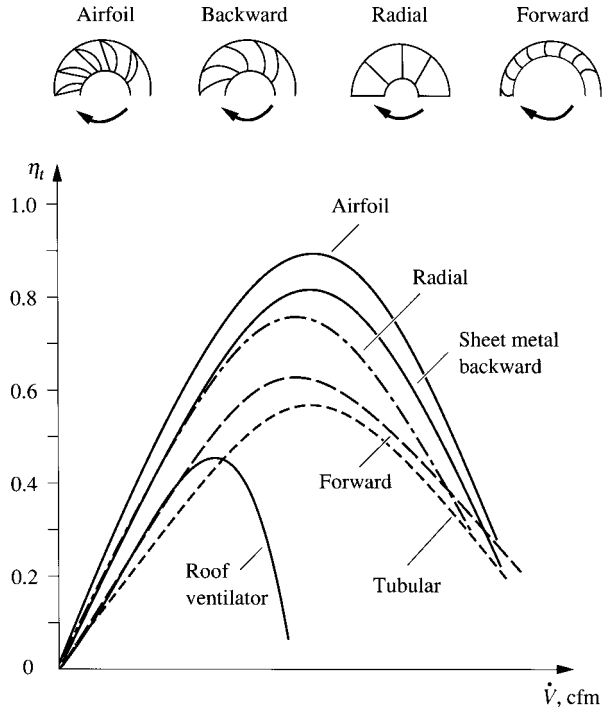


FIGURE 15.5 Total efficiency–volume η_t-$\dot{V}$ performance curves for centrifugal fans.

The fan total efficiency versus volume flow η_t-$\dot{V}$ curves for backward-curved centrifugal fans are concave. All have $\eta_t = 0$ at block tight condition as $\dot{V} = 0$. They also have a maximum efficiency between 50 and 65 percent of the wide-open volume flow, as shown in Fig. 15.5. Among these η_t-$\dot{V}$ curves, the airfoil blade backward-curved centrifugal fan has the highest fan total efficiency.

Figure 15.6 shows the fan power–volume flow P-$\dot{V}$ curves for centrifugal fans using impellers of the same size. Centrifugal fans need the minimum amount of power at block tight condition. This power input is used to offset the aerodynamic and mechanical losses even when the volume flow is zero. For backward-curved centrifugal fans, fan power P tends to increase initially as the volume flow rate increases. Then, P reaches a maximum point, after which P declines as the volume flow rate is further increased. The P-$\dot{V}$ curve of a backward-curved centrifugal fan has a shape similar to that of the Δp_t-$\dot{V}$ curve. If the size of the fan motor is rated at or near the highest fan total efficiency, the power required is often nearly the maximum value, and the motor needs no more fan power at the same speed and density than at any other operating conditions. Such a P-$\dot{V}$ curve is called a nonoverloading curve.

Radial-Bladed Fans

The blades in a radial-bladed centrifugal fan are either straight or curved at the blade inlet. The blade tip or blade outlet is always radial; that is, $\beta_2 = 90°$, as shown in Fig. 15.7*a*. Usually, there are 6 to 10 blades in a radial-bladed impeller. The construction of the radial blades is comparatively simple.

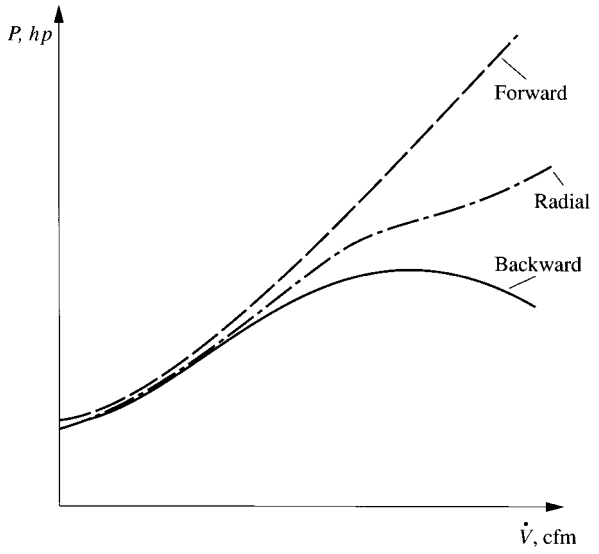


FIGURE 15.6 Power-volume P-$\dot{V}$ performance curves for centrifugal fans with impellers of same diameters.

Figure 15.8 shows the Δp_t-$\dot{V}$ curves for centrifugal fans with impellers of the same diameter. The Δp_t-$\dot{V}$ curve of a radial-bladed centrifugal fan is usually steeper than that of a backward-curved fan, with a higher Δp_t and a smaller $\dot{V}$. The fan total efficiency of a radial-bladed fan is lower than that of a backward-curved fan. Furthermore, the fan power input always increases as the volume flow becomes greater. Such an overloading P-$\dot{V}$ curve indicates that the motor may be overloaded.

The radial-bladed centrifugal fan is often used in industrial applications to transport particles or products because the spaces between the blades are not easily clogged.

Forward-Curved Fans

In a forward-curved centrifugal fan, the blade tip inclines in the direction of rotation of the impeller. At the outlet of the blade, blade angle $\beta_2 > 90°$, as shown in Fig. 15.7*b*. Because of a greater absolute velocity v_2 at the blade outlet, the blades are shorter and the speed required to produce a specific fan total pressure and volume flow rate is much lower than that for a backward-curved centrifugal fan. The impeller generally has 24 to 64 blades.

If the diameters of the impellers of a forward-curved fan and a backward-curved fan are the same, then the Δp_t-$\dot{V}$ curve of the forward-curved centrifugal fan is flatter than that of the backward-curved fans, as shown in Fig. 15.8. Forward-curved fans have a lower fan total efficiency than backward-curved fans, and the maximum efficiency of the forward-curved fan occurs between 40 and 50 percent of the wide-open volume flow. The forward-curved fan power curve tends to bend upward more steeply than that of other centrifugal fans, as shown in Fig. 15.6. Therefore, the motor of the forward-curved fan may be overloaded when the volume flow rate is greater than the rated value.

Forward-curved fans are the most compact centrifugal fans available in terms of volume flow rate delivered per unit space occupied. They are also less stable, as indicated by the saddle shape of their Δp_t-$\dot{V}$ curve. For the same size fan wheel, forward-curved fans have a greater inlet than backward-curved fans.

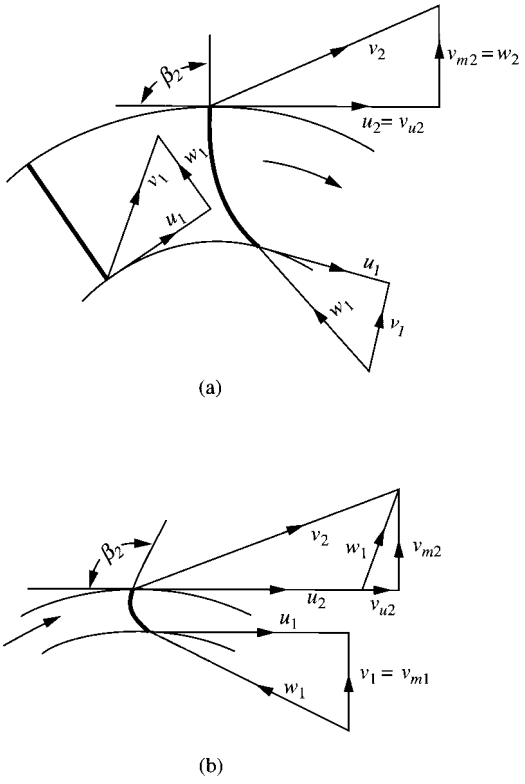


FIGURE 15.7 Velocity triangles of radial-bladed and forward-curved centrifugal fan: (*a*) radial-bladed; (*b*) forward-curved.

Tubular or In-line Fans

A tubular or in-line centrifugal fan generally consists of an impeller with airfoil blades, an inner cone, a set of fixed vanes, and a cylindrical casing, as shown in Fig. 15.9. Air enters the tubular fan at the inlet and then flows through the blade passages of the impeller. It is discharged radially against the inner surface of the cylindrical casing and is then deflected in a direction parallel to the axle of the fan, in order to produce a straight-through flow. The impeller of a tubular centrifugal fan may have 6 to 12 blades. Fixed vanes are used to convert velocity pressure to static pressure to prevent swirls and straighten the airflow.

In Fig. 15.8, it can be seen that the Δp_t-$\dot{V}$ curve for a tubular centrifugal fan is similar to that of a backward-curved fan. Because of the 90° turn of the airflow after the impeller, both total pressure and total efficiency are lower than those of a backward-curved fan, and the sound power level is greater. The fan power P-$\dot{V}$ curve of this type of fan is also a nonoverloading curve. The main advantage of the tubular fan is its straight-through airflow.

Unhoused Plug/Plenum Fans

Unhoused plug/plenum fans are centrifugal fans with no outer casing or scroll. The impeller is mounted in a plenum with a flexibility of the location of the discharged outlet. Unhoused centrifugal fans are simpler in construction and can plug into an air system where a plenum can be used as the outer casing of the impeller, and the two ends of the plenum used as the inlet and outlet of the fan.

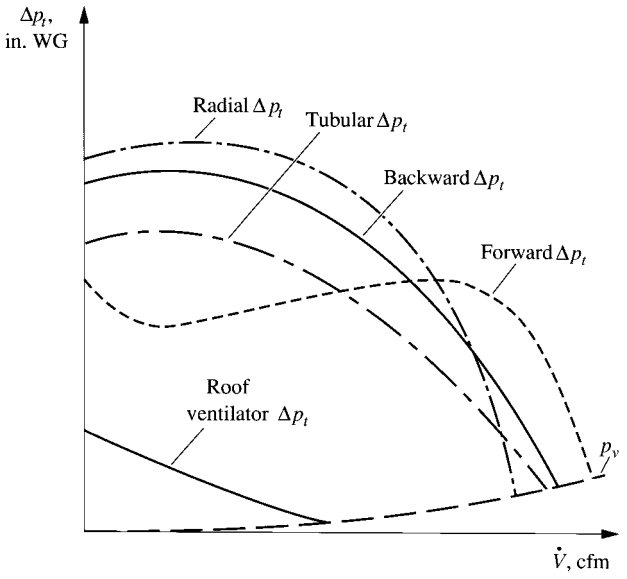


FIGURE 15.8 Pressure-volume Δp_t-$\dot{V}$ performance curves for centrifugal fans with same impeller diameter.

Coward (1997) noted the influence of the gap between the plenum wall and the impeller on the fan volume flow rate and fan power input as well as various types of duct entrance losses. The performance of an unhoused plug fan should be predictable and should be carefully taken into consideration during design.

Unhoused plug/plenum fans with backward-curved impellers may require more power than centrifugal fans with outer casings (fan total efficiency is usually between 58 and 63 percent). They are lower in sound power level if the plenum is properly sized and acoustically treated. Unhoused

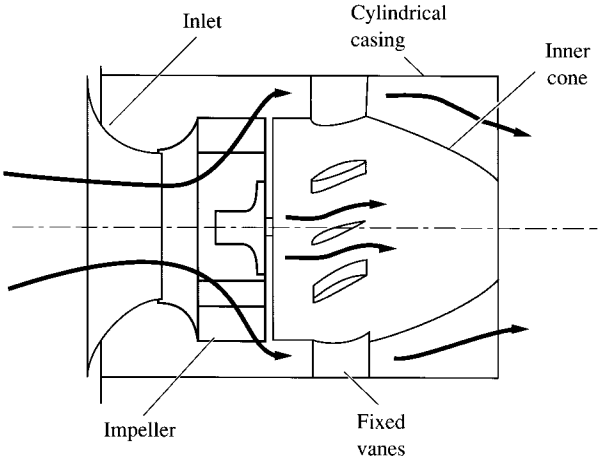


FIGURE 15.9 Tubular centrifugal fan.

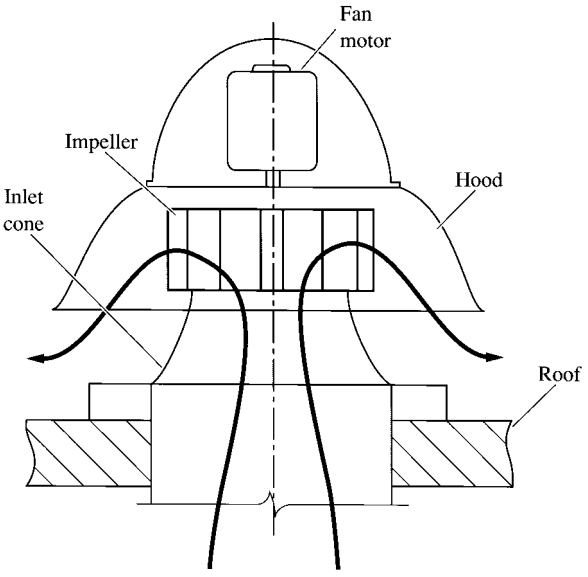


FIGURE 15.10 Centrifugal roof ventilator.

plug/plenum fans are often used as return fans in many rooftop packaged units and as exhaust fans in industrial applications.

Centrifugal Roof Ventilators

A centrifugal roof ventilator is often mounted on a roof to exhaust air from a ventilated space, as shown in Fig. 15.10. Roof ventilators can also be used as intake devices to draw outdoor air through the roof. In Fig. 15.10, air is extracted from ventilated space and enters the inlet cone directly or through a connecting duct having a total pressure loss usually less than 0.2 in. WC. After flowing through a backward-curved centrifugal impeller, air is discharged radially against the inner surface of the hood. The sheet-metal or airfoil blades of the centrifugal roof ventilator are made of aluminum alloy, stainless steel, or well-protected structural steel to provide good corrosive resistance above the roof.

The centrifugal roof ventilator usually operates at a low total pressure and a large volume flow rate, as illustrated by a downward inclined Δp_t-$\dot{V}$ curve, as shown in Fig. 15.8. The power P-$\dot{V}$ curve is a nonoverloading curve.

15.3 AXIAL FANS

Types of Axial Fans

For an axial fan, a parameter called the hub ratio is closely related to its characteristics. *Hub ratio* R_{hub} is defined as the ratio of hub diameter D_{hub}, in ft (m), to the tip-to-tip blade diameter or diameter of impeller D_{bt}, in ft (m),

$$R_{hub} = \frac{D_{hub}}{D_{bt}} \tag{15.19}$$

The higher the hub ratio, the greater the conversion of velocity pressure to static pressure because of the larger difference of the areas: $\pi(D_{bt}^2 - D_{hup}^2)/4$.

The capacity and fan total pressure of axial fans can be increased by raising their rotating speed or through the adjustment of the blade pitch angle to a higher value. This characteristic is important for axial fans that are driven directly by motor, without belts.

Axial fans can be subdivided into the following three types:

Propeller Fans. In a propeller fan, an impeller having 3 to 6 blades is mounted within a circular ring or an orifice plate, as shown in Fig. 15.11*a*. The blades are generally made of steel or molded plastic and sometimes may increase in width at the blade tip. If the impeller is mounted inside an orifice plate, the direction of airflow at the blade tip will not be parallel to the axle. Eddies may form at the blade tips. Propeller fans are usually operated at very low static pressure with large volume flow. They often have a hub ratio $R_{hub} < 0.15$.

Tube-Axial Fans. The impeller of a tube-axial fan usually has 6 to 9 blades. It is mounted within a cylindrical casing. The blades can be airfoil blades or curved sheet metal. Airfoil blades are

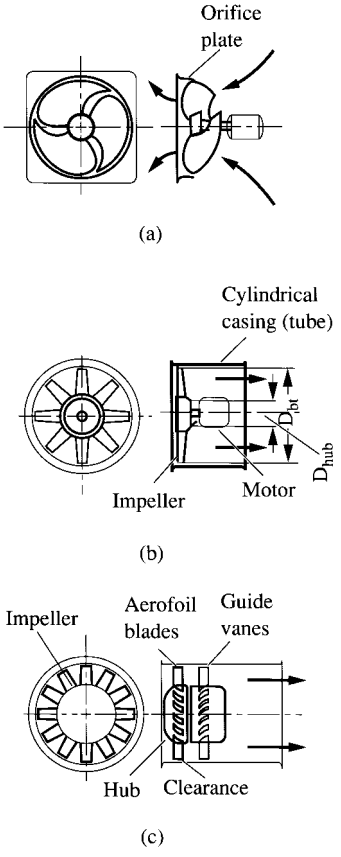


FIGURE 15.11 Axial fans: (*a*) Propeller; (*b*) tube-axial; (*c*) vane-axial.

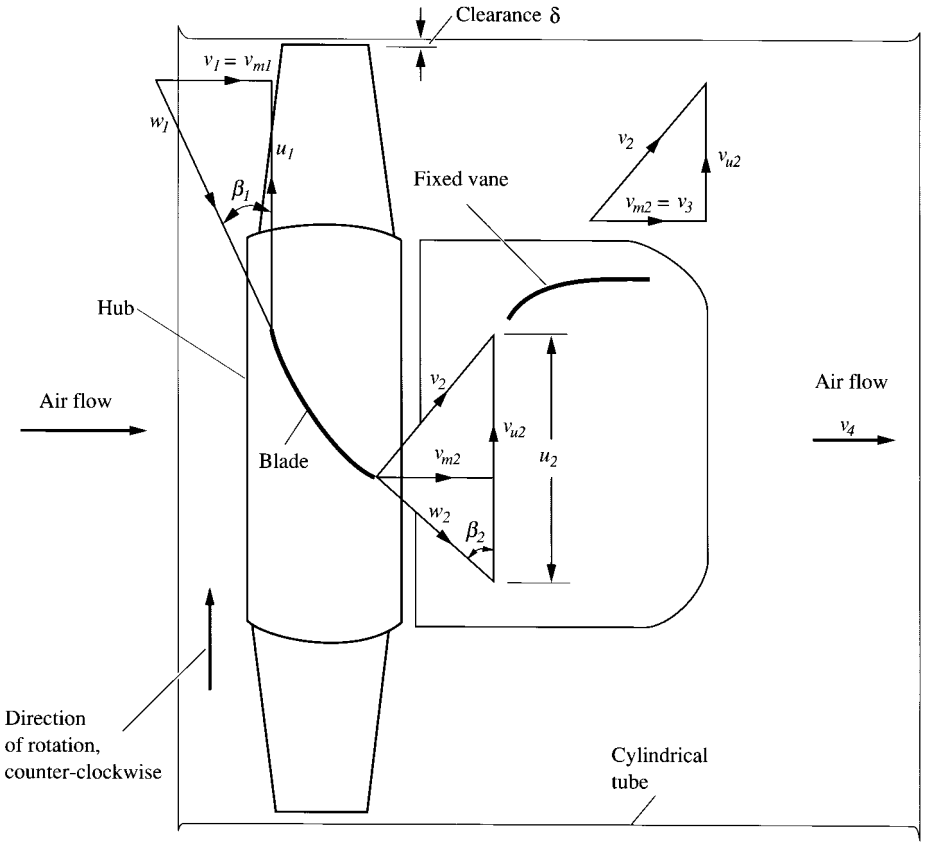


FIGURE 15.12 Velocity triangles for a vane-axial fan.

usually made of cast aluminum or aluminum alloy. The hub ratio R_{hub} is generally less than 0.3, and the clearance between the blade tip and the casing is significantly closer than in propeller fans.

In some tube-axial fans, the blade angle can be adjusted manually when the fan is not in operation. Thus, the volume flow and fan total pressure can be adjusted for tube-axial fans driven directly by motor.

Vane-Axial Fans. The impeller of a vane-axial fan has 8 to 16 blades, usually airfoil blades. The hub ratio is generally equal to or greater than 0.3 in order to increase the fan total pressure. Another important characteristic of vane-axial fans is the installation of fixed guide vanes downstream from the impeller, as shown in Figs. 15.11*c* and 15.12. These curved vanes are designed to remove swirl from the air, straighten the airflow, and convert a portion of the velocity pressure of the rotating airflow to static pressure. Sometimes guide vanes are also installed upstream from the impeller. Automatically controllable pitch (blade angle) for variable-air-volume system is one of the features of vane-axial fans.

Velocity Triangles

Consider a blade section of an axial fan, as shown in Fig. 15.12. Air enters the impeller with a velocity v_1, which is exactly equal to the axial velocity at the blade inlet v_{m1}. Because of the rotation

of the impeller, air leaves the blade at its outlet with a velocity v_2. For a blade section whose diameter at blade inlet equals the diameter at blade outlet, the peripheral velocities $u_1 = u_2$, and the axial velocities $v_{m1} = v_{m2}$. The total pressure rise Δp_t, in in. WG, as air flows through the impeller is given by the same equation as for a centrifugal fan when velocities are expressed in fpm:

$$\Delta p_t = 1.66 \times 10^{-6} \rho v_{u2} u_2$$

To design a blade such that the total pressure developed at various radii of the blade is constant, the blade angle near the hub of the same blade must be increased in order to provide a higher v_{u2} when the radius becomes smaller. A greater hub ratio $D_{\text{hub}}/D_{\text{bt}}$ results in a smaller variation of the blade angles of the same blade.

After air leaves the blade and before it enters the downstream fixed guide vanes, there is a rotational component v_{u2} in the direction of rotation. The function of the downstream guide vanes is to convert this dynamic pressure to static pressure with minimum energy loss.

The static pressure developed in an axial fan is the combined effect of the following:

- The corresponding drop in relative velocity pressure

$$C_2 \rho (w_1^2 - w_2^2)$$

- The conversion of rotating dynamic pressure to static pressure

$$C_2 \rho (v_2^2 - v_{m2}^2)$$

- The conversion of the difference in velocity pressures

$$C_2 \rho (v_3^2 - v_4^2)$$

where v_3, v_4 = mean axial velocities at cross-sectional areas with and without hub, respectively, fpm (m/s)

C_2 = constant equal to $1/(60^2 \times 2 \times 32.2 \times 5.192) = 8.3 \times 10^{-7}$

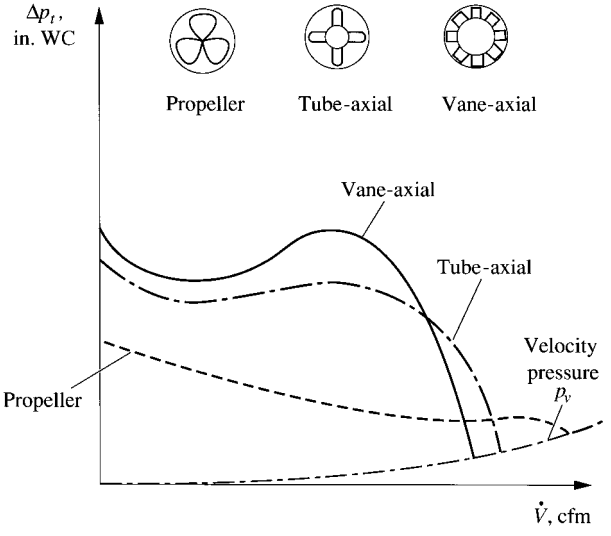


FIGURE 15.13 Pressure-volume Δp_t-$\dot{V}$ for axial fans with the same impeller diameter.

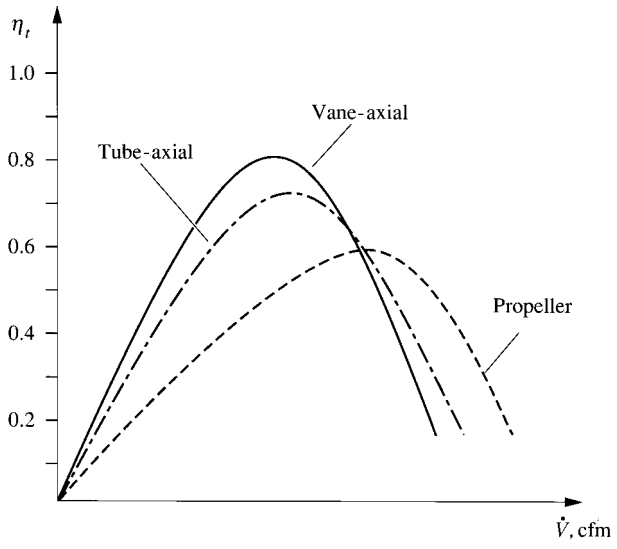


FIGURE 15.14 Total efficiency η_t-$\dot{V}$ curves for axial fans with the same impeller diameter.

Performance Curves

Figure 15.13 shows the pressure-volume flow Δp_t-$\dot{V}$ curves for various types of axial fans with impellers of the same diameter. Among these fans, the propeller fan has a low fan total pressure Δp_t and a high volume flow rate $\dot{V}$, whereas the tube-axial fan has a higher Δp_t and the vane-axial has the highest Δp_t.

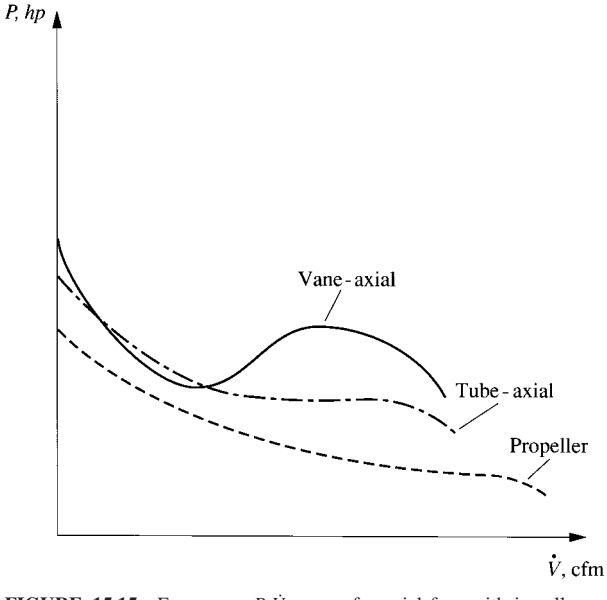


FIGURE 15.15 Fan power P-$\dot{V}$ curves for axial fans with impellers of the same diameter.

Figure 15.14 shows the total efficiency η_t-$\dot{V}$ curves for axial fans of the same diameter. Because of the installation of airfoil blades and the downstream guide vanes, vane-axial fans have the highest η_t of all axial fans. A well-designed vane-axial fan may have the same or even higher maximum η_t as a backward-curved airfoil centrifugal fan. In a vane-axial fan, it is important to use its velocity pressure, or convert it to static pressure by means of a diffuser. Tube-axial fans have lower η_t. Propeller fans have the lowest η_t, ranging between 0.4 and 0.55, because of their simple construction.

Figure 15.15 shows the fan power P-$\dot{V}$ curves for axial fans. It is important to see that all axial fans have their maximum fan power input at the shutoff or block tight condition. In this respect, they are quite different from the centrifugal fans, which have high fan power input at large volume flow rates. For axial fans, the greater the volume flow rate, the lower P, except in vane-axial fans, whose P-$\dot{V}$ curves are saddle-shaped. Because the P-$\dot{V}$ curves for vane-axial and tube-axial fans are nearly parallel to the Δp_t-$\dot{V}$ curves at the operating range, they are nonoverloading curves. Motors need not be sized according to the maximum fan power input as long as the axial fans will not operate at shutoff condition.

Figure 15.16 shows the performance curves of Δp_t-$\dot{V}$, P-$\dot{V}$, and total efficiency η_t for a typical vane-axial fan. The impeller is 43 in. (1.09 m) in diameter and rotates at a speed of 1750 rpm. From

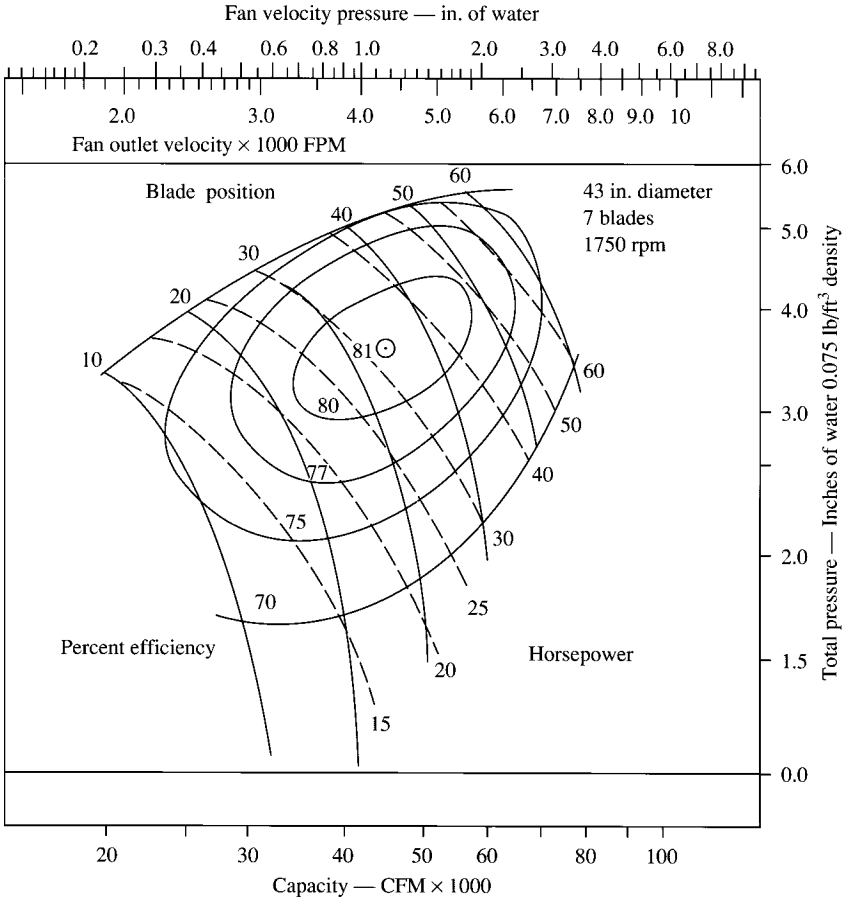


FIGURE 15.16 Controllable-pitch vane-axial fan performance curves. (*Source: Buffalo Forge Company. Reprinted with permission.*)

the performance curves, it can be seen that the greater the blade pitch or blade angle, the higher the fan total pressure Δp_t and the volume flow rate.

Reverse Operation. If the rotation of an axial fan is reversed, the direction of airflow is reversed. Propeller and tube-axial fans without guide vanes deliver about 60 to 70 percent of the volume flow rate in the original forward direction when their rotations are reversed. Vane-axial fans with guide vanes either downstream or upstream from the impeller do not perform efficiently when they rotate in reverse.

Tip Clearance and Number of Blades. For an axial fan, the size of the gap between the blade tip and the casing δ, as shown in Fig. 15.12, or the ratio of this gap to the outside diameter of the impeller δ/D_{imp}, has a definite effect on fan efficiency and sound power level. Here D_{imp} indicates the diameter of the impeller. Clearance δ and diameter D_{imp} must be expressed in the same units. Generally, δ/D_{imp} should be equal to or less than 0.0025. When the gap becomes greater, the amount of air leakage through the gap increases. Various tests have shown that fan total efficiency may drop from 0.7 to 0.68 if δ/D_{imp} increases from 0.0025 to 0.01.

Theoretically, the more blades in an axial fan, the better the guidance of air. At the same time, the friction between the fluid elements and the surface of the blade increases. An axial fan with a lot of blades always produces a higher fan total pressure than a fan with fewer blades at the same volume flow rate, if other conditions are the same.

15.4 FAN CAPACITY MODULATION

In a variable-air-volume (VAV) air system, more than 90 percent of its running hours operate at an airflow less than the design volume flow rate. Modulating the fan capacity by providing a new Δp_t-$\dot{V}$ curve with lower volume flow and fan total pressure not only corresponds with load reduction in the conditioned space, but also allows significant energy savings at part-load operation.

Four types of fan capacity modulation are widely used in air systems: fan speed, inlet vanes, fan inlet cones, and blade pitch modulation.

Fan Speed Modulation with Adjustable-Frequency, Variable-Speed Drives

The peripheral velocity at blade tip u_2 affects both the tangential component v_{u2} and radial component v_{m2} of the airflow and, therefore, the $\dot{V}$ and Δp_t of the fan. The variation of fan speed produces a family of similar Δp_t-$\dot{V}$ curves, as shown in Fig. 15.17*a*.

Most widely used, variable-speed drives (VSDs) for fans and pumps are adjustable-frequency VSDs. These VSDs modulate the speed of the ac motor by supplying a variable-frequency and variable-voltage power source. The synchronous speed of the motor n_m, in rpm, can be calculated as

$$n_m = \frac{120f}{N_p} \tag{15.20}$$

where f = frequency of applied power, Hz
N_p = number of poles

If the frequency varies, the motor speed changes accordingly. The voltage must be reduced with the frequency, to maintain a specific voltage-to-frequency ratio, in order to follow the decreasing inductive reactance of the motor.

The incoming three-phase ac power is rectified into dc supply through a rectifier. Then the dc power supply is shaped into a pseudo-sine wave of predetermined frequency by an inverter, as shown in Fig. 15.17*b*. The advantage of an ac inverter is that its speed can be reduced to one-tenth (or even less) of its original speed. Through the change of frequency, the speed-torque curve of the motor can

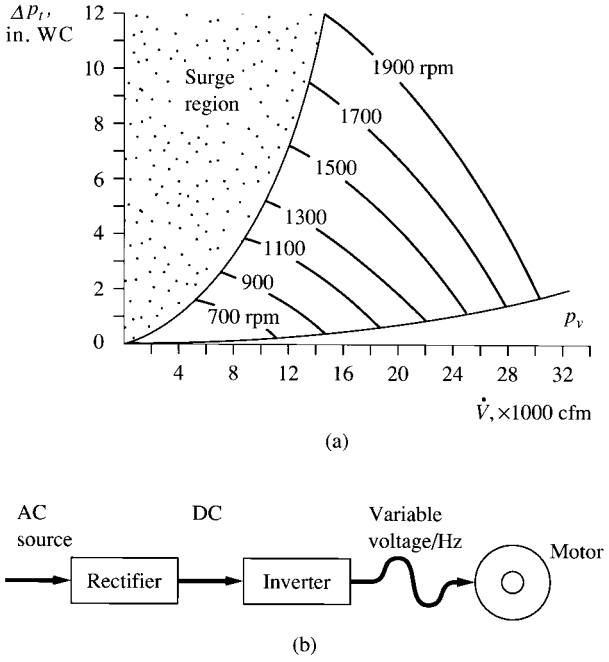


FIGURE 15.17 AC inverter and Δp_t-$\dot{V}$ curves of an airfoil fan at various speeds: (*a*) curves at various speeds; (*b*) ac inverter.

be adjusted so that it is always operated near its maximum efficiency. The electronics themselves consume little energy. They are usually about 97 to 98 percent efficient. However, because of the ac inverter, motor efficiency degrades 5 to 7 percent, so the total energy loss is about 7 to 8 percent.

The pulse-width-modulated (PWM) inverter is the predominate design of ac inverters used in HVAC&R systems. A PWM inverter operates from a fixed dc bus. Varying the timing on thyristors or transistors produces pulsed output. A logic circuit and software control the switching to provide the required variable voltage and frequency for a specific variable speed in an ac induction motor. The advanced power electronics, called insulated gate bipolar transistors, successfully provides high switchings with low energy lossses and a compact and quieter drive.

PWM inverters operate at a wide range of speeds and can also provide a "soft start," which brings the motor to the rated speed gradually to reduce abnormally high starting current. They are now often cost-effective for large fan motors in variable-air-volume systems and pump motors in variable-flow water systems. (They often have a simple payback of less than 3 years for motors with a capacity equal to and greater than 25 hp.)

Variable-speed drives using adjustable-frequency ac inverters have a good operating characteristic when they are used for centrifugal fans and pumps with impeller of backward-curved blades. Operation of forward-curved centrifugal fans may become unstable and more difficult to control because of their saddle-type fan characteristic curves.

Inlet Vanes Modulation

As with centrifugal compressors, inlet vanes in centrifugal fans are pivoted, movable vanes installed at the inlet of the impeller. These vanes are linked mechanically so that they can turn simultaneously around the axis, as shown in Fig. 15.18.

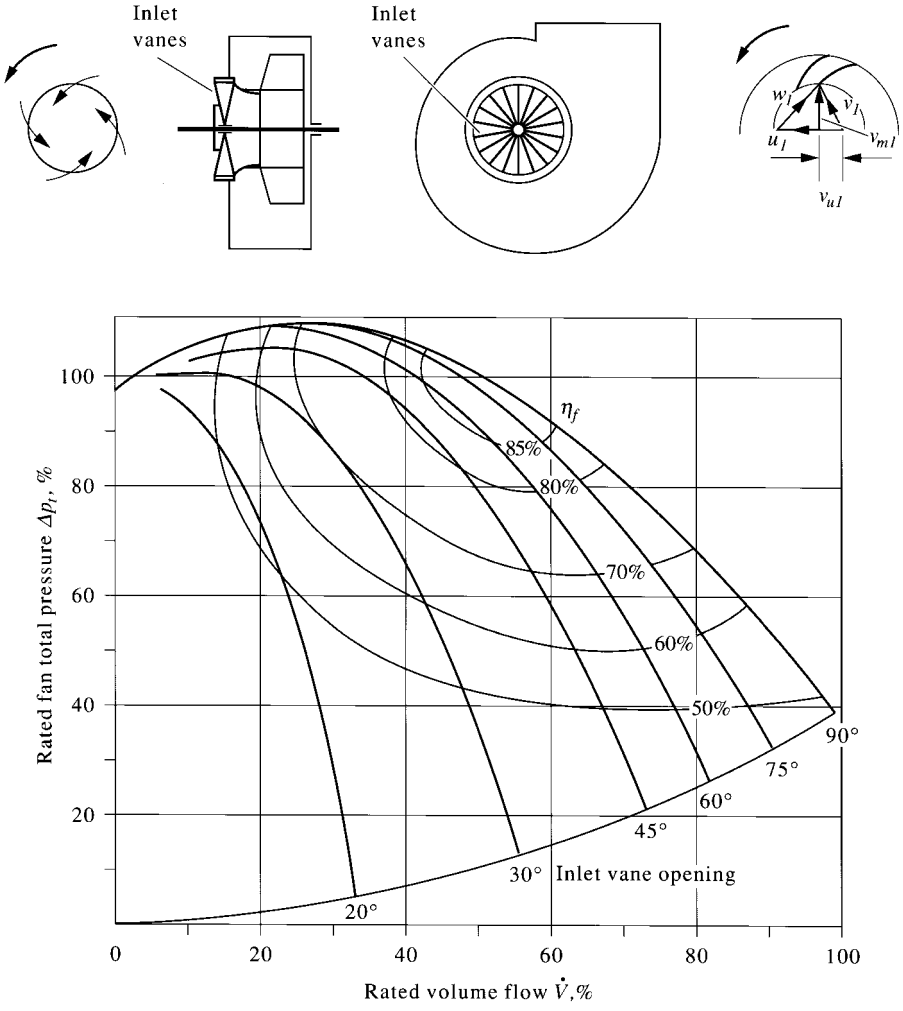


FIGURE 15.18 Inlet vane control of a backward-curved centrifugal fan.

Inlet vanes impart a spin on the airstream before it enters the impeller. From Eq. (15.15),

$$\Delta p_t = 1.66 \times 10^{-6} \rho (v_{u2} u_2 - v_{u1} u_1)$$

If the direction of the spin is the same as the direction of rotation of the impeller, the tangential component v_{u1} has a positive value. The fan total pressure thus developed is smaller than if air enters the impeller with a radial entry $v_{u1} = 0$. This produces a new set of Δp_t-$\dot{V}$ curves at various inlet vane opening angles with lower Δp_t and $\dot{V}$.

The fan total efficiency η_t decreases as the inlet vane opening angle decreases, as the air at inlet is further deviated from the shock-free condition and the inlet area is reduced. However, the reduction of shaft power from the decreases of Δp_t and $\dot{V}$ compensates for the effect of the drop in η_t. Energy savings are significant only when the inlet vanes are sufficiently near the eye of the impeller.

Inlet vanes reduce the inlet area. For large fans, this reduced inlet area causes a pressure loss of about 8 percent. For small fans, the additional pressure loss is far greater. If air velocity at the inlet v_i exceeds 5000 fpm (25 m/s), the fan total pressure is significantly reduced. Inlet guide vanes are usually not preferable for small backward-curved centrifugal fans. During the commissioning of an air-handling unit with VAV system, the available fan static pressure dropped from the specified 4 to only 2.5 in. WC (1000 to 625 Pa) when the air velocity in the fan inlet of the backward-curved centrifugal fan installed with inlet vanes exceeded 5000 fpm (25 m/s).

A backward-curved centrifugal fan with airfoil blades and inlet vanes has a higher ratio of reduced volume flow power input to design volume flow power input P_{part}/P_{full} than a forward-curved centrifugal fan has. However, a backward-curved airfoil fan has a higher inherent fan total efficiency than a forward-curved fan. Therefore, for a VAV system, actual fan power input should be compared between an airfoil fan and a forward-curved fan at reduced volume flow operation (according to the manufacturer's data) before the fan selection.

Use of inlet vanes for fan capacity modulation is still widespread for centrifugal fans because of their lower installation cost. Because axial fans require maximum fan power input at shutoff condition, inlet vanes are not suitable for axial fans.

Inlet Cone Modulation

The inlet cone of a backward-inclined or airfoil centrifugal fan can be moved so that a portion of the impeller is inactive. Such a modulation results in a lower fan capacity and fan total pressure.

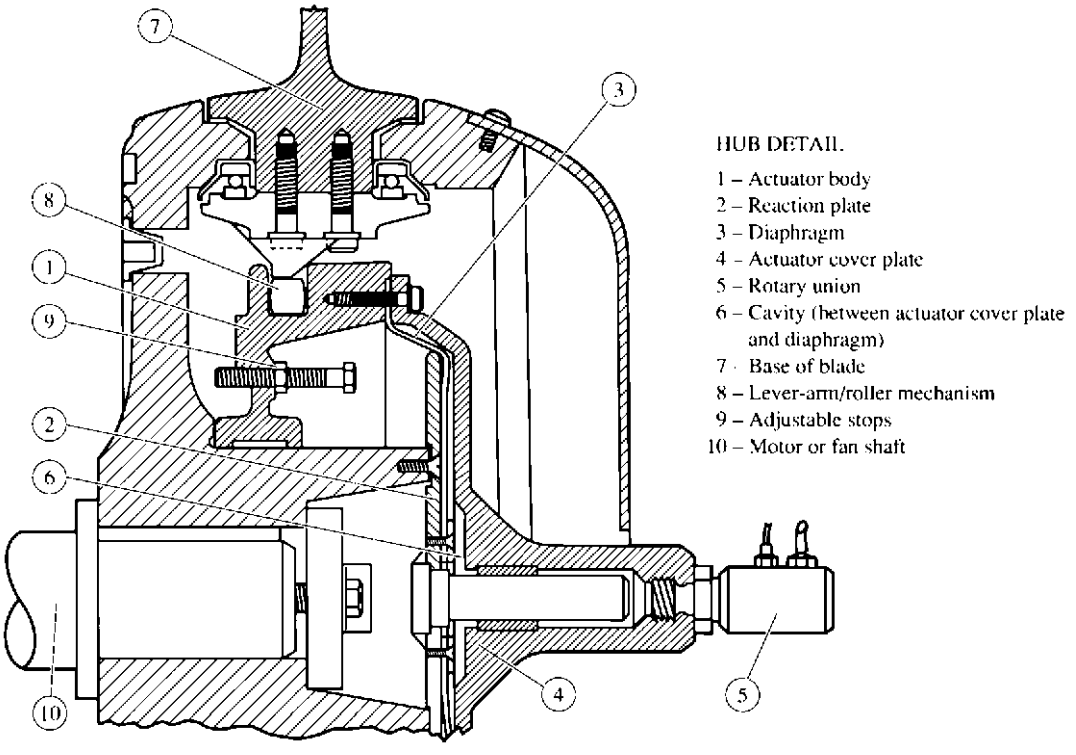

HUB DETAIL.
1 – Actuator body
2 – Reaction plate
3 – Diaphragm
4 – Actuator cover plate
5 – Rotary union
6 – Cavity (between actuator cover plate and diaphragm)
7 – Base of blade
8 – Lever-arm/roller mechanism
9 – Adjustable stops
10 – Motor or fan shaft

FIGURE 15.19 Controllable-pitch vane-axial fan blade pitch control mechanism.

Inlet cone modulation has a lower initial cost. Although its fan energy use is higher than that of variable-speed drive at part-load conditions, inlet cone modulation does not block the area of the inlet eye. Inlet cone modulation is used in rooftop packaged units in which airfoil centrifugal fans are installed.

Blade Pitch Modulation

From the velocity triangles shown in Fig. 15.12, for axial fans, the smaller the blade angle β_2, the lower v_{m2} and v_{u2} and, therefore, the lower $\dot{V}$ and Δp_t. Figure 15.16 shows the performance diagram of a typical controllable and adjustable-pitch vane-axial fan in which there is a corresponding Δp_t-$\dot{V}$ curve for each blade angle. For axial fans, blade pitch fan capacity modulation can take place both during the rotation of the fan impeller (controllable pitch) or at rest (adjustable pitch).

Figure 15.19 shows the blade pitch control mechanism of this controllable-pitch vane-axial fan. The blades are activated internally through level arms and a roller mechanism by pneumatic pressure against a flexible diaphragm. The control mechanism is direct-acting; i.e., the blade angle increases as the controlled pneumatic air pressure is increased. If the pneumatic pressure fails when the fan is operating, blades return to their minimum blade pitch. A pilot positioner, located outside the fan, ensures that the controlled blade pitch has the minimum deviation from the required one. The pneumatic pressure used to actuate the blade pitch is 25 psig (172 kPag) while the pressure used to activate the pilot positioner is 3 to 15 psig (21 to 103 kPag).

Of these four fan capacity modulating methods, inlet vane modulation is lower in installation cost and provides significantly lower energy savings than fan speed and blade pitch modulations at reduced airflow. Fan speed and blade pitch modulation also reduce fan noise and increase the fan's life.

15.5 FAN SURGE AND STALL

Fan Surge

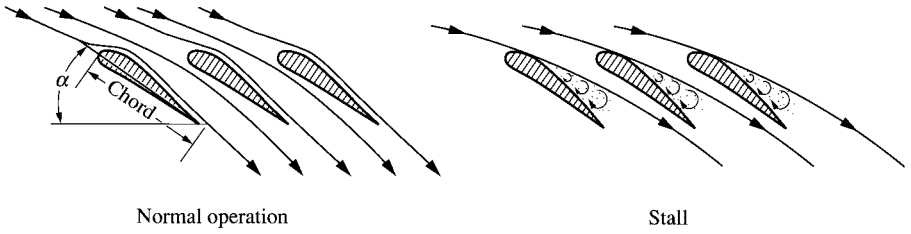
Fan surge (or stall) occurs when air volume flow through a centrifugal fan is not sufficient to sustain the static pressure difference between the discharge and suction sides of the fan. Air surges back through the fan impeller, and its discharge pressure is reduced momentarily. The surge of air enables the fan to resume its original static pressure. Fan surge will repeat if the air volume flow is not sufficient to sustain normal operation.

The volume flow and pressure fluctuations caused by fan surge create noise and vibration. The surge region of a backward-curved centrifugal fan is usually greater than that of a forward-curved fan. No fan should operate in the surge region, shown in Fig. 15.17. A large fan in surge is intolerable. Fan room walls have been cracked from the vibration of the ducts during a large fan surge.

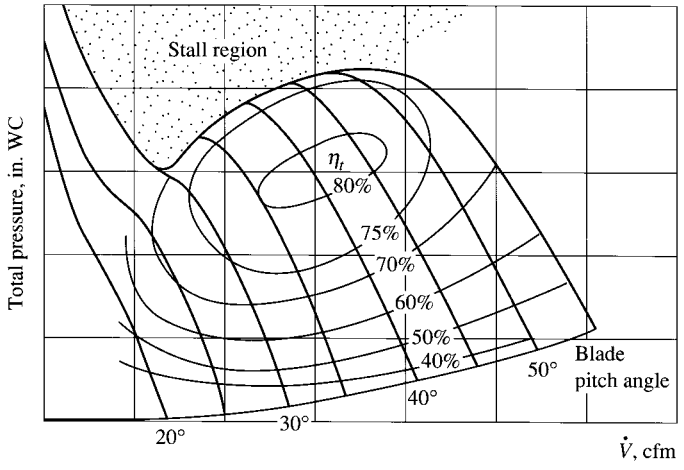
Some manufacturers show fan surge regions on their fan performance diagram in catalogs. If data and curves are not presented in the performance table or diagram, they are probably in or near the fan surge region.

Fan Stall

Stall is an operating phenomenon that often occurs in axial fans. For airfoil blades in an axial fan, the angle of attack α is the angle between the motion of the blades and the chord line, as shown in Fig. 15.20*a*. At normal operation, airflow around the streamlined airfoil blades causes a pressure difference between the top and bottom surfaces of the blades and produces lift. Stall occurs when



(a)



(b)

FIGURE 15.20 Stall and stall region of an axial fan: (*a*) normal operation and stall; (*b*) stall region of an axial fan.

the smooth airflow suddenly breaks and the pressure difference across the airfoil blades decreases. The axial fan loses its pressure capability drastically.

The stall region of an axial fan is generally located at the upper part of the fan performance diagram beyond the region of highest fan total efficiency, as shown in Fig. 15.20*b*. For each blade pitch angle, there is a corresponding stall region line. Usually, stall region is also the region in which no performance curves are given.

15.6 FAN CONSTRUCTION AND ARRANGEMENTS

Size and Class Standards

The Air Movement Control Association (AMCA) has established standards concerning fan size and classification. The size standard specifies the diameter of the impeller for a given fan size. There are 25 sizes, from 12.25 to 132.25 in. (311 to 3360 mm) in diameter. Each size is approximately 10 percent greater in diameter than the next-smaller size.

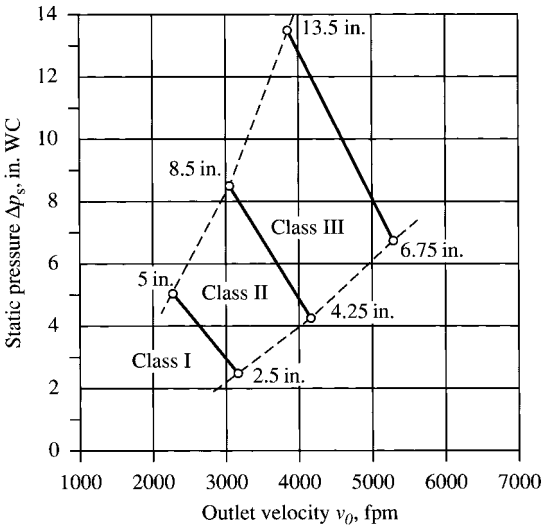


FIGURE 15.21 Class standards of fans.

Class standard specifies the construction of the fans as heavy, medium, or light structure. Construction is closely related to the static pressure developed Δp_s (in in. WC) by the fan and the fan outlet velocity v_o (in fpm or m/s), as shown in Fig. 15.21. Two classes of fans are widely used in HVAC&R applications: class I and class II. As shown in Fig. 15.21, a class I fan must provide 5 in. WC (1250 Pa) static pressure at an outlet velocity of 2300 fpm (11.5 m/s) and a static pressure of 2.5 in. WC (625 Pa) at an outlet velocity of 3200 fpm (16 m/s). A class II fan must provide 8.5 in. WC (925 Pa) static pressure at an outlet velocity of 3000 fpm (15 m/s) and 4.25 in. WC (1063 Pa) static pressure at 4175 fpm (20.9 m/s).

Fan Width and Inlets

For centrifugal fans, air velocity at the fan inlet v_i should be optimal. A lower v_i means a large inlet. A higher v_i results in a greater energy loss and therefore a lower fan total efficiency. A suitable relationship is $v_i/u_2 = 0.35$ to 0.4.

A single-width single-inlet (SWSI) centrifugal fan has a single inlet cone connected to the eye of the impeller, as shown in Fig. 15.22*a*. A double-width double-inlet (DWDI) centrifugal fan has a double-width impeller and provides double inlets from both sides, as shown in Fig. 15.22*b*. In such an arrangement, v_i can still be maintained at an optimal value, and the volume flow rate is approximately doubled. Because of the interaction of the two airstreams in the impeller and the scroll, the fan efficiency of a DWDI fan may be 1 to 2 percent lower than that of an SWSI fan.

Drive Arrangements and Direction of Discharge

Fans can be driven by motor directly or through V-belt and sheaves. In a belt-drive arrangement, fan speed can be changed by varying the diameter of the sheaves. On the other hand, belt drive requires 3 to 5 percent more energy input than direct drive.

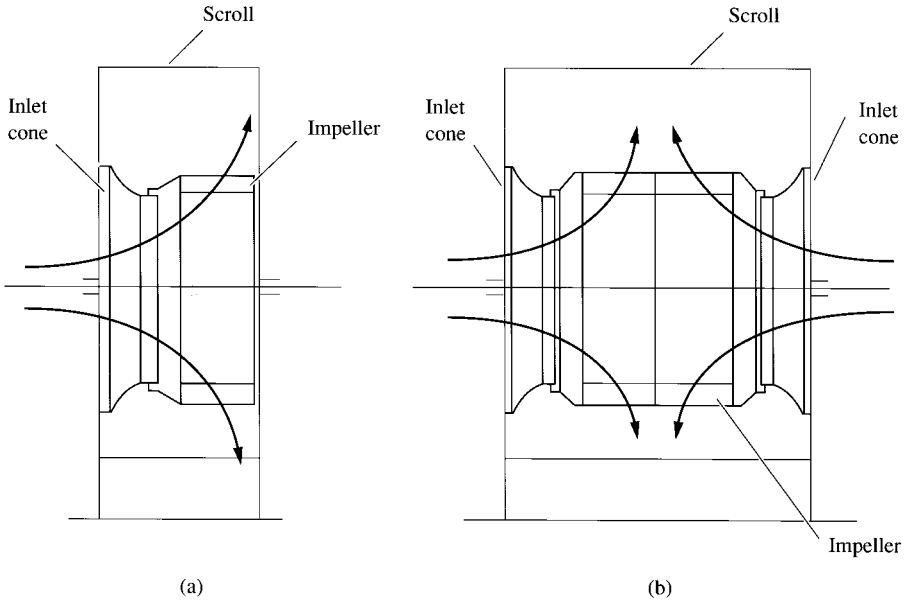


FIGURE 15.22 Single- and double-width centrifugal fans: (*a*) single-width single-inlet (SWSI); (*b*) double-width double-inlet (DWDI).

Drive arrangements include the location of bearing and sometimes the position of the motor. For centrifugal fans, there are eight standard drive arrangements, as shown in Fig. 15.23. There are only two drive arrangements, 4 and 9, for axial fans. Arrangements 1, 2, 3, 7, and 8 can include either belt drive or direct drive. Arrangements 4 is for direct drive only. Arrangements 9 and 10 are for belt drive only. However, in arrangement 9 for centrifugal fans, the motor is located outside the base, whereas in arrangement 10, the motor is located inside the base. This arrangement is often used with a weatherproof hood for outdoor installation.

Motor location is always specified as W, X, Y, or Z position, facing the fan drive side. The direction of rotation and discharge position of centrifugal fans are specified from the drive side of the fan, as shown in Fig. 15.24. In single-inlet fans, the drive side is always opposite the fan inlet, regardless of the actual position of the fan drive.

High-Temperature Fans

When fans are operated in high-temperature airstreams, the yield strength of the fan structure may decrease. Operating temperatures are usually classified as −20 to 200°F (−29 to 93°C) normal or standard fan construction, 201 to 300°F (94 to 150°C), 301 to 400°F (151 to 204°C), 401 to 750°F (205 to 400°C).

The rpm limits on classes I and II must be multiplied by a speed factor 0.9 when the operating temperature is 401 to 500°F (201 to 260°C). A lower speed factor should be used at higher temperatures. Cooling of the fan bearings is a significant problem. Generally, bearings should be kept out of the airstream except when operating at room temperature. In many antifriction bearings, grease can be used up to 200°F (93°C). Oil bearings or high-temperature grease is required for temperatures higher than 300°F (150°C). Special coolant should be applied to the shaft and bearings at higher temperatures. High-temperature aluminum paint can protect surface finishes.

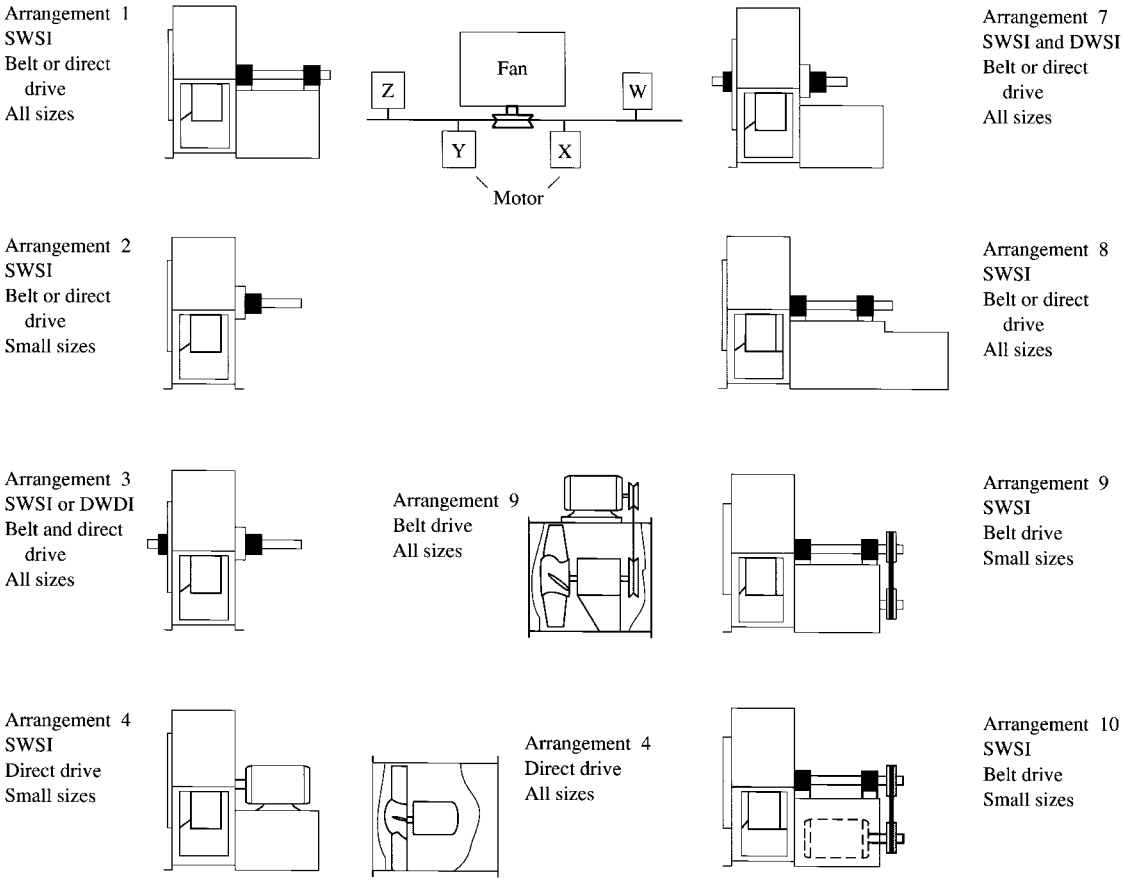


FIGURE 15.23 Drive arrangements for centrifugal and axial fans and motor positions.

Spark-Resistant Construction

AMCA specifies three types of spark-resistant fan construction:

- Type A construction requires that all parts of the fan in contact with the air be made of nonferrous metal.
- Type B construction requires that a fan have a nonferrous impeller and shaft rings.
- Type C construction requires that a shift of the impeller or shaft not allow the ferrous parts to rub or strike.

The bearings of spark-resistant construction fans should not be placed in the airstream, and all fan parts should be electrically grounded.

Safety Devices

When a fan is installed in a location where it is open to occupants or is accessible during operation, safety devices such as belt guards and protecting screens must be installed. Even a wire mesh installed at the outlet of a fan with a high air velocity may cause a pressure loss of 0.15 in. WC

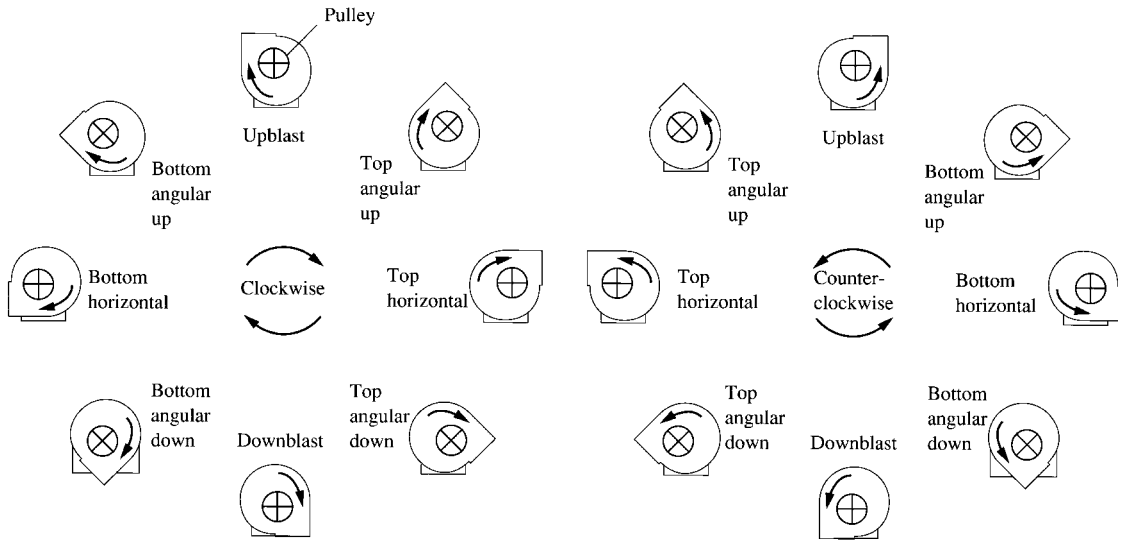


FIGURE 15.24 Direction of rotation and discharge position for centrifugal fans.

(37 Pa). Therefore, if the fan is covered by an enclosure or is accessible only when the fan is not in operation (during repair or maintenance), a safety device is not required.

15.7 FAN SELECTION

Selection of a fan for a given type of air system or mechanical ventilating system actually is done in two stages: selection of fan type and determination of fan size.

Conditions Clarified and Factors Considered

Before the selection, the following conditions must be clarified:

- Setting (in a commercial building to handle clean air at room temperature, or an industrial setting to handle dirty air)
- Special requirements (such as high-temperature operation or spark-resistant construction)
- Function (supply fan or a return fan in an air-handling unit, or supply or exhaust fan in a ventilating system)
- Characteristics of the air system (constant volume or variable air volume)
- Room *NC* curve
- Approximate annual operating hours
- Unit cost of energy at the specific location

During selection, the following factors should be considered:

- *Pressure-volume flow operating characteristics*. Selecting a fan to provide the required volume flow rate and total pressure loss for an air system or a ventilating system is of prime importance.

An undersized fan results in an uncontrolled indoor environment. An oversized fan wastes energy and money.

- *Fan capacity modulation*. A variable-air-volume system operates at a reduced volume flow rate during part-load operation. Effective and economical fan capacity modulation is an important factor that affects the operation of an air system.
- *Fan efficiency.* Fan efficiency is closely related to the energy consumption of an air system. Fans should be selected so that they can operate at high efficiency during as much of their operation time as possible.
- *Sound power level.* Most commercial and public buildings and many industrial applications need a quiet indoor environment. Fans are the major source of noise in an air system. Usually, the higher the fan total efficiency, the lower the sound power level of the selected fan. A fan with a low sound power level and sound power level at high frequencies is preferable. High-frequency sound is more easily attenuated than low-frequency sound.
- *Airflow direction.* In many applications, a straight-through or in-line flow occupies less space and simplifies layout.
- *Initial cost.* The initial cost of the fan modulation device, sound attenuator(s), and space occupied by a particular type of fan, in addition to the cost of the fan itself, should be considered.

Another important factor in fan selection is the fan inlet and outlet connections. Since the effects of these connections are closely related to the characteristics of the air system, they are covered in Chap. 21.

In an air system, a fan is usually operated with a duct system, which may also include other equipment and system components. A comprehensive and detailed analysis of fan operation must consider the characteristics of the duct system.

Estimated Fan Sound Power Level

The estimated sound power level of a fan L_w, in dB, can be calculated as

$$L_w = K_w + 10 \log \frac{\dot{V}}{\dot{V}_1} + 20 \log \frac{\Delta p_t}{\Delta p_{t1}} + C \tag{15.21}$$

where K_w = specific sound power level of a certain type of fan, dB
$\dot{V}$ = fan volume flow rate or capacity, cfm
$\dot{V}_1$ = 1 cfm
Δp_t = fan total pressure, in. WC
Δp_{t1} = 1 in. WC

In Eq. (15.21), C represents the correction for off-peak operation, i.e., fan operation in the region of the performance curve where fan efficiency is not maximum. For a fan operating at 90 to 100 percent of its maximum static efficiency, $C = 0$; between 85 and 89 percent, $C = 3$. For each decrease of 10 percent of fan maximum static efficiency up to 55 percent, the C value increases by 3 dB.

Table 15.1 lists the specific sound power level K_w and blade frequency increments BFI, in dB, of various types of fans. Here, BFI should be added to the sound power level at blade frequency f_{blade}, in Hz. Blade frequency can be calculated as

$$f_{\text{blade}} = \frac{\text{fan rpm} \times \text{number of blades}}{60} \tag{15.22}$$

A comparison of K_w values shows that the backward-curved centrifugal fan with airfoil blades has the lowest K_w value at various octave bands. The differences between K_w values at various

TABLE 15.1 Specific Sound Power Levels K_w for Typical Fans

	Octave bands								
Fan type	63	125	250	500	1K	2K	4K	8K	BFI
Centrifugal									
AF, BC, or BI wheel diameter									
Over 36 in.	40	40	39	34	30	23	19	17	3
Under 36 in.	45	45	43	39	34	28	24	19	3
Forward-curved									
All wheel diameters	53	53	43	36	36	31	26	21	2
Radial-bladed									
Low pressure									
(4 to 10 in. of water)	56	47	43	39	37	32	29	26	7
Medium pressure									
(6 to 15 in. of water)	58	54	45	42	38	33	29	26	8
High pressure									
(15 to 60 in. of water)	61	58	53	48	46	44	41	38	8
Axial fans									
Vane-axial									
Hub ratio 0.3 to 0.4	49	43	43	48	47	45	38	34	6
Hub ratio 0.4 to 0.6	49	43	46	43	41	36	30	28	6
Hub ratio 0.6 to 0.8	53	52	51	51	49	47	43	40	6
Tube-axial									
Over 40-in. wheel diameter	51	46	47	49	47	46	39	37	7
Under 40-in. wheel diameter	48	47	49	53	52	51	43	40	7
Propeller									
General ventilation	48	51	58	56	55	52	46	42	5

Note: Includes total sound power level in dB for both inlet and outlet. Values are for fans only—not packaged equipment.
AF - airfoil blades
BC - backward-curved blades
BI - backward-inclined blades
BFI - blade frequency increments
***Source*:** *ASHRAE Handbook 1991, HVAC Applications*. Reprinted with permission.

octave bands for axial fans are far smaller than those of centrifugal fans. For forward-curved and radial-bladed centrifugal fans, K_w values are higher in low frequencies and lower in high frequencies than vane-axial and tube-axial fans.

Comparison between Various Types of Fans

Table 15.2 compares the characteristics of various types of fans. The backward-curved centrifugal fan with airfoil blades has the highest fan total efficiency and the lowest specific sound power level, so it is still the most widely used type of large fan in commercial, institutional, and many industrial applications. The forward-curved centrifugal fan has a compact size, slower speed, and lighter weight per unit volume output. It is widely used in room air conditioners, fan coils, small air-handling units, and many packaged units. In recent years, vane-axial fans with controllable pitch, especially used as return fans, have more applications in commercial air systems than before. For exhaust systems that require a large volume flow rate and low fan total pressure, a propeller fan is often the best choice.

TABLE 15.2 Comparison between Various Types of Fans

	Backward, airfoil centrifugal fan	Forward-curved centrifugal fan	Vane-axial	Propeller fan
Fan total pressure Δp_{tf}	Higher Δp_t	Comparatively lower Δp_t	Higher Δp_t	Low Δp_t
Flow rate	All flow rates	Larger flow rate	All flow rates	Larger flow rate
Fan power input	Nonoverloading	Overloading	Nonoverloading	Nonoverloading
Fan modulation	Inlet vanes AC inverter	Inlet vanes AC inverter	Controllable pitch AC inverter	
Fan total efficiency	0.7 to 0.86	0.6 to 0.75	0.7 to 0.88	0.45 to 0.6
Sound power level	Lower, higher L_w at low frequencies	Medium, higher L_w at low frequencies	Medium, difference of L_w values is small at various Hz	Higher, higher L_w at high frequencies
Airflow direction	90° turn	90° turn	Parallel to axle	Parallel to axle
Volume and weight	Greater	Less	Greater	Medium volume and lower weight
Initial cost	Higher	Medium	Higher	Low
Applications	Large HVAC&R systems	Lower pressure, small HVAC&R systems	Large HVAC&R systems	Low-pressure, high-volume flow exhaust systems

Case Study—Selection of Fans

Select the appropriate type of fan for the following applications:

1. The weaving room of a textile mill
2. A convenience store
3. A high-rise office building

Solution

1. In the weaving room of a textile mill, the sound power level of the looms is far greater than the sound power level of the fan noise transmitted through the ductwork and supply and return grilles. Also the fan total pressure required for the air system used in the weaving room is usually less than 3 in. WC (750 Pa). Therefore, a tube-axial or vane-axial fan would be suitable.
2. A convenience store usually requires an air system with a volume flow rate less than 5000 cfm (2360 L/s). Usually, a small or a medium-size rooftop packaged unit is used. Forward-curved fan is usually used in this type of packaged unit.
3. A high-rise office building may use a central hydronic VAV system or a unitary packaged VAV system. Design criteria also vary considerably according to the location, occupancy, and design criteria of the building. A more detailed and comprehensive analysis must be made to determine whether backward-curved centrifugal fans with airfoil blades, forward-curved fans, vane-axial fans, or a combination of fans should be used.

15.8 COIL CHARACTERISTICS

Coils are indirect-contact heat exchangers that transfer heat between air and another medium (such as water, refrigerant, steam, or brine) for the purpose of heating, cooling, dehumidifying, or doing a

combination thereof. Chilled water and refrigerants used to cool and dehumidify moist air are called coolants. Coils consist of tubes and external fins arranged in rows along the airflow to provide greater surface contact. Coils are arranged in circuits to create a shorter path length. In a coil, water and refrigerant flow inside the tubes, and air flows over the outside surface of the tubes and fins.

Types of Coils

Coils can be classified into four categories according to the medium used.

Water Cooling Coil. A water cooling coil uses chilled water as the coolant inside the tubes. The chilled water cools or cools and dehumidifies the moist air that flows over the external surface of the tubes and fins, as shown in Fig. 15.25*a*. To maintain a higher rate of heat transfer, the air and water normally follow a counterflow arrangement; i.e., coldest air meets the coldest water and the warmest air meets the warmest water.

The water tubes are usually copper tubes of $\frac{1}{2}$- to $\frac{5}{8}$-in. (13- to 16-mm) diameter with a thickness of 0.01 to 0.02 in. (0.25 to 0.5 mm). They are spaced at a center-to-center distance of 0.75 to 1.25 in. (19 to 31 mm) longitudinally and 1 to 1.5 in. (25 to 38 mm) transversely. The tubes may be arranged along the airflows in 2, 3, 4, 6, or 8 rows, in either staggered or aligned form. The staggered arrangement provides better heat transfer and a higher air pressure drop. Chilled water coils are commonly rated at a pressure of 175 to 300 psig (1205 to 2070 kPag). Figure 15.26 shows the structure of a water cooling coil and a DX coil.

Direct-Expansion (DX) Coil. In a direct-expansion coil, the refrigerant (usually HCFC-22, HFC-134a, HFC-404A, HFC-410A, HFC-407A, or HFC-407C) is evaporated and expanded directly inside the tubes to cool and dehumidify the air flowing over it, and condensation occurs on the outer surface of the DX coil, as shown in Fig. 15.25*b*. This is why it is called a DX coil or wet coil. A DX coil acts as the evaporator in a refrigerating system. In a DX coil, coolant or the refrigerant is fed into a distributor and is evenly distributed to various tube circuits, which are made of copper tubes, typically 0.5 in. (13 mm) in diameter. Refrigerant distribution and loading to various circuits are critical to the performance of a DX coil. After evaporation, the vapor refrigerant is discharged from the header to the suction line. The characteristics and performance of a DX coil are covered in Sec. 10.2.

Water Heating Coil. The water heating coil shown in Fig. 15.25*c* is similar in construction to the water cooling coil. There are two main differences between them. Hot water, instead of chilled water, is used as the heating medium in a water heating coil. Also, there are fewer rows in the water heating coil than in the water cooling coil. Generally, only two-, three-, or four-row water heating coils are available on the market. Water heating coils are rated at pressures of 175 to 300 psig (1205 to 2070 kPag) at temperatures up to 250°F (120°C).

Steam Heating Coil. Steam heating coils use the latent heat of condensation released by steam inside the tubes to heat outside and recirculating air, as shown in Fig. 15.25*d*. In a standard steam heating coil, steam enters one end of the coil and condensate comes out the other end. For more even distribution of steam, a baffle plate is often installed just after the inlet.

In a steam heating coil, it is important that the coil core inside the casing expand and contract freely. The coil core is also pitched toward the return connection to facilitate drainage of the condensate. Steam heating coils usually have a rating of 100 to 200 psig (690 to 1380 kPag) at 400°F (205°C). Steam heating coil tubes are usually made of copper, steel, or stainless steel.

Fins

Fins are extended surfaces that are often called the coil's secondary surface (the outer surface of the tubes is called the primary surface of the coil). Fins are often made of aluminum, with a fin

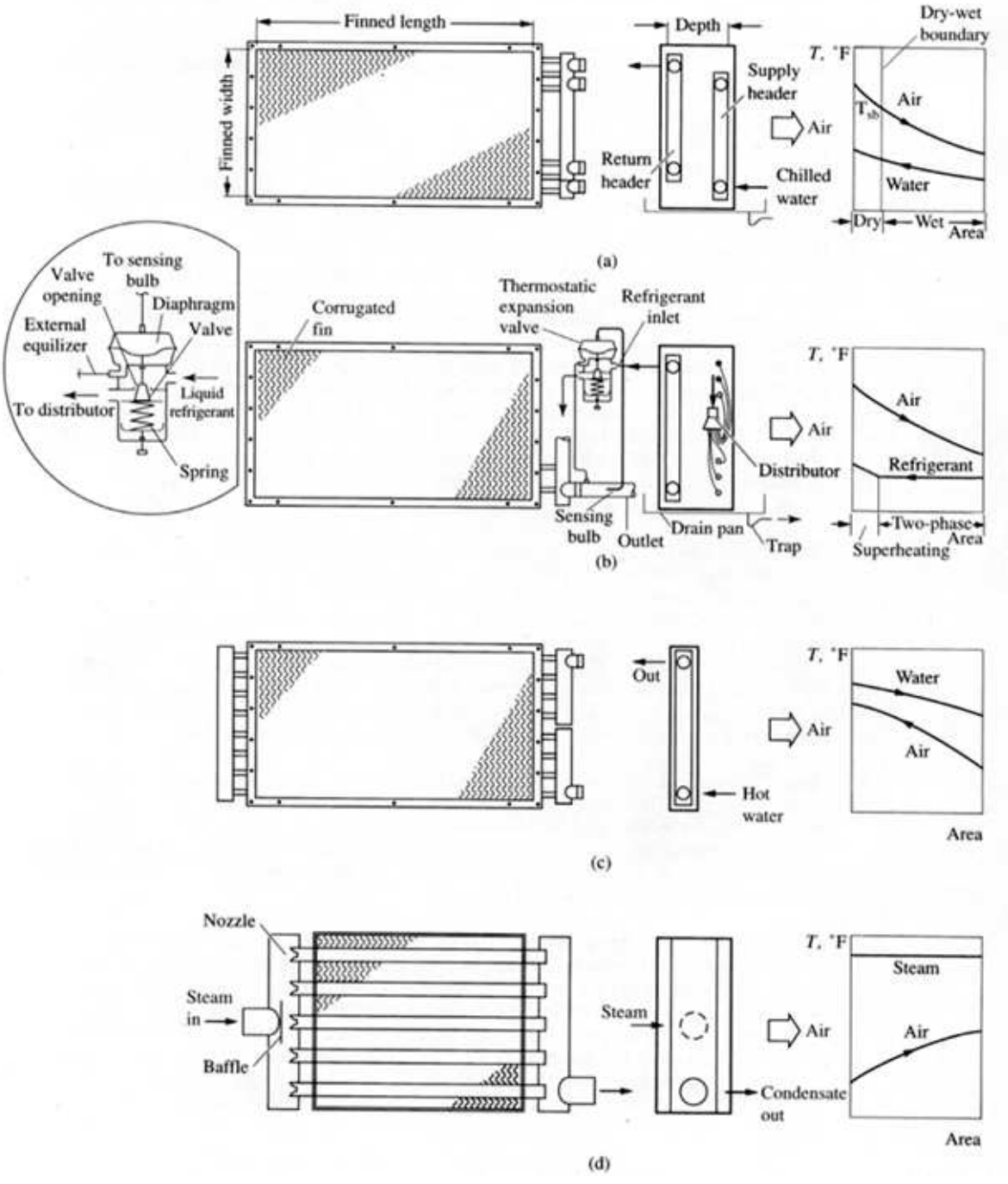


FIGURE 15.25 Types of coils: (*a*) Water cooling; (*b*) direct-expansion (DX); (*c*) water heating; (*d*) steam heating.

thickness F_t of 0.005 to 0.008 in. (0.13 to 0.2 mm). They can also be directly extruded, or shaved, from the parent tubes. Copper, steel, or stainless-steel fins are sometimes used.

Three types of fins are widely used today:

- Continuous plate fins and corrugated plate fins, as shown in Fig. 15.27*a* and *b*. Plate fins are usually made of aluminum.
- Smooth spiral and crimped spiral fins, used in many commercial and industrial applications, as shown in Fig. 15.27*c* and *d*. Smooth spiral fins may be extruded from the parent tubes.
- Spined pipes, as shown in Fig. 15.27*e*. Spined pipe is made by shaving the spines and lifting them from the parent tubes.

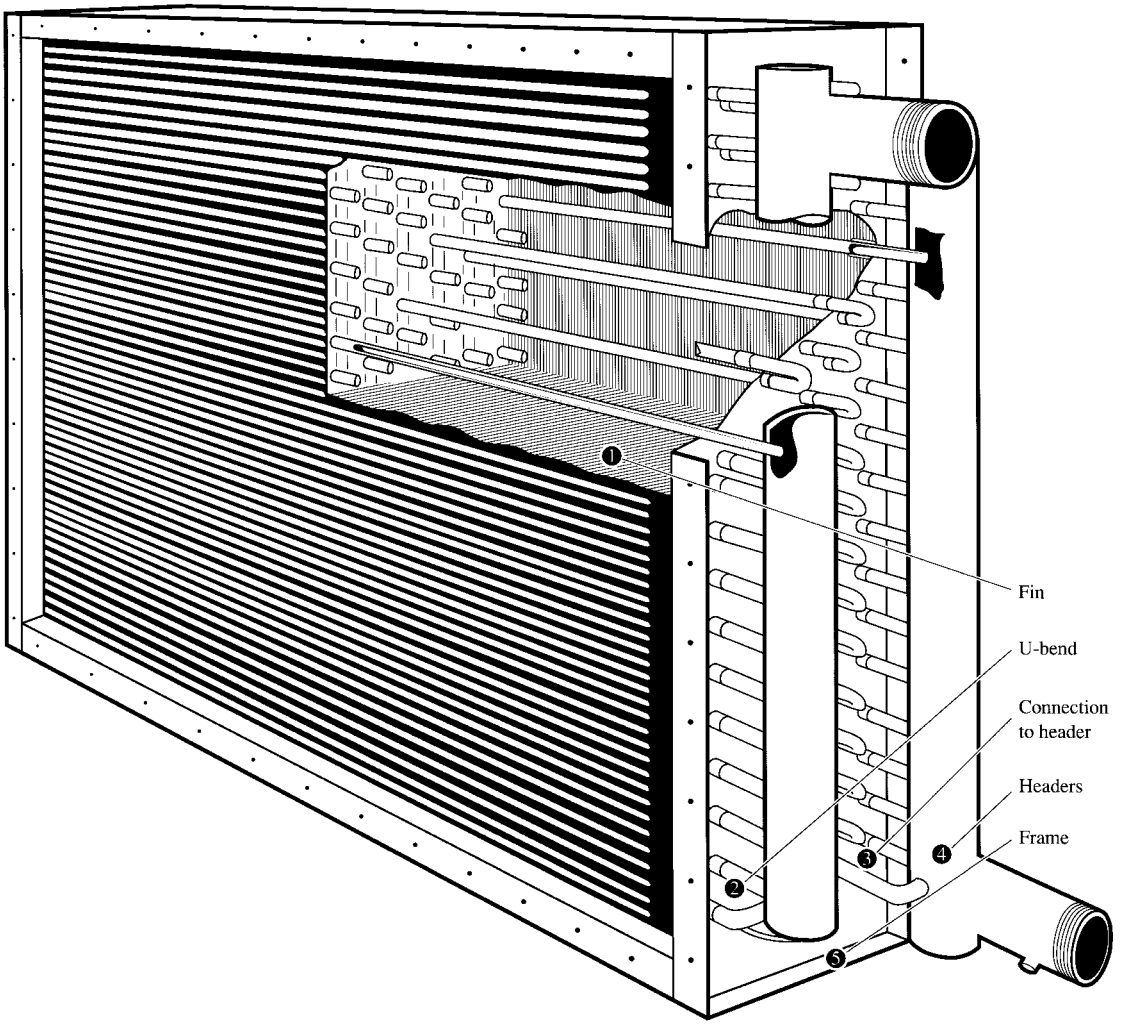


FIGURE 15.26*a* Structure of a water cooling coil. (*Source: York International Corporation. Reprinted with permission.*)

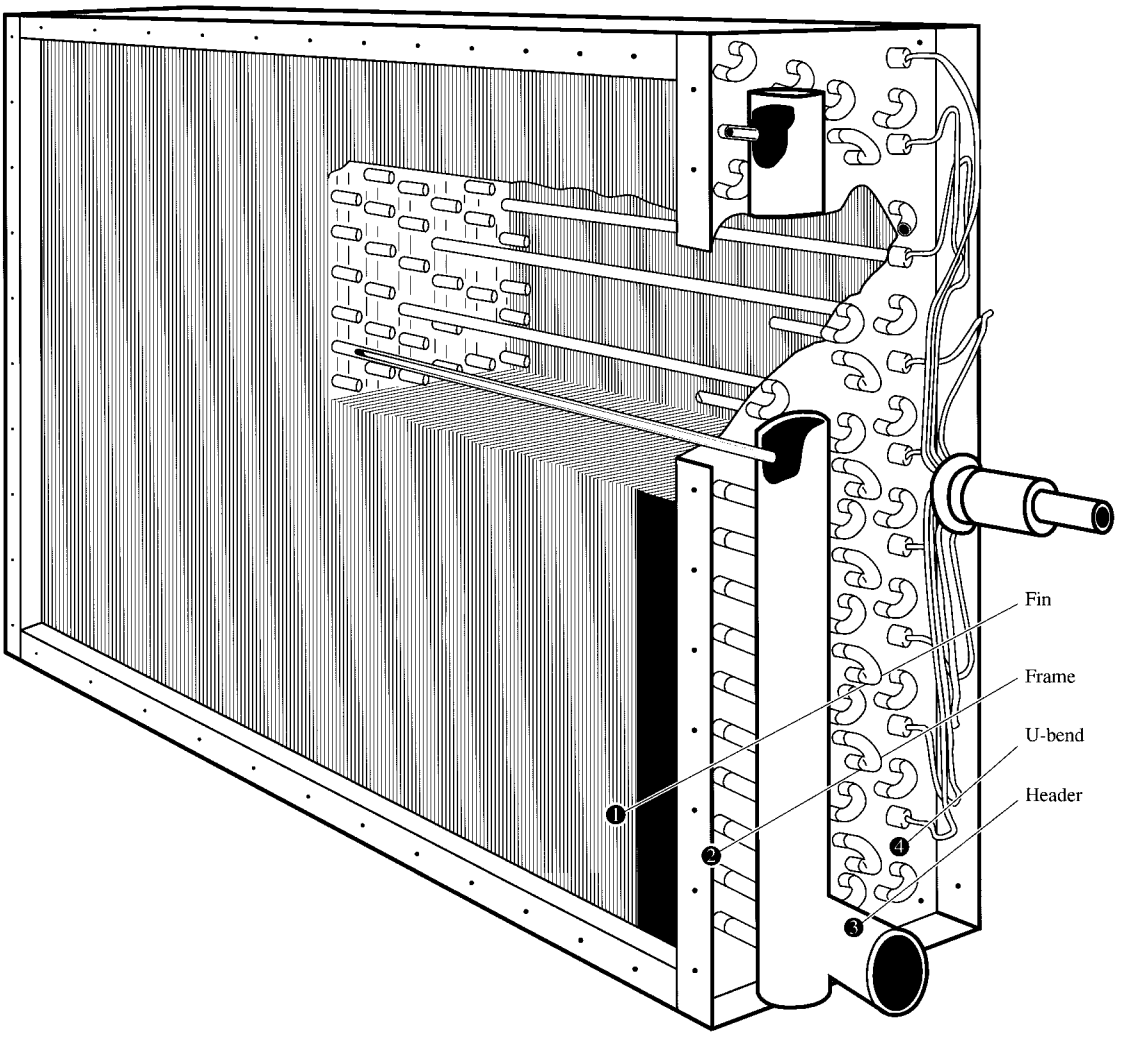


FIGURE 15.26*b* Structure of a DX coil. (*Source: York International Corporation. Reprinted with permission.*)

Corrugated and crimped fins create large turbulence and reduce the thickness of the boundary layer of the airstream. Consequently, their convective heat-transfer coefficient is increased significantly. However, corrugated and crimped fins also create a greater air-side pressure drop than plate fins do. Various manufacturers have developed rippled corrugated fins to accommodate the higher rate of heat transfer and the allowable air-side pressure drop.

Because spines are part of a spined pipe, they are firmly connected to the primary surface and there is no contact resistance between the spines and tube. According to Holtzapple and Carranza (1990), heat transfer of spine pipe per unit fan power energy is slightly better than that of plate fins with smaller tube spacing S_t and small tube diameters.

Fin spacing S_f, in in. (mm), is the distance between two fins. Fin density is usually expressed in fins per inch, and it often varies from 8 to 18 fins/in.(1.4- to 3-mm fin spacing) for coils used in air

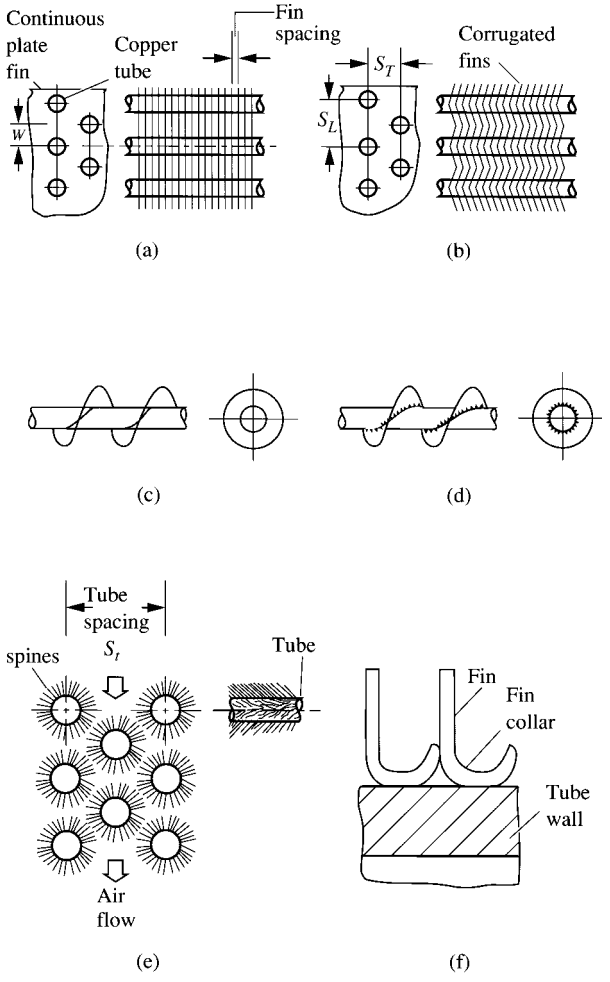


FIGURE 15.27 Types of fins: (*a*) Continuous plate fins; (*b*) corrugated plate fins; (*c*) smooth spiral fins; (*d*) crimped spiral fins; (*e*) spined pipe; (*f*) fin collar and tube bondings.

conditioning systems. Fin spacing has a definite effect on the extended surface area and, therefore, the capacity and air-side flow resistance of the coil.

For a finned-tube coil, finned width indicates the width of the continuous plate fin, as shown in Fig. 15.25*a*, and fin length is the length of the coil with plate fins.

Contact Conductance

Thermal contact conductance between the fins and tube in a coil h_{cc}, in $\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$ ($\text{W/m}^2 \cdot °\text{C}$), is the steady heat flux across the fin-tube bonding per °F (°C) temperature difference. Contact conductance is a function of contact pressure at the fin-tube bonding, fin thickness, fin density, tube diameter, and hardness of the tube. It directly affects the heat-transfer capacity of a finned-tube coil.

Contact between the fin and tube is a critical factor to the performance of the coil. Bonding between the fin and tube is usually formed by mechanically expanding the tube from inside, using a tool called a bullet to produce an interference between the fin collar and the tube, as shown in Fig. 15.27*f*. Interference I, in in. (mm), is the difference between the expanded tube diameter and the fin collar diameter before expanding. It can be also expressed as

$$I = D_t + 2W_{tk} - D_{\text{fc}} \tag{15.23}$$

where D_t = outside diameter of expanding tool, in. (mm)
W_{tk} = tube wall thickness, in. (mm)
D_{fc} = inner diameter of fin collar, in. (mm)

According to Wood et al. (1987), the interference of the coils from six manufacturers ranges from 0.003 to 0.0075 in. (0.08 to 0.2 mm).

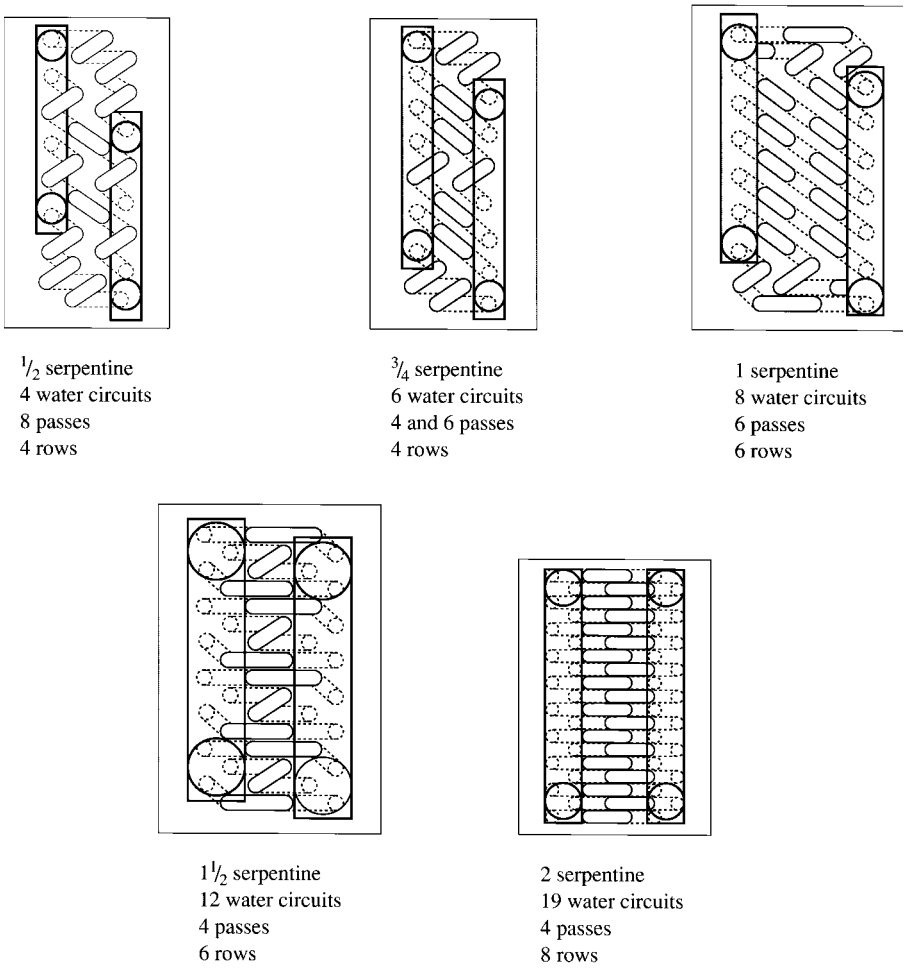


FIGURE 15.28 Water circuits for water cooling coils. (*Adapted from catalog of McQuay HVAC. Reprinted with permission of Snyder-General Corp.)*

Contact conductance is directly proportional to the magnitude of interference. Interference determines the quality of mechanical bonding during manufacturing. A higher interference means a better quality of mechanical bonding. Contact conductance increases as the thickness of the fin increases. It also increases as fin density (fins per inch) increases. Contact conductance decreases as tube diameter increases. It also decreases as the hardness of the copper tube increases in an aluminum fin and copper tube coil.

Water Circuits

In a water cooling or water heating coil, tube feeds or water circuits determine the number of water flow passages. The greater the finned width, the greater the number of tube feeds and thus the greater the number of flow passages. For two finned-tube coils of the same finned width, a difference in the number of tube feeds or water circuits means that the water flow rate, in gpm (L/s), number of passes, and pressure drop of the chilled water, in ft WC (Pa), inside the two coils are different. One pass means that water flows through the coil's finned length once.

The number of serpentines ($^1/_4$, $^1/_2$, $^3/_4$, 1, $1^1/_2$, or 2) of a water cooling or a water heating coil indicates its water flow arrangement. The greater the number of serpentines, the larger the total cross-sectional area of the water circuits and the greater the water volume flow rate. Figure 15.28 shows five water cooling coils, each specified by the number of serpentines, water circuits, passes, and rows, all made by the same manufacturer. In Fig. 15.28, $^1/_2$ serpentine means that at the first row, there are eight tubes across the finned width, but only four of them are tube feeds that are connected to the return header. For a full serpentine coil, all eight tubes in the first row are tube feeds and connect to the return header.

15.9 SENSIBLE COOLING AND SENSIBLE HEATING COILS—DRY COILS

Heat Transfer in a Sensible Cooling Process

In a sensible cooling process, the humidity ratio w is always constant. A sensible cooling process only exists when the outer surface temperature of the coil T_s, in °F (°C), is equal to or higher than the dew point of the entering air T''_{ae}, in °F (°C), that is, $T_s \geq T''_{ae}$. A sensible cooling process is indicated by a horizontal line toward the saturation curve on the psychrometric chart, as shown in Fig. 15.29.

If heat conduction through the coil framework is negligible, for a sensible cooling process occurring at steady state within a certain time interval, the rate of sensible heat transfer from the conditioned air Q_{cs} (in Btu/h or W) must be equal to the heat absorbed by the chilled water in the tubes. This relationship can be expressed as

$$\begin{aligned} Q_{cs} &= 60\dot{V}_a \rho_a c_{pa}(T_{ae} - T_{al}) \\ &= A_o U_o \Delta T_m = F_s A_a N_r U_o \Delta T_m \\ &= 60\dot{m}_w c_{pw}(T_{wl} - T_{we}) \end{aligned} \tag{15.24}$$

where $\dot{V}_a$ = volume flow rate of conditioned air, cfm [m/(60 s)]
ρ_a = air density, lb/ft^3 (kg/m^3)
T_{ae}, T_{al} = temperature of conditioning air entering and leaving sensible cooling coil, °F (°C)
c_{pa}, c_{pw} = specific heat of moist air and water, Btu/lb·°F (J/kg·°C)
A_o = total outside surface area of coil, ft^2 (m^2)
U_o = overall heat-transfer coefficient based on outside surface area of coil, Btu/h·ft^2·°F (W/m^2·°C)

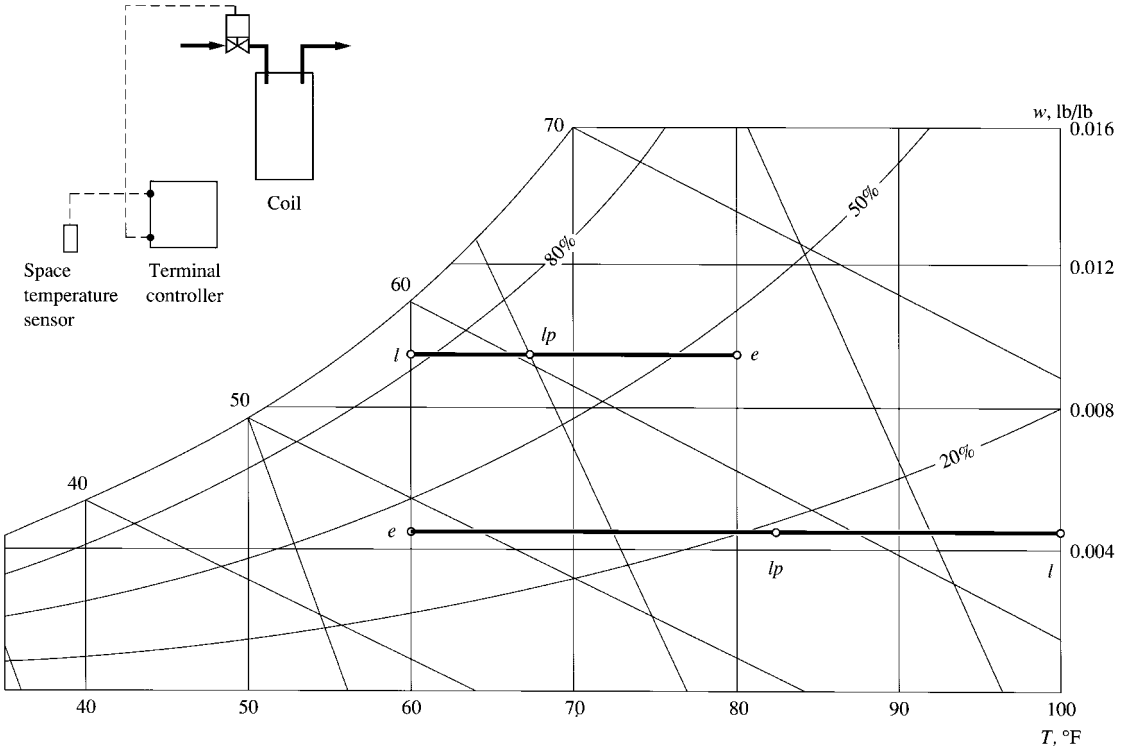


FIGURE 15.29 Dry coils at design and part-load operation.

F_s = coil core surface area parameter
A_a = face area of coil, ft^2 (m^2)
N_r = number of rows in coil
ΔT_m = log-mean temperature difference, °F (°C)
$\dot{m}_w$ = mass flow rate of chilled water, lb/min [kg/(60 s)]
T_{wl}, T_{we} = temperature of chilled water leaving and entering coil, °F (°C)

Because $\dot{m}_w = \dot{V}_w \rho_w$, here $\dot{V}_w$ indicates the volume flow rate of chilled water, in cfm [m/(60 s)], and ρ_w = water density, which can be estimated as 62.32 lb/ft^3. Also, 1 gal = 0.1337 ft^3 (3.785 L) and $c_{pw} = 1$, so

$$\begin{aligned} Q_{cs} &= 60\dot{V}_w \rho_w c_{pw}(T_{wl} - T_{we}) \\ &= 60 \times 0.1337 \dot{V}_{\text{gal}} \rho_w c_{pw}(T_{wl} - T_{we}) \\ &= 500\dot{V}_{\text{gal}}(T_{wl} - T_{we}) \end{aligned} \tag{15.25}$$

where $\dot{V}_{\text{gal}}$ = volume flow rate of chilled water, gpm (L/s)

Coil core surface area parameter F_s is defined as

$$F_s = \frac{A_o}{A_a N_r} \tag{15.26}$$

If the thermal resistance of copper tubes is ignored, the overall heat-transfer coefficient based on the outside surface area of the coil U_o and the inner surface area U_i, both in Btu/h·ft²·°F (W/m²·°C), can be calculated as

$$U_o = \frac{1}{1/(\eta_s h_o) + B/h_i}$$
$$U_i = \frac{1}{1/(\eta_s h_o B) + 1/h_i} \tag{15.27}$$

In Eq. (15.27), the finned surface efficiency η_s is given as

$$\eta_s = 1 - \frac{A_f}{A_o}(1 - \eta_f) \tag{15.28}$$

where h_o, h_i = heat-transfer coefficient of outer surface of coil (including fins and tubes) and inner surface of tubes, Btu/h·ft²·°F (W/m²·°C)
B = ratio of outside surface area A_o of coil to inner surface of tubes A_i, both in ft² (m²) (A_o/A_i)
A_f = area of fins, ft² (m²)
η_f = fin efficiency

Surface Heat-Transfer Coefficients

McQuiston (1981) developed the correlation between Chilton-Colburn j factors and parameter JP for dry coils as follows:

$$j_s = \left(\frac{h_o}{G_c c_{pa}}\right) \mathrm{Pr}^{2/3} = 0.00125 + 0.27\mathrm{JP} \tag{15.29}$$

$$\mathrm{JP} = \mathrm{Re}_D^{0.4}\left(\frac{A_o}{A_p}\right)^{-0.15}$$

$$\mathrm{Re}_D = \frac{G_c D_c}{\mu}$$

$$G_c = 60\,\rho v_{\min} \tag{15.30}$$

where G_c = mass velocity at minimum flow area, lb/h·ft² (kg/h·m²)
$v_{\min}$ = air velocity at minimum flow area, fpm (m/s)
A_p = primary surface area, or tube outer surface area, ft² (m²)
Pr = Prandtl number = $\mu c_p/k$; for air at temperature lower than 200°F, its value can be estimated as 0.71
μ = dynamic viscosity of fluid, lb/ft·h (kg/m·s)
c_p = specific heat of fluid, Btu/lb·°F (J/kg·°C)
k = thermal conductivity of fluid, Btu/h·ft·°F (W/m·°C)
Re_D = Reynolds number based on diameter D

The outer surface heat-transfer coefficient h_o calculated from Eq. (15.29) is for smooth plate fins. For corrugated fins, h_o should be multiplied by a factor F_{cor} of 1.1 to 1.25 to account for the increase in turbulence and rate of heat transfer.

For hot and chilled water at turbulent flow inside the tubes, the inner surface heat-transfer coefficient h_i, in Btu/h·ft²·°F (W/m²·°C), can be calculated by the Dittus-Boelter equation:

$$\frac{h_i D_i}{k_w} = 0.023\ \mathrm{Re}_D^{0.8}\ \mathrm{Pr}^n \tag{15.31}$$

where D_i = inner diameter of tubes, ft (m)
k_w = thermal conductivity of water, Btu/h·ft·°F (W/m·°C)

When the tube wall temperature $T_t > T_{\text{bulk}}$, $n = 0.4$; and when $T_t < T_{\text{bulk}}$, $n = 0.3$. Here T_{bulk} indicates the bulky water temperature beyond the boundary layer. All fluid properties in Re_D and Pr should be calculated at a mean temperature $T_m = (T_t + T_{\text{bulk}})/2$.

Fin Efficiency

Because the fin temperature is not uniform, fin efficiency η_f is used to calculate the rate of heat transfer of a heat exchanger with fins. Fin efficiency is defined as

$$\eta_f = \frac{\text{actual heat transfer}}{\text{heat transfer when fins are at base temperature}} \tag{15.32}$$

For a finned-tube coil with rectangular plate fins, fin efficiency can be calculated as recommended in *ASHRAE Handbook 1981*, *Fundamentals*,

$$\varphi_{f,\max} = \frac{R_{f,\max} F_t k_f}{W^2} \tag{15.33}$$

$$R_f = \frac{1}{h_o}\frac{1}{\eta_f - 1} \tag{15.34}$$

where $\varphi_{f,\max}$ = maximum fin resistance number, dimensionless
R_f, $R_{f,\max}$ = R value and maximum R value of fin, h·ft²·°F/Btu (m²·°C/W)
F_t = fin thickness, ft (m)
k_f = thermal conductivity of fin material, Btu/h·ft·°F (W/m·°C)

In Eq. (15.33), W represents a width dimension, as shown in Fig. 15.27*a*, in ft (m).

Figure 15.30 shows the relationship between $\varphi_{f,\max}$ and the ratio W/r_o at various S_L/W values. Here r_o indicates the outer radius of the tube, and S_L is the longitudinal spacing of the tubes, as shown in Fig. 15.27*b*. By assuming that $\varphi_f/\varphi_{f,\max} = 0.9$ and then $R_f/R_{f,\max} = 0.9$, fin efficiency η_f can be determined.

Dry Coil Effectiveness

As with the DX coil (wet coil) discussed in Sec.10.2, the dry coil effectiveness ε_{dry} can be calculated as

$$\varepsilon_{\text{dry}} = \frac{C_a(T_{ae} - T_{al})}{C_{\min}(T_{ae} - T_{we})} = \frac{C_w(T_{wl} - T_{we})}{C_{\min}(T_{ae} - T_{we})} \tag{15.35}$$

The heat capacity rate of the hot fluid (moist air) C_a, in Btu/h·°F (W/°C), in a sensible cooling coil is

$$C_a = 60\dot{m}_a c_{pa} = 60\dot{V}_a \rho_a c_{pa} \tag{15.36}$$

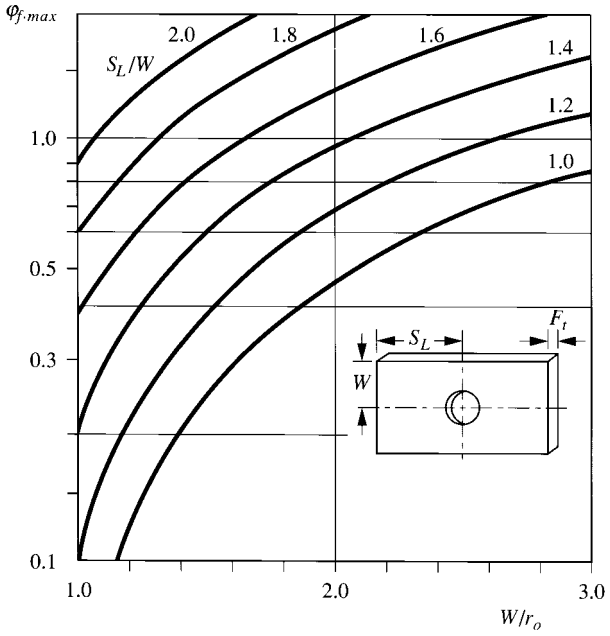


FIGURE 15.30 Maximum fin resistance number $\varphi_{f\max}$ of rectangular fins. (*Source: ASHRAE Handbook 1989, Fundamentals. Reprinted with permission.*)

The heat capacity of the cold fluid (chilled water) C_w, in Btu/h·°F (W/°C), is

$$C_w = 60\dot{m}_w c_{pw} = 500\dot{V}_{\text{gal}} c_{pw} \tag{15.37}$$

In Eq. (15.35), $C_{\min}$ is the smaller value of C_a and C_w, in Btu/h·°F (W/°C).

In a sensible cooling coil using chilled water as coolant, $\dot{m}_a c_{pa}$ is usually smaller than $\dot{m}_w c_{pw}$. Then

$$\varepsilon_{\text{dry}} = \frac{T_{ae} - T_{al}}{T_{ae} - T_{we}} \tag{15.38}$$

In Eq. (15.38), T_{al} is usually the required off-coil (leaving coil) temperature during the design of a sensible cooling system. Most finned coils are actually a combination of counterflow and crossflow arrangements. Because the effectiveness of a finned coil with a counterflow arrangement is nearly the same as that of a crossflow arrangement, the effectiveness can be calculated as

$$\varepsilon_{\text{dry}} = \frac{1 - \exp[\text{-NTU}(1 - C)]}{1 - C\exp[\text{-NTU}(1 - C)]} \tag{15.39}$$

where $C = C_{\min}/C_{\max}$

$C_{\max}$ = greater value of C_a and C_w, Btu/h·°F (W/°C)

The number of transfer units (NTU), dimensionless, actually indicates the size of the coil and can be calculated as

$$\text{NTU} = \frac{UA}{C_{\min}} = \frac{U_o A_o}{C_a} = \frac{U_i A_i}{C_a} \tag{15.40}$$

After calculation of ε of the sensible cooling coil from Eq. (15.39), the sensible cooling coil load can be found as

$$Q_{cs} = 60\dot{V}_a \rho_a c_{pa}(T_{ae} - T_{we})\varepsilon \tag{15.41}$$

From the energy balances between the moist air and chilled water, the temperature of conditioned air and chilled water leaving the coil T_{al} and T_{wl}, in °F (°C), can be calculated as

$$\begin{aligned} T_{al} &= T_{ae} - \varepsilon(T_{ae} - T_{we}) \\ T_{wl} &= T_{we} + C(T_{ae} - T_{al}) \end{aligned} \tag{15.42}$$

Heating Coils

In heating coils, only sensible heat change occurs during the heating process. The humidity ratio w remains constant unless humidifiers are installed before or after the heating coil. For steam heating coils, the steam temperature $T_{\text{steam}} \gg T_{ae}$. At the same time, T_{steam} remains constant during the heating process. Therefore, $C_{\max}$ equals infinity, and $C_{\min}/C_{\max} = 0$.

The system performance of both water heating and steam heating coils can also be determined from Eqs. (15.36) through (15.42), in which the hot fluids are hot water or steam, and moist air is cold fluid.

Fluid Velocity and Pressure Drop

Air velocity calculated based on the face area of a finned-tube coil v_a is a primary factor that determines the effectiveness of heat transfer, carryover of droplets of condensate in wet coils, air-side pressure drop, and energy consumption of the system. For dry coils, there is no danger of carryover. Their maximum face velocity is usually limited to a less than 800 fpm (4 m/s). The maximum air velocity calculated based on the minimum free flow area may be as high as 1400 fpm (7 m/s). For coils used in terminal units such as floor-mounted fan coil units, the face velocity is usually around 200 fpm (1 m/s) so that the pressure drop across the coil is low.

In addition to the face velocity v_a, the air-side pressure drop Δp_a depends also on fin-and-tube configuration. For dry coils with a fin spacing of 12 fins/in. (2.1-mm fins), and at a face velocity $v_a = 600$ fpm (3 m/s), Δp_a may vary from 0.1 to 0.2 in. WC (25 to 50 Pa) per row depth.

Selection of water velocity and $\dot{V}_{\text{gal}}$, in gpm (L/s), for a finned-tube coil of a given tube inner diameter and number of water circuits is closely related to the temperature rise or drop. For a finned-tube coil with a face area of 1 ft^2 (m^2), a full serpentine four-row coil may need 4.8 gpm (0.303 L/s) of chilled water at a water temperature rise around 10°F (5.6°C), whereas a $\frac{1}{2}$ serpentine coil needs only 2.4 gpm (0.151 L/s) and has a temperature rise of 20°F (11.1°C). Heat transfer, pressure drop, erosion, noise, energy, maintenance space, and initial cost should be considered during the selection of the temperature rise or drop and its corresponding water volume flow $\dot{V}_{\text{gal}}$ and water velocity. For finned-tube coils, a temperature rise of 10 to 20°F (5.6 to 11.1°C) is generally used. Water-side pressure drop is usually limited to 10 psi, about 22.5 ft WC (69 kPa). A water velocity between 2 and 6 ft/s (0.6 and 1.8 m/s) and a pressure drop of 10 ft WC (30 kPa) should be maintained for reasonable pump power consumption.

Dry Coil at Part-Load Operation

Because the sensible cooling and heating processes are horizontal lines on the psychrometric chart, if the entering chilled water temperature or hot water temperature remains the same at part-load as in full-load operation, then the reduction of water flow rates at part-load operation tends to shorten the horizontal lines from *el* to *elp* (as shown in Fig. 15.29) to maintain the required space temperature at part-load operation.

Coil Construction Parameters

Coil construction parameters of sensible cooling and heating coils (dry coils) are similar to those in DX coils (wet coils) covered in Sec. 10.2.

Example 15.1. A counterflow sensible cooling coil using corrugated plate fins has the following construction and operating data:

Face area (finned length × finned height)	10 ft^2 (0.93 m^2)
Fin spacing	12 fins/in. (2.1-mm fin spacing)
Number of rows	6 rows
Outside diameter of copper tubes	0.528 in. (13.4 mm)
Inside diameter of copper tubes	0.496 in. (12.6 mm)
Entering dry-bulb temperature	90°F (32.2°C)
Entering chilled water temperature	60°F (15.6°C)
Face air velocity	600 fpm (3 m/s)
Chilled water flow rate	28 gpm (1.77 L/s)
Specific heat: moist air	0.243 Btu/lb·°F (1.02 kJ/kg·°C)
water	1 Btu/lb·°F (4.187kJ/kg·°C)
Air density	0.075 lb/ft^3 (1.2 kg/m^3)
Water density	62.4 lb/ft^3 (998 kg/m^3)
At a temperature of 65°F (18.3°C):	
Air dynamic viscosity μ_a	0.0437 lb/ft·h (0.018 mPa·s)
Water dynamic viscosity μ_w	2.46 lb/ft·h (1.02 mPa·s)
Water thermal conductivity k	0.348 Btu/h·ft^2·°F (0.602 W/m·°C)

Calculate

1. Outer surface heat-transfer coefficient of coil
2. Inner surface heat-transfer coefficient of coil
3. Fin surface efficiency
4. Effectiveness of sensible cooling coil
5. Sensible cooling coil load
6. Off-coil air temperature and temperature of chilled water leaving the cooling coil

Solution

1. For corrugated fins, use a multiplying factor $F_{c,a} = 1.15$. Then, from Table 10.1, parameter $F_s = 14.21$. For a six-row coil with a surface area of 10 ft^2, the outer surface area is

$$A_o = 14.21 \times 1.15 \times 6 \times 10 = 981 \text{ ft}^2$$

From Table 10.1, $A_o/A_i = 11.4 \times 1.15$. The inner surface area of this sensible cooling coil is calculated as

$$A_i = \frac{981}{11.4 \times 1.15} = 74.9 \text{ ft}^2$$

For a fin spacing of 12 fins/in., $A_{\min}/A_a = 0.58$. Therefore, from Eq. (15.30), the air mass velocity at minimum flow area is

$$G_c = 60\rho v_{\min} = 60 \times 0.075 \times \frac{600}{0.58} = 4655 \text{ lb/ft}^2\cdot\text{h}$$

and the Reynolds number, based on the outside diameter of the tubes, is

$$\text{Re}_D = \frac{G_c D_c}{\mu} = 4655 \times \frac{0.528}{0.0437 \times 12} = 4687$$

Also parameter JP is calculated as

$$\text{JP} = \text{Re}_D^{-0.4}(A_o/A_p)^{-0.15} = (4687)^{-0.4}(13.27)^{-0.15} = 0.0231$$

From Eq. (15.29), at JP = 0.0231,

$$\frac{h_o}{G_c c_{pa}} \text{Pr}^{2/3} = 0.00125 + 0.27 \times 0.0231 = 0.00748$$

For smooth fins,

$$h_{o,s} = 0.00748 \times 4655 \times 0.243 \frac{1}{0.71} = 11.92 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

For corrugated fins, a multiplying factor $F_{\text{cor}} = 1.20$ is used to account for the increase in turbulence. Therefore, the outer surface heat-transfer coefficient is calculated as

$$h_o = 11.92 \times 1.2 = 14.3 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F} \ (812 \text{ W/m}^2\cdot{}^\circ\text{C})$$

2. Assume a water velocity of 4.7 ft/s inside the copper tubes. Then the Reynolds number of water, based on the inner diameter of copper tube, is

$$\text{Re}_D = \frac{\rho v_w D_i}{\mu_w} = 62.4 \times 4.7 \times 3600 \times \frac{0.496}{2.46 \times 12} = 17{,}740$$

And the Prandtl number for water is

$$\text{Pr} = \mu c_{pw}/k = 2.46 \times \frac{1}{0.348} = 7.07$$

From Eq. (15.31), the inner surface heat-transfer coefficient can be calculated as

$$h_i = 0.023\left(\frac{k}{D_i}\right)\text{Re}_D^{0.8}\,\text{Pr}^{0.4}$$

$$= 0.023\left(\frac{0.348 \times 12}{0.496}\right)(17{,}740)^{0.8}(7.07)^{0.4} = 1061 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F} \ (6019 \text{ W/m}^2\cdot{}^\circ\text{C})$$

3. For a coil with a longitudinal spacing equal to 1.083 in., $S_T = 1.25$ in., $r_o = 0.528/2$ in., $k_f = 100$ Btu/h·ft·°F, copper tubes in staggered arrangement, $W = 0.625$ in., and $S_L = 0.542$ in. From Fig. 15.29, at $W/r_o = 0.625/0.264 = 2.37$ and $S_L/W = 0.87$, $\varphi_{f,\max}$ is about 0.5. Then, from Eq. (15.33)

$$R_{f,\max} = \frac{\varphi_{f,\max} W^2}{F_t k_f} = \frac{0.5(0.625)^2}{0.006 \times 100 \times 12}$$

$$= 0.0271$$

And from Eq. (15.34),

$$\eta_f = \frac{1}{1 + h_o R_{f,\max}} = \frac{1}{1 + 14.3 \times 0.0271} = 0.721$$

Assume $\varphi_f/\varphi_{f,\max} = 0.9$, then $R_f = 0.9 \times 0.0271 = 0.0244$. Therefore, fin efficiency is calculated as

$$\eta_f = \frac{1}{1 + 14.3 \times 0.0244} = 0.741$$

From Eq. (15.28), fin surface efficiency is calculated as

$$\eta_s = 1 - \frac{A_f}{A_o}(1 - \eta_f) = 1 - 0.913(1 - 0.741) = 0.763$$

4. From given data, heat capacity rates are calculated as

$$C_a = 60\dot{V}_a \rho_a c_{pa} = 60 \times 600 \times 10 \times 0.075 \times 0.243 = 6561 \text{ Btu/h} \cdot °\text{F}$$

$$C_w = 500\dot{V}_{\text{gal}} c_{pw} = 500 \times 30 \times 1 = 15{,}000 \text{ Btu/h} \cdot °\text{F}$$

Minimum to maximum ratio is

$$C = \frac{C_a}{C_w} = \frac{6561}{15{,}000} = 0.4374$$

Then, from Eq. (15.27),

$$U_o A_o = \frac{1}{1/(\eta_s h_o A_o) + 1/(h_i A_i)}$$

$$= \frac{1}{1/(0.763 \times 14.3 \times 981) + 1/(1061 \times 74.9)} = 9432 \text{ Btu/h} \cdot °\text{F}$$

From Eq. (15.40), the number of transfer units is

$$\text{NTU} = \frac{U_o A_o}{C_a} = \frac{9432}{6561} = 1.438$$

Also, $$\exp[\text{-NTU}(1-C) = \exp[-1.438(1 - 0.4686)] = 0.4657$$

From Eq. (15.39), the effectiveness of the sensible cooling coil is

$$\varepsilon = \frac{1 - \exp[\text{-NTU}\,(1 - C)]}{1 - C\exp[\text{-NTU}(1 - C)]}$$

$$= \frac{1 - 0.4657}{1 - 0.4374 \times 0.4657} = 0.6834$$

5. Then, from Eq. (15.41), the sensible cooling coil load is

$$Q_{cs} = 60\dot{V}_a \rho_a c_{pa} (T_{ae} - T_{we})\varepsilon$$

$$= 60 \times 600 \times 10 \times 0.075 \times 0.243(90 - 60)0.6834 = 134{,}522 \text{ Btu/h (38,415 W)}$$

6. From Eq. (15.42), the temperature of conditioned air leaving the sensible cooling coil is

$$T_{al} = T_{ae} - \varepsilon(T_{ae} - T_{we}) = 90 - 0.6834(90 - 60) = 69.5°\text{F}$$

The temperature of chilled water leaving the coil is

$$T_{wl} = T_{we} + C(T_{ae} - T_{al})$$
$$= 60 + 0.4686\,(90 - 69.5) = 69.6°\text{F}\ (20.9°\text{C})$$

15.10 WATER COOLING COILS—DRY-WET COILS

As in a DX coil, the airflow and water flow arrangement in a water cooling coil is usually a combination of counterflow and crossflow. When chilled water is used as the coolant flowing inside the copper tube of a cooling coil, the outer surface temperature of the coil at the air intake side T_{se} is probably greater than the dew point of the entering air T''_{ae}, that is, $T_{se} > T''_{ae}$. The outer surface temperature of the coil at the air leaving side $T_{sl} < T''_{ae}$. A water cooling coil is then a dry-wet coil. It often has a partially dry surface at the air intake side and partially wet surface at the air leaving side.

When the temperature of chilled water entering the coil T_{we} is low, or the dew point temperature of the entering air T''_{ae} is high, and if $T_{se} < T''_{ae}$, the water cooling coil is then a wet coil. The differences between a wet water cooling coil and a wet DX coil are mainly the lack of a superheated region in a water cooling coil and the variation of chilled water temperature inside a water cooling coil, which is often greater than the variation of evaporating temperature T_{ev} at the two-phase region in a DX coil.

Dry-Wet Boundary

In a dry-wet coil, there is always a dry-wet boundary that divides the coil between the dry surface and the wet surface, as shown in Fig. 15.31. The dry-wet boundary must be determined so that the performance of the dry surface and the wet surface can be calculated separately.

Because the outer surface temperature of the copper tubes at the dry-wet boundary T_{sb} is exactly equal to the dew point of the air entering the coil T''_{ae}, that is, $T_{sb} = T''_{ae}$, the dry-wet boundary can be determined. The dry surface extends from the dry-wet boundary to the air intake side, and the wet surface extends to the air leaving side.

At the dry-wet boundary, if the thermal resistance of the metal tube and the heat energy released by condensate are ignored, then

$$h_{\text{wet}}\eta_s A_o(T_{ab} - T_{sb}) = h_i\left(\frac{A_o}{B}\right)(T_{sb} - T_{wb}) \tag{15.43}$$

where T_{ab} = temperature of airstreams at dry-wet boundary,°F (°C)
T_{wb} = temperature of chilled water inside tubes at dry-wet boundary,°F (°C)
h_i = inner surface heat-transfer coefficient, Btu/h·ft^2·°F (W/m^2·°C)

In Eq. (15.43), h_{wet} is the heat-transfer coefficient of the wetted surface, in Btu/h·ft^2·°F (W/m^2·°C), as discussed in Sec. 10.8. From Eq. (10.21), $h_{\text{wet}} = m'' h_o / c_{pa}$. Here h_o is the heat-transfer coefficient of the outer surface, in Btu/h·ft^2·°F (W/m^2·°C); m'' is the slope of air enthalpy saturation curve, $m'' = dh_s/dT_s$, as shown in the upper right-hand corner of Fig. 15.31; h_s denotes the enthalpy of saturated air film, in Btu/lb (J/kg); and T_s is the saturated temperature, in °F (°C). At $T_{o,s}$ = 60°F (15.6°C), the ratio $h_{\text{wet}}/h_o = m''/c_{pa}$ is about 2.8.

In Eq. (15.43), there are two unknowns, T_{sb} and T_{wb}. An additional equation is needed to determine them. At the dry-wet boundary of the dry-wet coil, there exists an energy balance between the

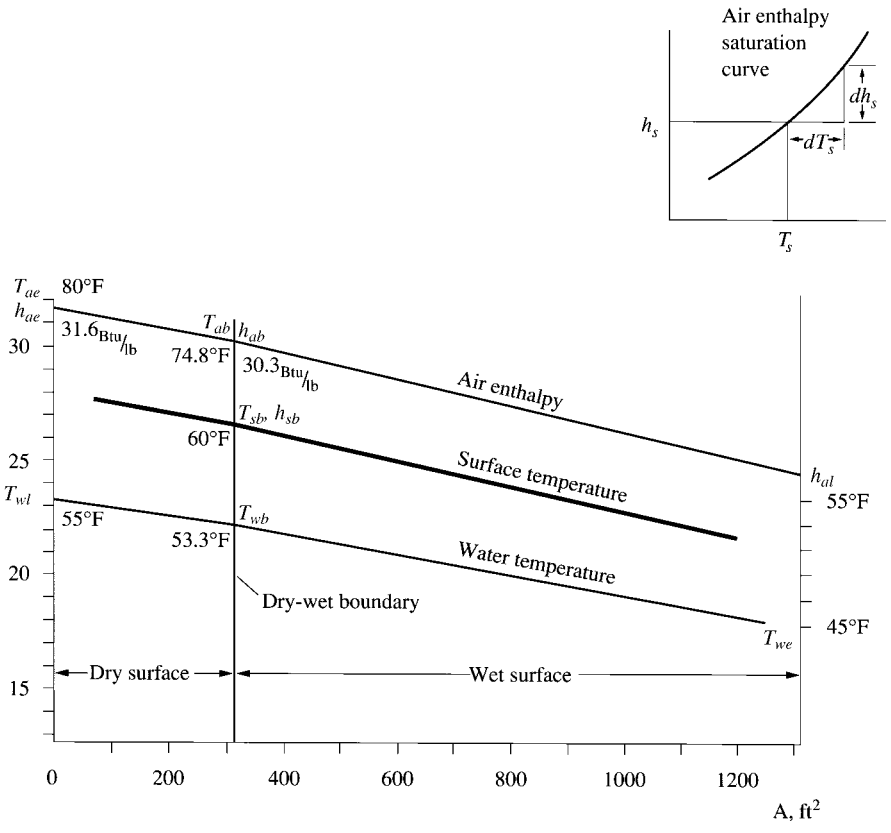


FIGURE 15.31 Dry-wet boundary of a dry-wet coil.

airstream and water stream

$$\dot{V}_a \rho_a c_{pa}(T_{ae} - T_{ab}) = 8.33 \dot{V}_{\text{gal}} c_{pw}(T_{wl} - T_{wb}) \tag{15.44}$$

where $\dot{V}_a$ = volume flow rate of conditioned air, cfm
$\dot{V}_{\text{gal}}$ = volume flow rate of chilled water, gpm

Because the temperature of chilled water leaving the coil T_{wl} is a known value, from Eqs. (15.43) and (15.44), T_{ab} and T_{wb} can be determined.

Dry Part

The sensible cooling capacity Q_{cs}, in Btu/h (W), of the dry part of the dry-wet coil can be calculated from the left-hand side of Eq. (15.44) as

$$Q_{cs} = 60 \dot{V}_a \rho_a c_{pa}(T_{ae} - T_{ab}) \tag{15.45}$$

The average temperature of the airstream in the dry part $T_{ad,\,m}$, in °F (°C), is approximately equal to $\frac{1}{2}(T_{ae} + T_{ab})$, and the average temperature of the chilled water corresponding to the dry part of the

coil $T_{wd,m}$, in °F (°C), is $\frac{1}{2}(T_{wb} + T_{wl})$. Unit sensible cooling capacity of each ft² (m²) outer surface area of the dry part of the water cooling coil, including the tube and fins, can then be calculated as

$$\frac{Q_{cs}}{A_{o,d}} = U_o(T_{ad,m} - T_{wd,m})$$

Because U_o can be calculated from Eq. (15.27), the total outer surface area of the dry part of the coil $A_{o,d}$, in ft² (m²) is

$$A_{o,d} = \frac{Q_{cs}}{U_o\,(T_{ad,m} - T_{wd,m})} \tag{15.46}$$

Wet Part

As for the DX coil, the cooling and dehumidifying capacity of the wet part of a dry-wet coil $Q_{c,w}$, in Btu/h (W), can be calculated as

$$Q_{c,w} = \varepsilon_{\text{wet}} 60 \dot{V}_a \rho_a (h_{ab} - h_{s,we}) \tag{15.47}$$

where $h_{s,we}$ = enthalpy of saturated air film at chilled water entering temperature,°F (°C). The effectiveness of the wet part of the dry-wet coil (wet, the number of transfer units (NTU), and the heat capacity rates C_a and C_w can be similarly calculated by Eqs. (15.39), (15.40), (15.36), and (15.37), respectively.

Dry-Wet Coil at Part-Load Operation

When the dry-wet coil of a constant-volume system is at part-load operation, the reduction of the space cooling load results in a drop in space temperature. This causes the control system to modulate the two-way valve at the inlet of the water cooling coil to reduce the mass flow rate of chilled water flowing through the coil, as shown in Fig. 15.32. Because of the reduction of the chilled water flow rate, the following conditions occur:

- The velocity of chilled water inside the coil tubes is reduced, and therefore the rate of heat-transfer between the air and water decreases.
- Although the reduction of the space cooling load tends to lower the temperature of conditioned air leaving the coil T_{al}, nonetheless T_{al} increases because of the reduction of the chilled water flow rate and has a greater influence than the drop of the space cooling load.
- From the psychrometric analysis shown in Fig. 15.32, the space conditioning line *sp-rp* at part-load operation is shorter because of the reduction of space cooling load.
- The chilled water flow rate is reduced so that the off-coil temperature T_{ccp} and supply temperature T_{sp} match the reduction of space cooling load and maintain a space temperature T_{rp} at part-load operation.
- The higher dry-bulb and dew point temperatures of conditioned air leaving the coil cause an increase in space relative humidity φ_r. Sometimes, φ_r may be raised to 70 percent during part-load operation. This is the main disadvantage in controlling space temperature by means of chilled water flow rate modulation at part-load operation.

Selection of a Dry-Wet Coil

The construction of a water cooling coil, a dry-wet coil, is similar to that of a DX coil, covered in Sec. 10.2. As for a DX coil, the procedure for selecting a water cooling coil according to the manufacturer's data is as follows:

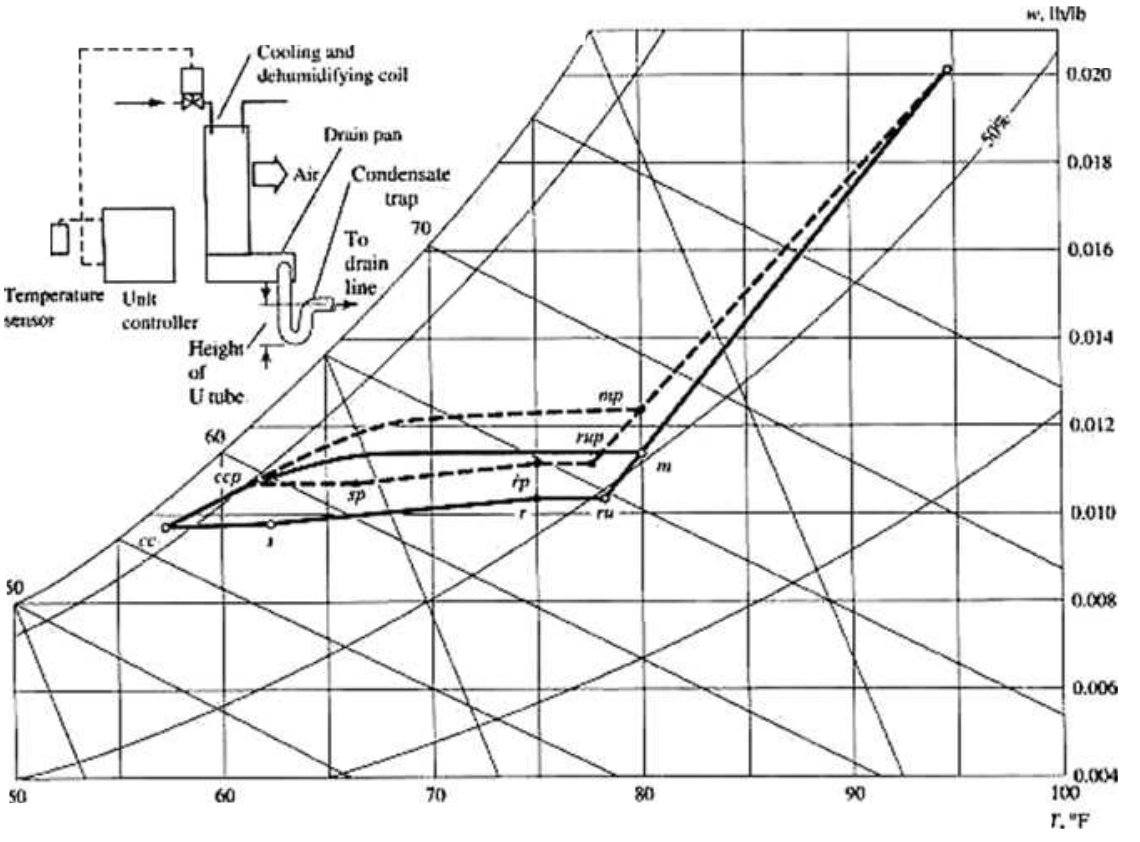


FIGURE 15.32 Psychrometric analysis of part-load operation of a dry-wet coil.

1. According to the required supply volume flow rate, choose an optimum face velocity v_a; it is usually not greater than 550 fpm (2.75 m/s) to prevent condensate carryover, wet surfaces and possible mold growth in air system, and a 500 fpm (2.5 m/s) or less cooling coil face velocity is preferable. Calculate the face area A_a of the coil.

2. From the coil load, calculate the unit cooling and dehumidifying capacity per ft^2 face area Q_c/A_a.

3. Determine the chilled water temperature entering the coil T_{we}. This is closely related to the required condition of air leaving the coil T_{al}.

4. Select the optimum chilled water temperature rise ΔT_w. A lower ΔT_w means a greater water flow rate, a higher water velocity in the tubes, and a greater heat-transfer coefficient. A lower ΔT_w also results in a greater pump power and a higher initial investment in the water system. Water temperature rise ΔT_w also influences air system and refrigeration system performance. It must be matched with the expected temperature of conditioned air off the coil in the air system as well as the chilled water temperature difference in the chiller.

The current practice is to use a ΔT_w from 10 to 20°F (5.6 to 11.1°C) in order to reduce water flow and save pump power. When ΔT_w is determined, $\dot{V}_{\text{gal}}$, in gpm (L/s), can be calculated accordingly.

5. Check the water velocity inside the tubes v_w, in ft/s (m/s), by selecting an appropriate finned width for the water cooling coil. Water velocity of 2 to 6 ft/s (0.6 to 1.8 m/s) typically between 3 and 5 ft/s (0.9 and 1.5 m/s) is appropriate. As in sensible water cooling coils, the water-side pressure drop of the coil should not exceed 10 psi, or 22.5 ft WC (69 kPa). Typically, a pressure drop of 10 ft WC (30 kPa) across the coil is considered suitable to keep the pump power and energy cost reasonable. Many manufacturers provide the formula to calculate v_w at a given finned width and the corresponding tube feeds. Some manufacturers also provide water turbulators, which can be added to the tubes to increase the inner surface heat-transfer coefficient if the water velocity is less than 4 ft/s (1.2 m/s). In tubes with water turbulators, the water-side pressure drop is considerably higher than in smooth tubes.

6. Choose the optimum number of rows and fins per inch. An even number of rows are preferable because the supply and return header can be placed on the same side.

7. Select the coil from the manufacturer's catalog at given air and water entering and leaving conditions v_a and v_w and the required Q_c/A_a.

8. Determine the air-side and water-side pressure drop.

Selection of the DX coils and water cooling and heating coils is usually performed by following the procedure given in the manufacturer's catalog. Calculations by means of dry-wet boundary and wet-coil effectiveness are sometimes necessary in detailed analysis and simulation in research and development.

Example 15.2. A water cooling coil in a constant-air-volume system has a supply volume flow rate of 11,000 cfm (5190 L/s). Air enters the coil at a dry-bulb temperature of 80°F (26.7°C) and a wet-bulb temperature of 67°F (27.4°C) and leaves the coil at a dry-bulb temperature of 57.5°F (14.2°C) and a wet-bulb temperature of 56.5°F (13.6°C). Chilled water enters the coil at 45°F (7.2°C) and is expected to leave the coil at 55°F (12.7°C). At an average temperature of 50°F (10°C), the fluid properties are as follows:

Thermal conductivity of water k_w	0.339 Btu/h·ft·°F (0.587 W/m·°C)
Dynamic viscosity of water μ_w	3.09 lb/ft·h (1.28 mPa·s)
Specific heat of water c_{pw}	1.0 Btu/lb·°F (4.187 kJ/kg·°C)

Assume that the air-side heat transfer coefficient $h_o = 13.4$ Btu/h·ft²·F (76.1 W/m²·°C)

1. Select a water cooling coil from the manufacturer's catalog.
2. Determine the dry-wet boundary of this dry-wet coil.
3. Calculate the sensible cooling capacity and the outer surface area of the dry part of this dry-wet coil.
4. Determine the cooling and dehumidifying capacity of the wet part as well as the dry-wet coil.
5. At part-load operation, if the water velocity in the copper tubes is reduced to 63 percent of the full-load value, and air enters the coil at a wet-bulb temperature of 68°F (20°C), determine the operating conditions of this dry-wet coil.

Solution

1. For a corrugated coil, select a face velocity $v_a = 550$ fpm. From the given values, the required face area of the coil is

$$A_a = \frac{\dot{V}_a}{v_a} = \frac{11{,}000}{550} = 20 \text{ ft}^2$$

From the psychrometric chart, the enthalpy of entering air at a dry-bulb temperature of 80°F and a wet-bulb temperature of 67°F is 31.6 Btu/lb. The enthalpy of leaving air at a dry-bulb temperature of 57.5°F and a wet-bulb temperature of 56.5°F is 24.2 Btu/lb. From the given conditions, the total

cooling and dehumidifying capacity of this dry-wet coil is

$$Q_c = 60\dot{V}_a\rho_a(h_{ae} - h_{al}) = 60 \times 11{,}000 \times 0.075(31.6 - 24.2)$$
$$= 366{,}300 \text{ Btu/h}$$

Cooling and dehumidifying capacity per ft^2 of face area is calculated as

$$\frac{Q_c}{A_a} = \frac{366{,}300}{20} = 18{,}315 \text{ Btu/h or } 18.3 \text{ MBtu/h}$$

Similarly, the required sensible cooling capacity per ft^2 of coil face area is

$$\frac{Q_{cs}}{A_a} = 60v_a\rho_a c_{pa}(T_{ae} - T_{al})$$
$$= 60 \times 550 \times 0.075 \times 0.243(80 - 57.5) = 13{,}532 \text{ Btu/h} \cdot \text{ft}^2$$

From the given values, the water temperature rise is 55 − 45 = 10°F, and the water volume flow rate is therefore

$$\dot{V}_{\text{gal}} = \frac{Q_c}{500(T_{wl} - T_{we})} = \frac{366{,}300}{500 \times 10} = 73.3 \text{ gpm } (4.63 \text{ L/s})$$

2. For a face area of 20 ft^2, choose a coil with a finned width of 30 in. and finned length of 96 in. Assume that the water velocity inside the tubes of this water cooling coil is 4 ft/s.

At an air entering condition of 80°F dry-bulb temperature and 67°F wet-bulb temperature v_a = 550 fpm, v_w = 4 ft/s, T_{we} = 45°F, ΔT_w = 10°F, and required Q_c/A_a = 18.3 MBtu/h, a four-row, 12 fins/in. coil selected from manufacturer's catalog has a cooling and dehumidifying capacity of 19.1 MBtu/h per ft^2 of face area and a sensible cooling capacity per ft^2 of

$$\frac{Q_{cs}}{A_a} = 60 \times 550 \times 0.075 \times 0.243\left(80 - \frac{56.4 + 57.4}{2}\right)$$
$$= 13{,}893 \text{ Btu/h} \cdot \text{ft}^2 \text{ } (43{,}832 \text{ W/m}^2)$$

		Face velocity, fpm					
		500			600		
Water temperature rise	10°F	MBtu/h	LDB	LWB	MBtu/h	LDB	LWB
Water velocity	4 ft/s	18.1	56.4	55.4	20.0	57.4	56.4

Here LDB represents the leaving dry-bulb temperature and LWB the leaving wet-bulb temperature, both in °F (°C). Both the cooling capacity per ft^2 of face area and the sensible cooling capacity per ft^2 of face area of this water cooling coil are greater than the required values.

From the psychrometric chart, the dew point of the entering air at 80°F (26.7°C) dry-bulb temperature and 67°F (19.4°C) wet-bulb temperature is 60°F (15.6°C), that is, T_{sb} = 60°F (15.6°C). The slope of the air saturation curve at 60°F (15.6°C) is the difference in saturation enthalpy between 59.5 and 60.5°F (15.3 and 15.8°C), or

$$m'' = 28.81 - 26.12 = 0.69 \text{ Btu/lb} \cdot \text{°F } (2.89 \text{ kJ/kg} \cdot \text{°C})$$

Therefore,

$$\frac{m''}{c_{pa}} = \frac{0.69}{0.243} = 2.84$$

Using an air-side heat-transfer coefficient $h_o = 13.4$ Btu/h·ft^2·°F as in Example 15.1, from Eq. (10.21), the heat-transfer coefficient for the wet surface is

$$h_{\text{wet}} = \frac{m''}{c_{pa}} h_o = 2.84 \times 13.4 = 38.06 \text{ Btu/h·ft}^2\text{·°F (216 W/m}^2\text{·°C)}$$

3. On the water side,

$$\text{Re}_D = \frac{\rho v_w D_i}{\mu_w} = \frac{62.4 \times 4 \times 3600 \times 0.496}{3.09 \times 12} = 12{,}020$$

$$\text{Pr} = \frac{\mu_w c_{pw}}{k_w} = \frac{3.09 \times 1}{0.339} = 9.12$$

Then

$$h_i = 0.023 \text{ Re}_D^{0.8} \text{ Pr}^{0.4} = 0.023 \times (12{,}020)^{0.8}(9.12)^{0.4}$$
$$= 838.6 \text{ Btu/h·ft}^2\text{·°F}$$

As in Example 15.1, $B = 11.4 \times 1.15 = 13.1$, from Eq. (15.43), for 1 ft^2 of outer surface area at the dry-wet boundary,

$$h_{\text{wet}} \eta_s (T_{ab} - T_{sb}) = \frac{h_i}{B} (T_{sb} - T_{wb})$$

$$38.06 \times 0.76 (T_{ab} - 60) = \frac{838.6}{13.1} (60 - T_{wb})$$

From Eq. (15.44),

$$\dot{V}_a \rho_a c_{pa} (T_{ae} - T_{ab}) = 8.33 \dot{V}_{\text{gal}} c_{pw} (T_{wl} - T_{wb})$$

$$11{,}000 \times 0.075 \times 0.243(80 - T_{ab}) = 8.33 \times 74 \times 1(55 - T_{wb})$$

According to the above simultaneous equations,

$$T_{ab} = 74.8\text{°F} \qquad \text{and} \qquad T_{wb} = 53.3\text{°F (11.83°C)}$$

The dry-wet boundary can therefore be determined as shown in Fig. 15.31.

4. From Eq. (15.45), the sensible cooling capacity of the dry part of the dry-wet coil is

$$Q_{cs} = 60 \dot{V}_a \rho_a c_{pa} (T_{ae} - T_{ab})$$
$$= 11{,}000 \times 60 \times 0.075 \times 0.243 \times (80 - 74.8) = 62{,}548 \text{ Btu/h (18,327 W)}$$

The average temperature of the airstream in the dry part is

$$T_{ad,m} = \frac{T_{ae} + T_{ab}}{2} = \frac{80 + 74.8}{2} = 77.4\text{°F}$$

The average temperature of the water stream corresponding to the dry part of the dry-wet coil is

$$T_{wd,m} = \frac{T_{wl} + T_{wb}}{2} = \frac{55 + 53.3}{2} = 54.15\text{°F}$$

From Eq. (15.27),

$$U_o = \frac{1}{1/(\eta_s h_o) + B/h_i} = \frac{1}{1/(0.76 \times 13.42) + 13.11/838.6}$$

$$= 8.796 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Then the outer surface area of the dry part is

$$A_{o,d} = \frac{Q_{cs}}{U_o(T_{ad,m} - T_{wd,m})} = \frac{62{,}548}{8.796(77.4 - 54.15)}$$

$$= 306 \text{ ft}^2 \text{ (28.4 m}^2\text{)}$$

5. From Eqs. (15.36) and (15.37), the heat capacity rates and their ratio are

$$C_a = \dot{m}_a c_{pa} = 60 \times 11{,}000 \times 0.075 \times 0.243 = 12{,}028 \text{ Btu/h} \cdot {}^\circ\text{F}$$

$$C_w = \dot{m}_w c_{pw} = 74 \times 500 \times 1 = 37{,}000 \text{ Btu/h} \cdot {}^\circ\text{F}$$

$$C = \frac{C_{\min}}{C_{\max}} = \frac{12{,}028}{37{,}000} = 0.3251$$

From Eq. (15.26) and Table 10.1, for such a four-row, 12 fins/in. corrugated plate fin coil, the total outer surface area is

$$A_o = F_s A_a N_r = 14.21 \times 1.15 \times 20 \times 4 = 1307 \text{ ft}^2$$

The outer surface area of the wet part of the coil is

$$A_{\text{wet}} = A_o - A_{o,d} = 1307 - 306 = 1001 \text{ ft}^2 \text{ (93 m}^2\text{)}$$

Then, from Eq. (15.27),

$$U_o A_o = \frac{1}{1/(\eta_s h_o A_o) + 1/(h_i A_i)}$$

$$= \frac{1}{1/(0.76 \times 13.42 \times 1001) + 13.11/(838.6 \times 1001)} = 8805 \text{ Btu/h} \cdot {}^\circ\text{F}$$

And from Eq. (15.40)

$$\text{NTU} = \frac{U_o A_o}{C_a} = \frac{8805}{12{,}028} = 0.7321$$

Also $\exp[\text{-NTU}(1 - C)] = \exp[-0.7321(1 - 0.3251)] = 0.6101$

Therefore, from Eq. (15.39), the effectiveness of the wet part of the dry-wet coil is

$$\varepsilon_{\text{wet}} = \frac{1 - \exp[\text{-NTU}(1-C)]}{1 - C \exp[\text{-NTU}(1-C]}$$

$$= \frac{1 - 0.6101}{1 - 0.3251 \times 0.6101} = 0.4864$$

6. From the psychrometric chart, at a T_{ab} of 74.8°F and a dew point of 60°F, $h_{ab} = 30.3$ Btu/lb· Also the enthalpy of saturated air film $h_{s,we} = 17.65$ Btu/lb when the temperature T_{we} is 45°F. Then the cooling and dehumidifying capacity of the wet part of the dry-wet coil is

$$Q_{c,w} = \varepsilon_{\text{wet}} \dot{V}_a \rho_a (h_{ab} - h_{s,we})$$
$$= 0.4864 \times 11{,}000 \times 60 \times 0.075(30.3 - 17.65) = 304{,}571 \text{ Btu/h}$$

The cooling capacity of the dry-wet coil is

$$Q_c = Q_{cs} + Q_{c,w} = 62{,}548 + 304{,}571 = 367{,}119 \text{ Btu/h (107,566 W)}$$

or 367,119/20 = 18.4 MBtu/h · ft^2 which is close to the value 19.1 MBtu/h · ft^2 given in the manufacturer's catalog.

7. If the water velocity inside the copper tubes is reduced to 63 percent of the full-load value $v_w = 0.63 \times 4 = 2.5$ ft/s and the water temperature rise is increased to 16°F, from the performance data shown in manufacturer's catalog, at an entering dry-bulb temperature of 79°F, entering wet-bulb temperature of 68°F, and an enthalpy of 32.4 Btu/lb, the cooling capacity is reduced from 19.1 to 15.3 MBtu/h · ft^2. Total cooling and dehumidifying capacity of the water cooling coil at part load is

$$Q_c = 15.3 \times 1000 \times 60 \times 20 = 306{,}000 \text{ Btu/h (89,660 W)}$$

The reduction of enthalpy Δh during the cooling and dehumidifying process at part-load operation is

$$\Delta h = \frac{306{,}000}{11{,}000 \times 60 \times 0.075} = 6.2 \text{ Btu/lb}$$

The enthalpy of the air leaving the coil is

$$h_{al} = 32.4 - 6.2 = 26.2 \text{ Btu/lb}$$

At an entering dry-bulb temperature of 79°F and wet-bulb temperature of 68°F, from the cooling and dehumidifying curve on the psychrometric chart, the condition of air off the coil at part-load operation is 61.3°F dry-bulb and 59.5°F wet-bulb temperature. The load ratio at part load = 15.3/19.1 = 0.80.

If the sensible heat ratio and system heat gain at part-load operation are the same as in full-load operation, the air conditioning cycle at part load can then be drawn on the psychrometric chart, as shown by the dotted line in Fig. 15.32. The space relative humidity has been increased from 55 percent at full load to 59 percent at 80 percent part-load operation. Air conditioning processes and cycles are discussed in Chap. 20.

15.11 COIL ACCESSORIES, CONDENSATE CONTROL, AND FREEZE-UP PROTECTION

Coil Accessories

Coil accessories include air vents, drain valves, isolation valves, flow metering valves, balancing valves, thermometers, pressure gauge taps, condensate drain taps, and even distribution baffles.

Air Vents. These devices discharge air from the coils. Air vents are usually located at the highest point of the coil and at the pipe risers. Manual vents are inexpensive and more reliable, whereas automatic vents are more convenient to operate.

Drain and Isolating Valves. Drain valves should be provided for each coil for maintenance and repair. Hose-end type of drain valves may be used for remote drains. Isolating valves should be installed at both supply and return piping to cut off the water supply during repair and maintenance.

Even Distributing Baffles. When heating coils are stacked above cooling coils, usually perforated baffles should be installed in the path with less flow resistance to balance the difference in pressure losses between two parallel flow paths. Heating coils usually have fewer rows and less flow resistance than cooling coils.

Coil Cleanliness

Coil cleanliness is important for proper maintenance and operation. Coil cleanliness includes outer surface cleanliness and internal coil care. If an air filter of a minimum of 25 percent dust-spot efficiency is installed upstream from the coil without leakage air bypassing the air filter, then dirt will not accumulate on the outer surface of the coil. If an air filter has efficiency lower than 25 percent dust-spot rating or with many air leaks bypassing the air filter, dirt accumulation may eventually block the air paths in a coil and significantly increase the pressure drop across the coil. In air systems, coils usually should be inspected and cleaned every 3 months in urban areas. Blowing out contaminants with compressed air and flushing with chemical detergent are the two cleaning methods often used.

A chilled water system is usually a closed water system. Most of the makeup water taken from the city water supply in the United States is well treated. If the water strainer is periodically inspected and works properly, the inner surface of the copper tubes of a water cooling coil is usually not contaminated.

Symonds (1997) pointed out that the internal coil care is needed after a burnout of a compressor or a compressor motor in a packaged system. In such a condition, the DX coil is often dismantled into segments. It is preferable to use chemical agent to flush the inner surface of the coil to clean the existing acid, sludge, and solid particles.

Condensate Collection and Drain System

In DX coils and water cooling coils, air is often cooled and dehumidified in summer. Moisture is removed from air and forms condensate on the outer surface of the DX coil and the wet part of the water cooling coil. A condensate collection device must be installed, and a condensate drain line should be designed. This is to prevent the leakage of condensate and the stagnancy of the condensate flow which often cause the indoor air quality problems due to the growth of the bacteria and microorganisms inside the the drain pans. Trent and Trent (1997b) discussed the problems involved in condensate control.

Condensate Flow. The amount of condensate removed from the conditioned air depends mainly on the type of air system (whether it is a comfort system, a processing system, a makeup system—all outdoor air—or a recirculating system) and the location of the air system. If air enters the coil at 80°F (26.7°C) dry-bulb temperature, 67°F (19.4°C) wet-bulb temperature and a humidity ratio $w_e = 0.0116$ lb/lb (0.0116 kg/kg) and leaving the coil at 57°F (13.9°C) dry-bulb temperature, 56°F (13.3°C) wet-bulb temperature, and a humidity ratio $w_l = 0.00905$ lb/lb (0.00905 kg/kg), for a cooling and dehumidifying capacity of 1 refrigeration ton (400 cfm or 189 L/s supply air) in a comfort system using recirculating air, the mass flow rate of condensate $\dot{m}_{\text{con}}$ can be calculated as

$$\dot{m}_{\text{con}} = \dot{V}_a \rho_a (w_e - w_l) = 400 \times 0.075(0.0116 - 0.0095)$$
$$= 0.063 \text{ lb/min or } 0.0075 \text{ gpm } (0.00047 \text{ L/s})$$

Drain Pan. Drain pans have been widely installed under the DX and water cooling coils to collect condensate, as shown in Fig. 15.32. When two or more DX coils are banked vertically, there must be an individual drain pan and drainpipe for each coil. The drainpipe of the top drain pan should be connected to the lower main drain pan. Coils are preferably installed vertically to avoid condensate drips from sloped coils. To prevent the carryover of the water drains down on the coil, drain pans should be spaced at a maximum of 42 in. (1.07 m) vertically.

The size of the drain pan is nearly the same as the size of the base of the coil except that a drain pan should extend in the direction of airflow a depth equal to 1.5 to 3 times the depth of the coil for the installation of the drain port and the inspection of condensate flow. A larger extended drain pan to protect the floor from condensate carryover is not a correct approach. Condensate carryover is prevented only by choosing a coil face velocity lower than 550 fpm (2.75 m/s) for corrugated fins.

Condensate drain pans are preferably made of 16-gauge (1.6-mm-thick) stainless steel and are usually 2 in. (50 mm) deep. A 1-in. (25-mm) stainless-steel pipe joint should be used as the drain port at the end of the pan wall from which the condensate drain line is connected. Drain port is preferably located at the same side as where the chilled water enters the coil. The floor of the condensate pan should be sloped $\frac{1}{4}$ in./ft (21 mm/m length) toward the drain port. Tests showed that about $\frac{1}{8}$ in. (3-mm) depth of water in the condensate pan is required to overcome the surface tension and to allow the condensate to flow freely.

Condensate drain pans should be cleaned monthly to prevent the buildup of bacteria and microorganisms which contaminate the indoor air.

Condensate Trap. Most of the supply fans in air-handling units (AHUs) and packaged units (PUs) are located downstream from the coil (a draw-through fan). The coil section in an AHU or PU with a draw-through fan is always in a negative static pressure $-p_{s,\text{coil}}$ of -1.5 to -4 in. WG (-375 to -1000 Pag). A condensate trap, as shown in Fig. 15.32, is often used to isolate the coil section from the outside atmosphere. The height of the water column in the U tube of the condensate trap, in in. WC, must be 1 or 2 in. (250 to 500 Pa) greater than $-p_{s,\text{coil}}$. When there is no condensate in an idled DX or water cooling coil, the ambient air outside the AHU or PU can be blocked by the water filled in the U tube. Outside ambient air is not extracted into the AHU or PU because of the condensate trap. A condensate trap should discharge freely the liquid condensate that has collected in the drain pan.

Condensate Drain Line. Most condensate drain lines are gravity drain systems. The difference in water levels between the water surface in the drian pan and the water level in the discharge sump provides the available head Δp_{ava} to overcome the pressure loss Δp_{drain} of the drain line, both in ft WC (m head). The diameter of the drain line should be determined according to the calculated gpm (L/s) of the condensate flow rate Δp_{ava} and Δp_{drain}. When the coil and the drain line discharge sump are on the same floor, the drain pan is sometimes elevated for a greater Δp_{ava}. The diameter of the drain line should not be less than a minimum size (such as 1-in. or 25-mm diameter) so that it may not be easily blocked by dirt. The drain line should be sloped away from the condensate trap at a rate of no less than $\frac{1}{8}$ in./ft (10 mm/m).

Condensate drain line is often made of PVC, steel, or sometimes copper. PVC or steel pipes must be schedule 40 or heavier. For PVC pipes, fixed supports should be installed at 2- to 3-ft (0.6- to 0.9-m) intervals, and steel pipes in 8- to 10-ft (2.4- to 3-m) intervals.

Coil Freeze Protection

One of the most common problems encountered with water and steam coils is coil freeze-up. Water inside tubes freezes when its temperature is below the freezing point.

Air Stratification. Improper mixing of outdoor and recirculating air in the mixing box of an air-handling unit or packaged unit often causes coil freeze-up when the outdoor air temperature is below 32°F (0°C). A mixture of 30 percent outdoor air at 0°F (-17.8°C) and 70 percent recirculating

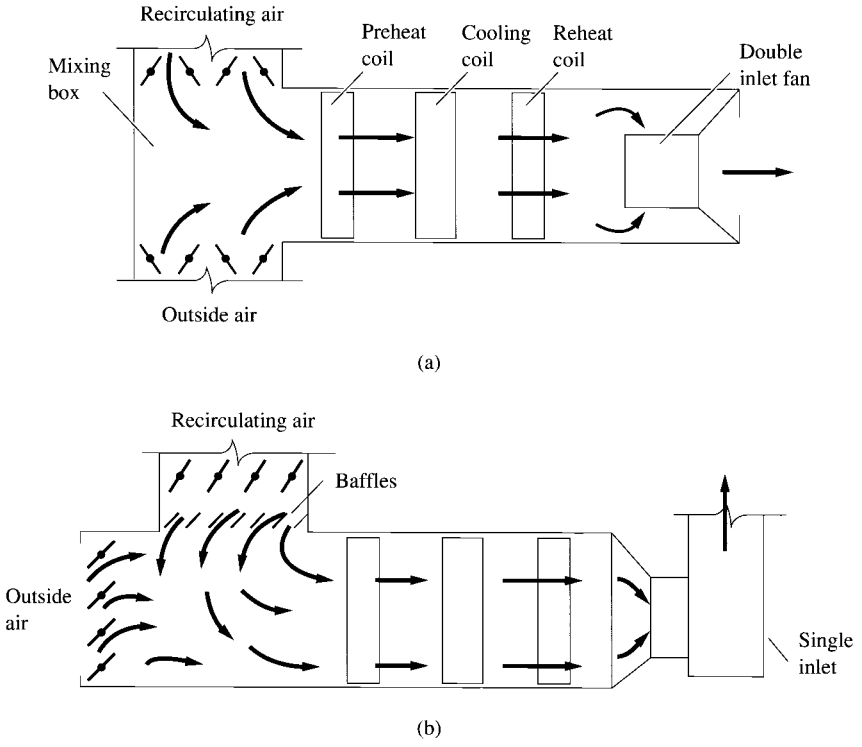


FIGURE 15.33 Air stratification and freeze-up protection: (*a*) freeze-up due to air stratification; (*b*) preventing freeze-up.

air at 70°F (21.1°C) theoretically gives an average temperature of 49°F (9.4°C), which is far above the freezing point of water. However, such a temperature is only maintained by thorough mixing. If outdoor air and recirculating air enter the mixing box of an air-handling unit, as shown in Fig. 15.33*a*, the outdoor air and recirculating air will not mix thoroughly and will form air stratification. Low-temperature outdoor air flows over part of the preheat coil, cooling coil, or even the reheat coil, and may freeze part of the coil.

Baffles can be used to direct the airstream to promote thorough mixing, as shown in Fig. 15.33*b*. The face velocity of the baffles should be between 1000 and 1500 fpm (5 and 7.5 m/s). Temperature sensors, installed at the coil face where the lowest air intake temperature may occur, can be used to shut off the fan and the outdoor air damper when the temperature is below freezing. For coils installed downstream from a supply fan, a single-inlet fan always provides better mixing than a double-inlet fan.

Draining Water and Inner Distributor Tubes. Freeze-up of idled chilled water coils can be prevented by draining the water inside the tube completely. Larocca (1997) recommended to blow air from the supply fan continuously through the coils to ensure that the tubes inside become completely dry. For most air-handling units, the fan total pressure produced by a supply fan is sufficient to overcome the internal resistance of the coil tubes. A 3-in.- (75-mm-) diameter flexible duct is used to connect the supply branch duct and the coil supply header via a shutoff valve. Compared with the addition of glycol solution to water to lower the freezing point, this method saves the cost of glycol and the labor involved in handling, mixing, and disposing of the glycol.

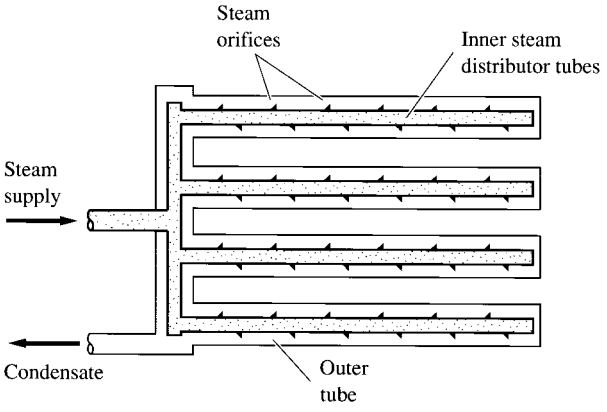


FIGURE 15.34 Steam heating coil with inner distributor tubes for freeze-up prevention.

If a steam heating coil is installed in a location where the outdoor air temperature may drop below 32°F (0°C) in winter, as described by Delaney et al. (1984), a coil with inner distributor tubes for freeze-up protection should be used, as shown in Fig. 15.34. Steam condensate can drain from the outer tubes easily without blocking the steam supply.

Incorporating Freeze Plugs. Larocca (1997) suggested to incorporate freeze plugs in the coil design (pressure relief) to relieve the high pressure caused inside the coil tubes, bends, and header during freezing.

Running the Water Pump. Freeze-up of water cooling coils can be prevented by running the chilled water pump to provide a certain velocity inside the coil tubes when the outside air temperature is below freeezing. If a water velocity of 2.5 ft/s (0.75 m/s) is maintained inside a water cooling coil, the minimum air temperature can drop to 32°F (0°C) without freezing the coils.

15.12 AIR CLEANING, RATING, AND ASSESSMENTS

Air Filtration and Industrial Air Cleaning

Air cleaning is the process of removing airborne particles present in the air. It can be classified into two categories: air filtration and industrial air cleaning. Air filtration involves the removal of airborne particles present in outdoor air as well as recirculated air from a given space. Most airborne particles removed by air filtration are smaller than 1 μm, and the concentration of these particulates in the conditioned airstream is often less than 2 mg/m^3. The purpose of air filtration is to benefit the health and comfort of the occupants in the conditioned space as well as maintain the cleanliness required in manufacturing processes. Air filtration is one of the essential factors that affects the indoor air quality of the conditioned space.

Industrial air cleaning mainly involves the removal of dust and gaseous contaminants from the industrial manufacturing processes, and it provides pollution control to the exhaust gas and flue gas. Air contaminants that are discharged to the outdoor environment are governed by the U.S. Environmental Protection Agency (EPA), and those that are exhausted to the indoor working space are regulated by the Occupational Safety and Health Administration (OSHA). The size of airborne

particles may range from $< 1\mu$m to several hundred micrometers. The amount of airborne particles present in the airstream often varies from several to 40,000 mg/m^3.

In this chapter, only air filtration is discussed.

Atmospheric Dust and Space Air Contaminants

The materials other than the oxygen, nitrogen, carbon dioxide, water vapor, and rare gases present in the air are called *air contaminants.* They are a mixture of dust, smoke, fumes, fogs, mists, and living organisms. Dusts are defined as solid granular particles and fibers less than 100 μm (1 μm = 0.00004 in.) derived from natural and mechanical processes. Smoke is a solid, a liquid, or a mixture of solid and liquid particles of sizes averaging from 0.1 to 0.3 μm. It is the product of incomplete combustion. Fumes are solid particles smaller than 1 μm formed in the condensation of vapors. Fogs are very small liquid particles, ranging from 2 to 60 μm, formed by the condensation of the vapors; and mists are relatively large liquid particles, from 60 to 200 μm, produced by atomizing and spraying processes. Living organisms include viruses, from 0.003 to 0.06 μm, and bacteria, usually from 0.4 to 5 μm. Both of these attach themselves to a larger particle for transportation. Fungal spores and pollens, which derive from living organisms, range from 10 to 100 μm. The mixture of granular particles, fibers, smoke, fumes, and mists is often collectively called dust or particulates, and a mixture of air and dust is called an aerosol.

Airborne particles that are smaller than 0.1 μm remain suspended and moving owing to brownian movement, whereas particles that range from 0.1 to 1 μm have a negligible settling velocity. Particles between 1 and 10 μm have a constant and significant settling velocity and tend to drop onto a solid surface. Particles that are larger than 10 μm settle on a surface even more rapidly.

Atmospheric dust denotes dust contained in the outdoor air, or outdoor atmosphere. According to Sec. 4.10, concentration of particle matter in atmospheric air should meet the EPA Ambient Air Quality Standard of 260 mg/m^3 within a period of 24 h. In atmospheric dust, (1) 99 percent of the dust particles are smaller than 0.3 μm, and they constitute only 10 percent of the total weight; and (2) 0.1 percent of the dust particles are larger than 1 μm and make up 70 percent of the total weight.

The dust concentration of an indoor conditioned space may vary considerably according to the requirements of the space, the sources releasing dust particles, the type of equipment and devices used in removing the dust particles, and the type of building enclosing the conditioned space. The concentration of total airborne particles of a conditioned space usually varies, from the very low particle count in a clean room to about 0.2 mg/m^3.

Rating and Assessments

An air cleaner is a device that removes dust particles from the airstream passing through it. An air filter is an air cleaner that removes dust particles from a conditioned airstream by means of fibrous, metallic, or other filter media. Filter media are fabricated materials to perform air filtration.

The operating characteristics of an air filter depend on the size, shape, specific gravity, concentration, and electric properties of the dust particles as well as the filter medium. However, the most important factor for effective removal is the particle size, or size distributions, in a dust sample. For air filtration in air conditioning systems, the dust particles are small and their concentrations are comparatively low.

The rating and assessment of air filters are mainly based on the following characteristics:

- *Efficiency*. The efficiency of an air filter shows the effectiveness of the dust removal process and, therefore, is the primary rating index. The efficiency of a specific filter is affected by the size of the dust particles, dust-holding capacity m_{dust}, or loading, in gr/ft^2 (g/m^2) and the test methods.
- *Pressure drop*. Total pressure drop of an air filter Δp_t, in in. WC (Pa), can be calculated as

$$\Delta p_t = C\left(\frac{v_f}{4005}\right)^2 = C_a\left(\frac{v_a}{4005}\right)^2 \tag{15.48}$$

where C, C_a = local loss coefficients of air filter
v_f = air velocity flowing through filter media, fpm
v_a = face velocity of air filter, fpm

The pressure drop of an air filter is affected by v_f or v_a, as well as the loading of dust held inside the air filter m_{dust}. The greater v_f, v_a, and m_{dust}, the higher Δp_r. At the rated volume flow rate, the pressure drop of a clean air filter when it first starts to collect dust particles is called the initial pressure drop.

- *Service life.* The pressure drop of an air filter which is about to be replaced or cleaned is called its final pressure drop. The operating period between the initial and final pressure drop is called the service life of an air filter. When the pressure drop of an air filter is at its final pressure drop, the dust-holding capacity of the air filter is usually the maximum dust-holding capacity $m_{\text{max, }d}$.

The efficiency of an air filter is significantly affected by the service life, or loading, of an air filter. The rating and assessment of a specific air filter at a specific volume flow rate require complete data on its efficiency and initial and final pressure drops.

15.13 TEST METHODS FOR AIR FILTERS

Test Methods

Since the 1940s, three test methods were widely used for air filters: weight arrestance, dust spot, and di-octyl phthalate (DOP) methods.

Weight Arrestance Method. In ANSI/ASHRAE Standard 52.1-1992, *Gravimetric and Dust Spot Procedure for Testing Air Cleaning Devices Used in General Ventilation for Removing Particulate Matter,* standard synthetic dust particles are used as the test dust in the weight arrestance method. By measuring the weight of dust fed and the weight gain on the membrane filter after the tested air filter, the arrestance can be calculated in percentage as

$$\text{Arrestance} = 100\left(1 - \frac{W_{\text{mf}}}{W_{\text{df}}}\right) \tag{15.49a}$$

where W_{mf} = weight gain of membrane filter, gr (g)
W_{df} = weight of dust fed, gr (g)

This method primarily determines the efficiency of a coarse or low-efficiency air filter to remove large dust particles present in the airstream.

Dust Spot Method. The dust spot test method in Standard 52.1-1992 uses an undefined and uncontrolled test dust, the atmospheric air. By measuring the change in light transmission of the discolored white filter paper before and after the test air filter, the efficiency of the air filter can be calculated as

$$E = 100\left[1 - \frac{\dot{V}_1(T_{20} - T_{21})(T_{10})}{\dot{V}_2(T_{10} - T_{11})(T_{20})}\right] \tag{15.49b}$$

where $\dot{V}_1, \dot{V}_2$ = volume flow rate of air drawn through upstream and downstream papers, cfm (L/s)
T_{10}, T_{11} = initial and final light transmission of upstream filter paper
T_{20}, T_{21} = initial and final light transmission of downstream filter paper

This method is widely used for medium-efficiency air filters.

DOP Method. The DOP test method was developed for the military to test the performance of high-efficiency particulate air (HEPA) filter. According to U.S. Military Standard MIL-STD-282 (1956), a smoke cloud of uniform di-octyl phthalate (DOP) droplets 0.18 μm in diameter is generated from the condensation of DOP vapor as the test dust. DOP vapor produced by thermal generation is oily liquid particles. By measuring the concentration of these particles in the air sample upstream and downstream from the tested air filter with an electronic particle counter, the penetration P and the efficiency E of the filter, both in percent, can be calculated as

$$P = 100 \frac{\text{downstream concentration}}{\text{upstream concentration}}$$
$$E = 100 \frac{1 - \text{penetration}}{100} \tag{15.50}$$

Unfortunately, confusion had arisen because of the uses of undefined and uncontrolled test dust have been used to assess air filters with 1930s discoloration technology. ANSI/ASHRAE Standard 52.2-1999, *Method of Testing General Ventilation Air Cleaning Devices for Removal Efficiency by Particle Size*, uses particle size and the varying-loading life-cycle method to assess air filter efficiency. The primary characteristics of the new test method are as follows:

- The efficiency of an air cleaner is determined as a function of dust particle size to be removed from the airstream by an electronic particle counter. The performance curve of an air filter is determined according to 12 size ranges from 0.3 to 10 μm.
- The particle removal efficiency is evaluated over a multiple-loading life cycle between initial (clean filter) and five loading stages. These efficiency curves including the minimum performance (initial clean filter) are the basis of a composite curve, as shown in Fig. 15.35.

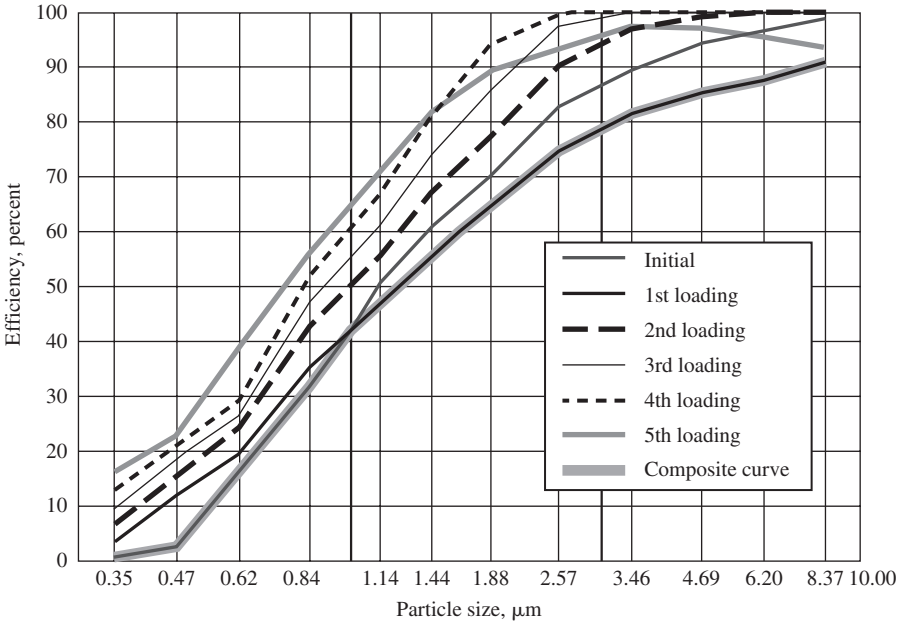


FIGURE 15.35 Composite efficiency curves of an air filter. (*Source: HPAC, August 1997. Burroughs, H.E.B. Reprinted by permission.*)

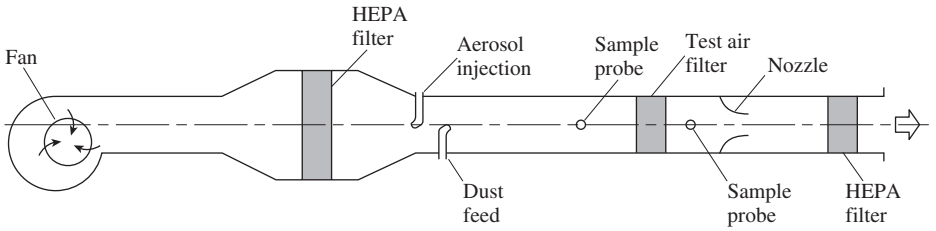


FIGURE 15.36 ANSI/ASHRAE Standard 52.2 specified air filter test unit.

- The test method uses laboratory-generated potassium chloride dispersed into the airstream as the test aerosol.
- Air filters are classified according to their minimum efficiency report value (MERV) into five categories: coarse, low, medium, high, and ultrahigh efficiency.

The weight arrestance test method to assess coarse filters, the new particle size method to assess low-, medium- and high-efficiency filters and DOP method for filters used in clean rooms and clean spaces. Table 15.3 lists MERV for air filters.

Test Unit

The performance of an air filter is tested in a test unit. Figure 15.36 shows an air filter test unit specified by ANSI/ASHRAE Standard 52.2-1999. The test unit consists of a blow-through fan, a test duct, a tested filter, two samplers, a particle counter, a flow nozzle, two sets of Pitot tubes and

TABLE 15.3 Minimum Efficiency Reporting Value (MERV) Parameters

Standard 52.2 minimum efficiency reporting value (MERV)	Composite average particle size efficiency, % in size range, μm			Average arrestance, % by Standard 52.1 method	Minimum final resistance	
	Range 1 0.30–1.0	Range 2 1.0–3.0	Range 3 3.0–10.0		Pa	in. of water
1	n/a	n/a	$E_3 < 20$	$A_{avg} < 65$	75	0.3
2	n/a	n/a	$E_3 < 20$	$65 \le A_{avg} < 70$	75	0.3
3	n/a	n/a	$E_3 < 20$	$70 \le A_{avg} < 75$	75	0.3
4	n/a	n/a	$E_3 < 20$	$75 \le A_{avg}$	75	0.3
5	n/a	n/a	$20 \le E_3 < 35$	n/a	150	0.6
6	n/a	n/a	$35 \le E_3 < 50$	n/a	150	0.6
7	n/a	n/a	$50 \le E_3 < 70$	n/a	150	0.6
8	n/a	n/a	$70 \le E_3$	n/a	150	0.6
9	n/a	$E_2 < 50$	$85 \le E_3$	n/a	250	1.0
10	n/a	$50 \le E_2 < 65$	$85 \le E_3$	n/a	250	1.0
11	n/a	$65 \le E_2 < 80$	$85 \le E_3$	n/a	250	1.0
12	n/a	$80 \le E_2$	$90 \le E_3$	n/a	250	1.0
13	$E_1 < 75$	$90 \le E_2$	$90 \le E_3$	n/a	350	1.4
14	$75 \le E_1 < 85$	$90 \le E_2$	$90 \le E_3$	n/a	350	1.4
15	$85 \le E_1 < 95$	$90 \le E_2$	$90 \le E_3$	n/a	350	1.4
16	$95 \le E_1$	$95 \le E_2$	$95 \le E_3$	n/a	350	1.4

Note: The minimum final resistance shall be at least twice the initial resistance, or as specified above, whichever is greater.
***Source*:** ANSI/ASHRAE Standard 52.2–1999. Reprinted with permission.

inclined manometers, two ultrahigh-efficiency air filters, and other necessary instruments and accessories. Laboratory-generated dust is fed into the test unit from one end. The blow-through fan is mounted at one end of the unit to maintain a positive pressure in the test unit. For details, refer to Standard 52.2-1999.

15.14 AIR FILTERS

Filtration Mechanism

The removal or collection of dust particles in air filtration is performed by various combinations of the following mechanisms.

- *Inertial impaction.* A sudden change in direction causes a collision between the dust particles, and fibrous media.
- *Straining.* If the filter spaces are smaller than the size of the dust particles, the particles are trapped.
- *Diffusion.* For very fine dust particles, brownian movement causes the particles to settle.
- *Interception.* Dust particles may follow the airstream, contact the fibrous media, and remain there.
- *Electrostatic effects.* Particles and the filter medium are charged to collect the dust in the airstream.

Classification of Air Filters

Except for ultrahigh efficiency filters, each category is subdivided into four MERVs, such as coarse air filters into MERV 1, 2, 3 and 4, low-efficiency into MERV 5, 6, 7 and 8, medium-efficiency into MERV 9, 10, 11 and 12, and high-efficiency into MERV 13, 14, 15, and 16.

Coarse Air Filters

Coarse air filters are often used to remove coarse dusts (5 to 80 μm) such as standing dust on surfaces, pollen, and textile fibers. They are generally panel filters of MERV 1, 2, 3 and 4 with weight arrestance efficiencies between 65 and 75 percent. The dimensions of the framework of these filters are often 20-in. (500-mm) width by 20-in. (500-mm) height, or 24-in. (600 mm) width by 24-in. (600-mm) height. The depth of the filter may vary from $\frac{1}{2}$ to 4 in. (13 to 100 mm).

Filter media for coarse filters are divided into three categories:

- *Viscous and reusable.* Some examples are corrugated wire mesh and screen strips. They are usually coated with oil, which acts as an adhesive, to increase their dust removal ability. Detergents may be used to wash off the dust when the filter media are to be cleaned and reused.
- *Dry and reusable.* Certain materials such as synthetic fibers (nylon, terylene) and polyurethane foam can be cleaned or washed if reuse is required.
- *Dry and disposable.* Synthetic and cellulose fibers, and bonded glass fiber mats with most of the glass fibers greater than 10 μm in diameter belong to this category. The air filter is discarded as soon as the final pressure drop is reached.

The face velocity of panel filters usually lies between 300 and 600 fpm (1.5 and 3 m/s). The minimum final pressure drop is 0.3 in. WC (75 Pa).

Coarse air filters are widely used in window air conditioners and residential small packaged units.

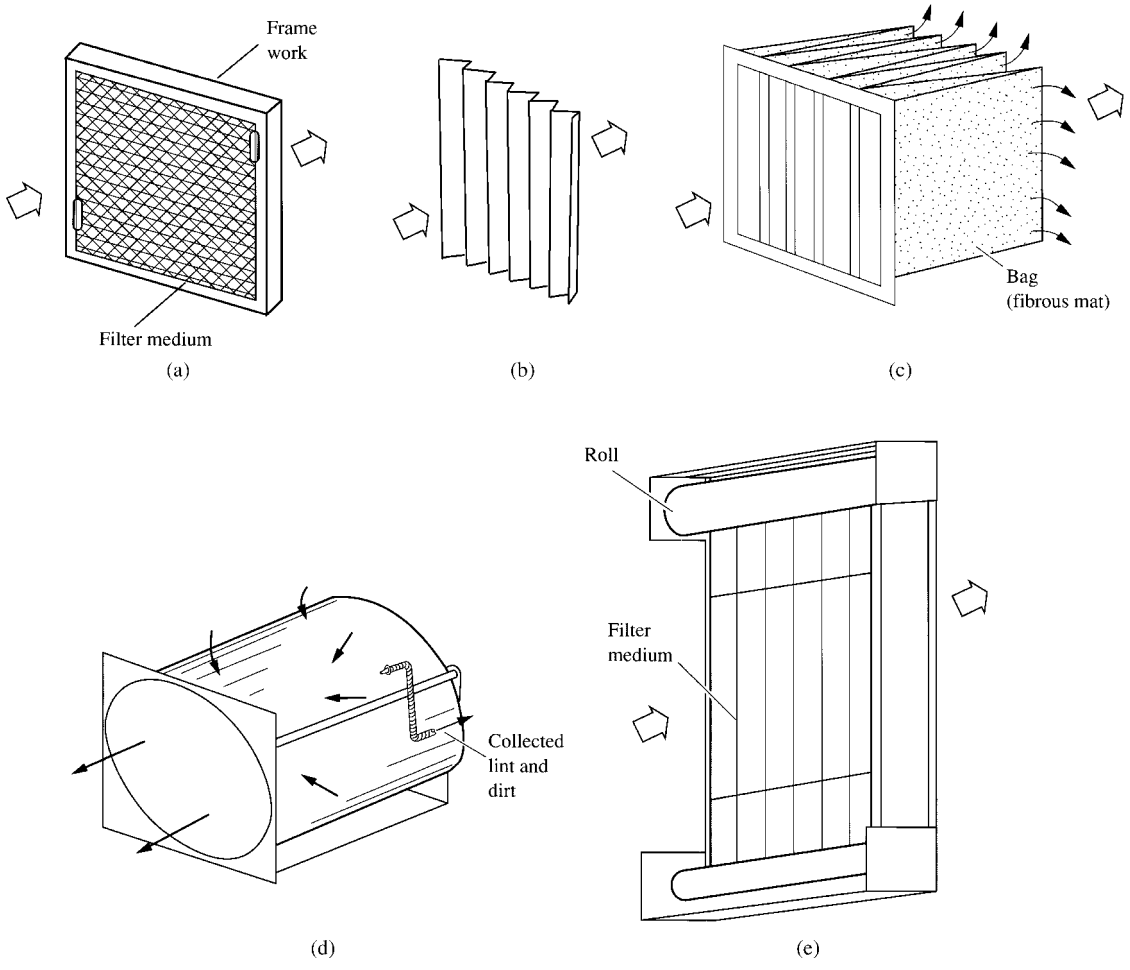


FIGURE 15.37 Types of air filters: (*a*) panel filter; (*b*) pleated filter; (*c*) extended surface (bag type); (*d*) rotary filter; (*e*) automatic renewable rolling filter.

Low-Efficiency Air Filters

Low-efficiency air filters including MERV 5, 6, 7 and 8 are often used to remove dusts between 3 and 10 μm, such as spores, molds, hairspray, cement, and other solid particles. Low-efficiency air filters are often pleated filters or sometimes bags as shown in Fig. 15.37*b* and *c*. They often have the same size of framework as in panel filters. The pleated mat is used to extend the surface area of the filter media.

In low-efficiency air filters, natural and synthetic fibers including glass fibers, viscous corrugated wire mesh, bonded glass fibers, and electrostatically discharged fibers or fabrics called electret have been used as filter media.

- For synthetic fibers and bonded glass fibers, their diameters may vary from 1 μm to several μm. As the efficiency increases, the diameter of the fiber is reduced and they are spaced comparatively closer together.

- Most low-efficiency filters have a face velocity of 500 fpm (2.5 m/s) to match the face velocity of the coil in AHUs or PUs.
- Pleated filters often extend their filter area 2 to 8 times their face area, so that the air velocity penetrating the filter media is reduced to 60 to 250 fpm (0.24 to 0.5 m/s). A lower air velocity will eventually increase the diffuse effect of the small particles and decrease the pressure drop across the filter medium.
- There are three types of filters using permanently electrostatically charged fibers and fabrics which increase the collection of small dust particles: (1) charges on the resin wool are produced by friction during the carding process; (2) a corona charge emits positive charges on one side of a thin polypropylene film and negative charges on the other side, and the film is shredded into fibers; and (3) it spins a liquid polymer into fiber under the effect of a strong electric field. Because of the combined effect of the inertial impaction, straining, diffusion, and local electrostatic effect of electret filters, their efficiency of dust removal is enhanced. Dust loading may reduce the efficiency of electret filters.

Low-efficiency air filters have a final minimum pressure drop of 0.06 in. WC (150 Pa). Low-efficiency filters are widely used in packaged units and air-handling units in commercial and institutional buildings, industrial workplaces, and more demanding residential applications.

Medium-Efficiency Air Filters

Medium-efficiency air filters including MERV 9, 10, 11 and 12 are used to remove dusts of size 1 to 3 μm such as welding fumes, Legionella bacteria and coal dusts. They are often bag and box filters with pleated mat to extend surface area, and are made of synthetic fibers including bonded glass fibers of diameters from <1 μm to several micrometers. MERV 9, 10, 11 and 12 have a 1 to 3 μm particle size efficiency from $E_2 < 50$ percent (corresponding to a dust spot efficiency of 40 to 45 percent up to $E_2 > 80$ percent (corresponding to a dust spot efficiently of 70 to 75 percent), and a minimum final pressure drop of 1 in. WC (250 Pa). Medium-efficiency air filters are used in demanding commercial building and industrial applications.

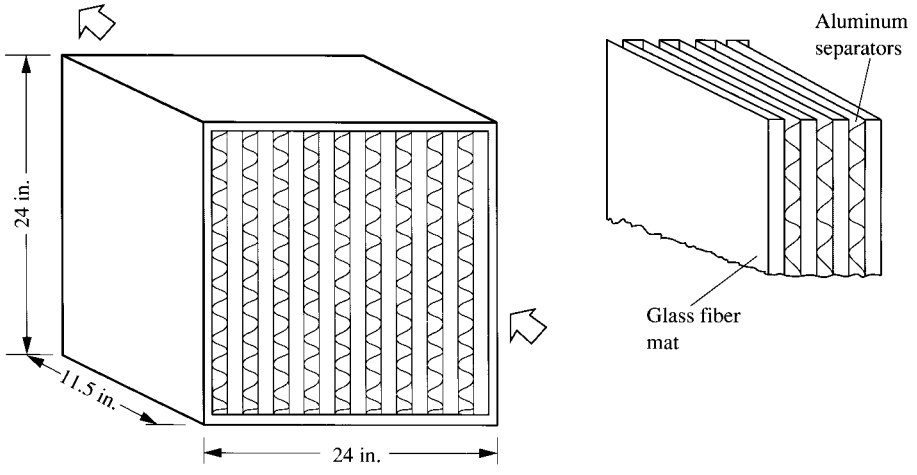


FIGURE 15.38 Ultrahigh-efficiency particulate filter (box filter).

High-Efficiency Air Filters

High-efficiency air filters including MERV 13, 14, 15, and 16 are used to remove particles of 0.3 to 1 μm such as bacteria, viruses, cooking oil fumes, tobacco smoke, and other smoke.They have a 0.3 to 1μm particle size efficiency $E_1 < 75$ percent (corresponding to a dust spot efficiency of 80 to 90 percent) up to $E_1 > 95$ percent. The filter media are often made of glass fibers of submicrometer and micrometer diameter. They are often in the form of a pleated mat in a cartridge, as shown in Fig. 15.38, or in the form of a bag filter. The air velocity flowing through the filter media is lower, and the minimum final pressure drops across the filter media are 1.4 in. WC (350 Pa). High-efficiency air filters are often protected by low- or medium-efficiency prefilters to extend their service life. High-efficiency filters are widely used in air systems for hospitals, high-demand commercial buildings, and precision manufacturing workshops.

Ultrahigh-Efficiency Air Filters

Ultrahigh-efficiency filters include high-efficiency particulate air (HEPA) filters, ultralow penetration air (ULPA) filters, and gaseous adsorbers and chemisorbers. Activated carbon adsorbers are discussed in Sec. 15.16, and chemisorbers are discussed in Sec. 24.5.

HEPA filters have an efficiency of 99.97 percent for dust particles ≥ 0.3 μm using the DOP test method. ULPA filters have an efficiency of 99.999 percent for dust particles ≥ 0.12 μm using the DOP method. A typical HEPA filter is shown in Fig. 15.38. The dimensions of this filter are 24 in. by 24 in. by 11.5 in. (600 by 600 by 287 mm). The filter media are made of glass fibers of submicrometer diameter that are formed into pleated paper mats. Some of the larger fibers act as the carrier of the web. The performance of the filter medium is often assessed by an index called the alpha value α, which can be calculated as follows:

$$\alpha = \frac{(2 - \log P)100}{25.4\ \Delta p_t} \tag{15.51}$$

where Δp_t = pressure drop of the filter medium, in. WC (Pa). Both penetration P, in percent, and pressure drop Δp_t are measured at an air velocity flowing through the filter medium of 10.5 fpm (0.05 m/s). The α value is usually between 10 and 11. Because of the development of new media with lower pressure drops, an α value of 13 or even higher can now be achieved. The surface area of the filter medium may be 50 times the face area of the ultrahigh-efficiency filter, and the rated face velocity may vary from 190 to 390 fpm (0.95 to 1.95 m/s) for ultrahigh-efficiency filters at a pressure drop of 0.65 to 1.35 in. WC (162 to 337 Pa) for clean filters. The face velocity of high-capacity ultrahigh-efficiency filters can be raised to 500 fpm (2.5 m/s).

The filter media themselves have efficiencies higher than that of the mounted filter. Sealing of the filter pack within its frame and sealing between the frame and gasket are critical factors that affect HEPA and ULPA filter penetration and efficiency. Penetration of dusts represents aerosol passing through the medium and through pinholes in the filter medium, as well as leaks between the pack and the frame and between the frame and the gasket.

To extend the service life of an ultrahigh-efficiency filter, it should be protected by either a medium-efficiency filter or two filters: a low-efficiency filter and a medium-efficiency filter located just upstream from the ultrahigh-efficiency filter. The removal of large particles in the prefilter reduces the dust load and prolongs the life of the ultrahigh-efficiency filter.

HEPA and ULPA filters are used to remove air contaminants such as unattached viruses, carbon dust, combustion smoke, and radon progeny of particles in sizes ≤ 0.3 μm. They are widely used in clean rooms and clean spaces for the microelectronics industry, pharmaceutical industry, precision manufacturing, and operating theaters in hospitals.

15.15 ELECTRONIC AIR CLEANERS

An electronic air cleaner uses the attraction between particles of opposite charges. Dust particles charged within the cleaner attract and agglomerate to greater sizes at the collecting plates. They are therefore easily removed from the airstream.

A typical electronic air cleaner is shown in Fig. 15.39. A high dc potential of 1200 V is supplied to the ionizing field. The positive ions generated from the ionizer wire charge the dust particles. Right after the ionizing section, the dust particles come to a collecting section, which consists of several plates that are alternately grounded and insulated. A strong electric field is produced by supplying a dc potential of 6000 V to these plates. The positively charged dust particles are attracted by the grounded plates of opposite charge, and attach themselves to the plates.

Because of the numerous points of contact, the bond between particles held together by intermolecular forces is greater than that between the particles and plates. Therefore, the dust particles agglomerate and grow to such sizes that they are blown off and carried away by the airstream. The agglomerates are then collected by a medium-efficiency air filter located downstream from the collecting section.

Kemp et al. (1995) showed that electronic air cleaners are efficient for removing small particles of 0.5 to about 8 μm. When dust particle sizes were approached and exceeded 10 μm, the filter's efficiency dropped from its highest peak. In a year-long test with the outdoor ambient air, no microbial growth was observed on any of the filters.

The pressure drop across the ionizer section and collecting section is low and ranges from 0.15 to 0.25 in. WC (37 to 62 Pa) against an air velocity of 300 to 500 fpm (1.5 to 2.5 m/s). Safety

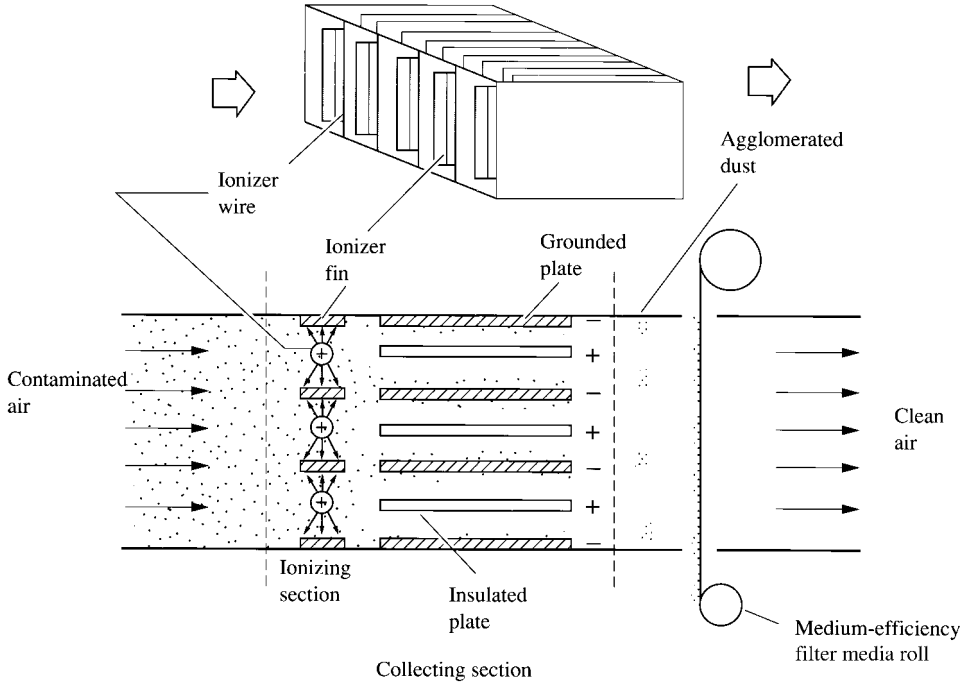


FIGURE 15.39 An electronic air cleaner.

measures must be provided for protection against the high dc potential. If the positive charges are not removed at the collecting section, the accumulation of these positively charged dust particles may build up a space charge. This space charge drives the charged particles to the walls and other building envelope surfaces in the conditioned space, causing them to be smudged.

15.16 ACTIVATED CARBON ADSORBERS

Granular activated carbon adsorbers are most widely used to remove objectionable odors and irritating vapors (including indoor volatile organic compounds, VOCs) of very small gaseous molecules from the airstream by adsorption. Adsorption is the physical condensation of a gas or vapor onto an activated substance. Activated substances are highly porous. One pound of extremely porous carbon contains more than 5,000,000 ft^2 (465,000 m^2) of internal surface. Gas molecules diffuse to the micropores or macropores of activated carbon, bond to these surfaces, and come in contact with the carbon granules. One pound (0.45 kg) of activated carbon may adsorb 0.2 to 0.5 lb (0.1 to 0.25 kg) odorous gases.

Activated carbon in the form of granules or pellets is made from coal, coconut shells, or petroleum residues. They are heated in steam and carbon dioxide to remove foreign matter such as hydrocarbons and to produce internal porosity. Granular activated carbon is placed in special trays, which slide easily into position, to form activated carbon beds that are sealed into the cell housing by face plates, as shown in Fig. 15.40. A typical carbon tray, which is 23 by 23 by $\frac{5}{8}$ in. thick (0.58 by 0.58 by 0.015 m thick), weighs 12 lb (5.4 kg). Low- or medium-efficiency air filters are used as prefilters for protection. When air flows through a typical assembly with a face velocity of 375 to 500 fpm (1.88 to 2.5 m/s), the corresponding pressure drops are between 0.2 and 0.3 in. WC (50 and 75 Pa). Activated carbon can also be mounted in fixed frames with perforated sheets in continuous pleats. Adsorption capacity is defined as the amount of carbon tetrachloride adsorbed by a given weight of activated carbon. For various odors, the adsorption capacity is also affected by the operating temperature and humidity. In general, a higher humidity or higher operating temperature

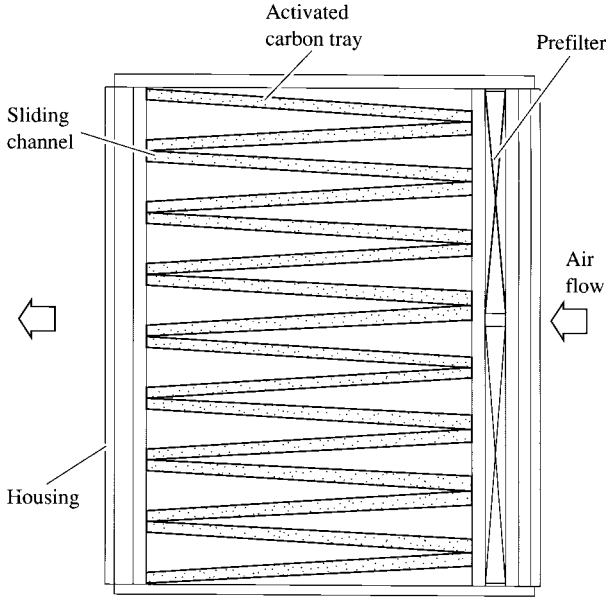


FIGURE 15.40 Granular activated carbon assembly.

usually decreases the adsorption capacity of the activated carbon. The maximum operating temperature is 100°F (37.8°C).

The removal efficiency of an activated carbon adsorber is the ratio of pollutant concentrations in the air upstream and downstream from the absorber. Toluene is used as the surrogate for total volatile organic compounds. The removal efficiency of the carbon bed remains relatively constant during its service life. Service life is the operating period between when an activated carbon absorber begins to remove pollutant and the time when the removal efficiency drops below an acceptable minimum. When the removal efficiency drops below an acceptable minimum, it must be replaced either by reactivation or regeneration. Reactivation is the process of removing spent carbon and replacing it with fresh carbon. Regeneration is the process by which spent carbon is converted to fresh carbon. Regeneration can only be performed by the activated carbon manufacturer. The simplest way to investigate whether the carbon needs to be replaced is to detect downstream the odor that the carbon filter is supposed to remove. A sample of carbon can also be tested to determine its remaining capacity.

Burroughs (1997b) reported that active particle fabrics demonstrate excellent indoor air contaminant control of VOC gases. These new fabric-based sorption filter media can be pleated into various configurations to maximize sorption capacity and airflow.

15.17 SELECTION OF AIR FILTERS

During the selection of air filters, electronic air cleaners, and activated carbon filters to remove air contaminants for the health and comfort of the occupants in indoor conditioned space, the following requirements and arrangements should be carefully considered:

- Characteristics of the air contaminants, the size of the particles, and the concentration of dust particles or irritating vapors to be removed must be identified.
- The degree of air cleanliness required in the conditioned space must be specified, especially the design criteria for clean spaces or clean rooms.
- The minimum efficiency of the air filter at specific particle sizes and loadings must be specified.
- The initial, average, and final pressure drops during the operating period, which affect the energy consumption and service life of the filter and the air system, must be determined.
- Service life of the air filter influences the installation cost, the pressure drop, and the efficiency of the air filter.

The following filtration tactics are recommended:

- Select an air filter based on the dust particle size MERV to remove air contaminants.
- Burroughs (1997) recommended the use of medium-efficiency particulate filters for protection of coils and air distribution systems, preventing the nutrition for biological growth; high-efficiency particulate filters for control of respirable particulate and bacteria; and gaseous absorbers to control objectionable odors and harmful volatile organic compounds.
- High-efficiency and ultrahigh-efficiency air filters, gaseous adsorbers such as granular activated carbon, and electronic air cleaners must be protected by a prefilter of medium efficiency to extend the life of these high-efficiency and ultrahigh-efficiency gaseous adsorbers and electronic air cleaners. However, the addition of two low- or two medium-efficiency filters does not significantly improve the efficiency to collect submicron dust and irritating vapors.
- Monitoring the pressure drop of the air filters and periodic maintenance of low-efficiency viscous air filters have a direct impact on filter performance.
- Muller (1995) reported that granular activated carbon is very good to remove most hydrocarbons, many aldehydes and organic acids, as well as nitrogen dioxide. It is not particularly effective against sulfur oxide, nitric oxide, formaldehyde, hydrogen sulfide, or lower-molecular-weight

aldehydes and organic acids. Potassium permanganate–impregnated alumina (chemisorbers are discussed in Sec. 24.5) is effective against sulfur and nitric oxides, hydrogen sulfide, and low-molecular-weight aldehydes.

15.18 HUMIDIFICATION AND HUMIDIFIERS

Space Relative Humidity

As mentioned in Sec. 4.9, for comfort air conditioning systems the space relative humidity is tolerable between 35 and 65 percent in summer, preferably from 40 to 50 percent. During winter, for space served by a comfort air conditioning system that is installed with a humidifier, the space relative humidity should not exceed 30 percent except in hospitals. The actual humidifying capacity of a humidifier should not exceed the humidifying requirement so that wet surfaces do not occur inside the AHU, PU, or supply ducts. Wet surfaces and dirt often cause the growth of microorganisms and thus indoor air quality problems.

For air conditioning systems without humidifiers, space relative humidity is usually not specified.

For processing air conditioning systems, the space relative humidity should be specified as required by the manufacturing process.

If the space temperature is maintained at 72°F (22.2°C) and only free convection is provided in the occupied space, the space relative humidity should not exceed 27 percent when the outdoor temperature is 30°F (−1.1°C), in order to prevent condensation on the inner surface of single-glazed windows during winter. To prevent condensation on the inner surface of double-glazed windows, the space relative humidity should not exceed 33 percent when the outdoor temperature is 0°F (−17.8°C).

Humidifiers

A humidifier adds moisture to the air. Humidifiers may (1) inject steam directly into air or add heat and evaporate steam from water supplied to the conditioned space; (2) atomize or spray liquid water, so that water evaporates and is added to the air; or (3) force air to flow over a wetted element so that as water evaporates, it is added to the air as vapor. All these increase the humidity ratio of the space air and, therefore, its relative humidity. One important index of a humidifier is its humidifying capacity $\dot{m}_{\text{cap}}$ in pounds of water per hour, lb/h (kg/h), or the rate at which water vapor is added to the air.

Humidifying Load

Humidifying load $\dot{m}_{\text{hu}}$, in lb/h or kg/h, is the amount of water vapor required to be added to the air by a humidifier so as to maintain a predetermined space relative humidity. The humidifying load of an air system installed with a humidifier can be calculated as (see Fig. 15.41)

$$\dot{m}_{\text{hu}} = 60\dot{V}_s\rho_s(w_{\text{lv}} - w_{\text{en}}) = 60\dot{V}_s\rho_s(w_s - w_m) \tag{15.52}$$

where $\dot{V}_s$ = supply volume flow rate of AHU or PU, cfm (m^3/min)
ρ_s = density of supply air, lb/ft^3 (kg/m^3)
$w_{\text{en}}, w_{\text{lv}}$ = humidity ratio of air entering and leaving humidifier, lb/lb (kg/kg)
w_s = humidity ratio of supply air, lb/lb (kg/kg)
w_m = humidity ratio of mixture of outdoor air and recirculating air, lb/lb (kg/kg)

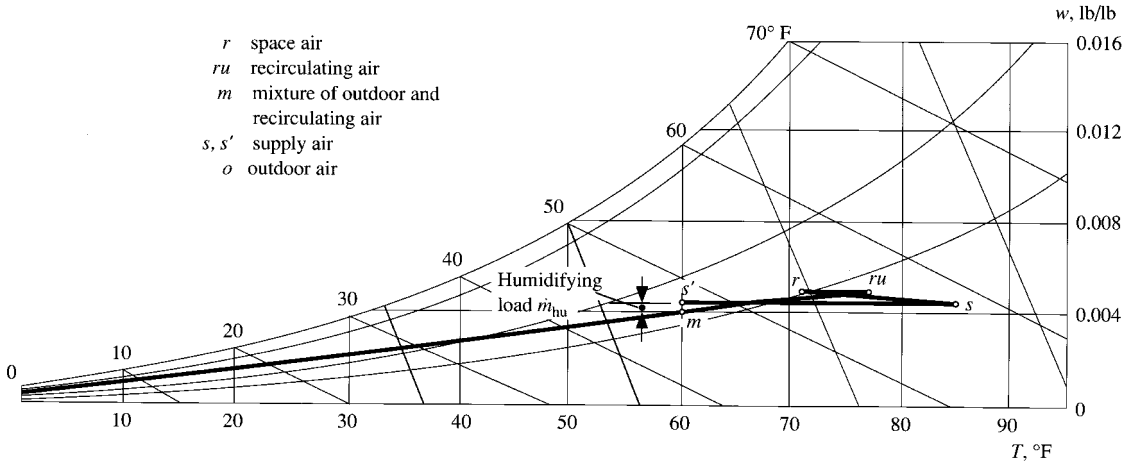


FIGURE 15.41 Humidifying load for a typical air system.

In Fig. 15.41, s is the state of supply air when the space has a heating load, and s' is the condition when the space has a cooling load, $w_s = w_{s'}$.

The humidifying load can also be calculated from the following relationship:

$$\dot{m}_{hu} = 60(\dot{V}_o + \dot{V}_{inf})\rho_o(w_r - w_o) - \dot{m}_{wr} \tag{15.53}$$

where $\dot{V}_o$ = supply volume flow rate of outdoor air intake, cfm (m^3/min)
$\dot{V}_{inf}$ = volume flow rate of infiltrated air, cfm (m^3/min)
w_r,w_o = humidity ratio of space air and outdoor air, lb/lb (kg/kg)

In Eq. (15.53), $\dot{m}_{wr}$ represents the space moisture gains, in lb/h (kg/h). Space moisture gains include the latent load from the occupants, appliances, equipment, and products. The moisture gains from the building structures are often ignored.

Types of Humidifier

According to the mechanism used for evaporation of water vapor from water, humidifiers can be classified as steam and heating element humidifiers, atomizing humidifiers, and wetted element humidifiers.

15.19 STEAM AND HEATING ELEMENT HUMIDIFIERS

At a low partial pressure when mixed with dry air, steam is the ready-made water vapor. With proper water treatment at the boiler, steam is free of mineral dust and odor, and it does not support the growth of bacteria that create sanitation problems. Two types of steam humidifiers are currently used in air systems: steam grid humidifiers and steam humidifiers with separators.

Steam Grid Humidifiers

A steam grid humidifier installed inside ductwork is shown in Fig. 15.42*a*. A steam grid humidifier may have a single distribution manifold or multiple manifolds. Each distribution manifold has an

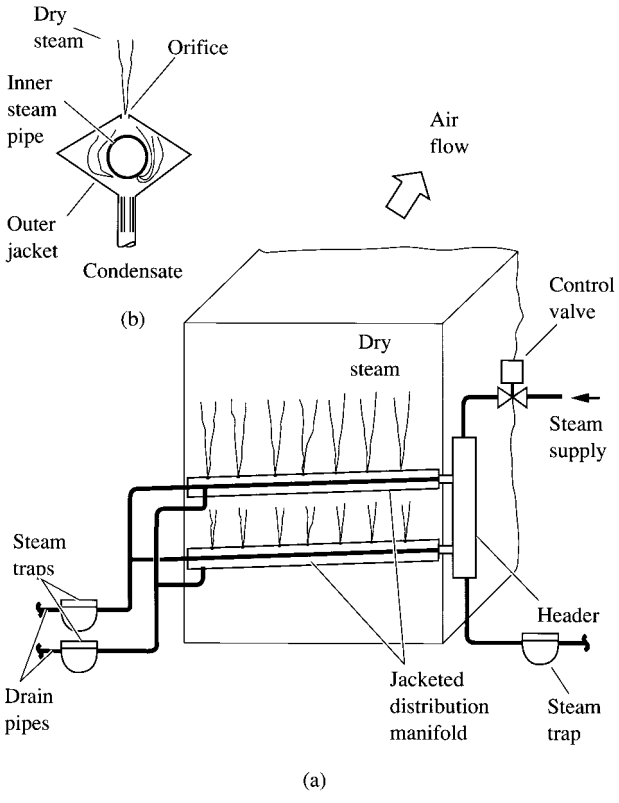


FIGURE 15.42 Steam grid humidifiers: (*a*) steam grid and piping connections; (*b*) cross section of jacketed distribution manifold.

inner steam pipe and an outer jacket. The inner steam tube is connected to a header and controlled by a control valve. Steam is supplied at a pressure less than 10 psig (69 kPag) and throttled to a lower pressure after the controlled valve. It then enters the inner tube and discharges through the small holes to the outer jacket. The dry steam is again discharged through the orifices of the jacket to the ambient airstreams to humidify them, as shown in Fig. 15.42*b*. The condensate inside the jacket is discharged to a drainpipe and steam trap located at the opposite side. The inner tube and the outer jacket are slightly pitched toward the drainpipe and steam trap.

Steam Humidifiers with Separators

A steam humidifier with a separator is illustrated in Fig. 15.43. Steam is supplied to a jacketed distribution manifold and then enters a separating chamber with its condensate. It then flows through a control valve, is throttled to slightly above atmospheric pressure, and enters a drying chamber. Because of the lower pressure and temperature in the drying chamber compared with the higher pressure and temperature in the surrounded separating chamber, the steam is superheated. Dry steam is then discharged into the ambient airstream through the orifices of the inner steam discharging tubes. Noise is produced mainly during the throttling of the steam at the control valve. It is attenuated as steam flows through the silencing materials.

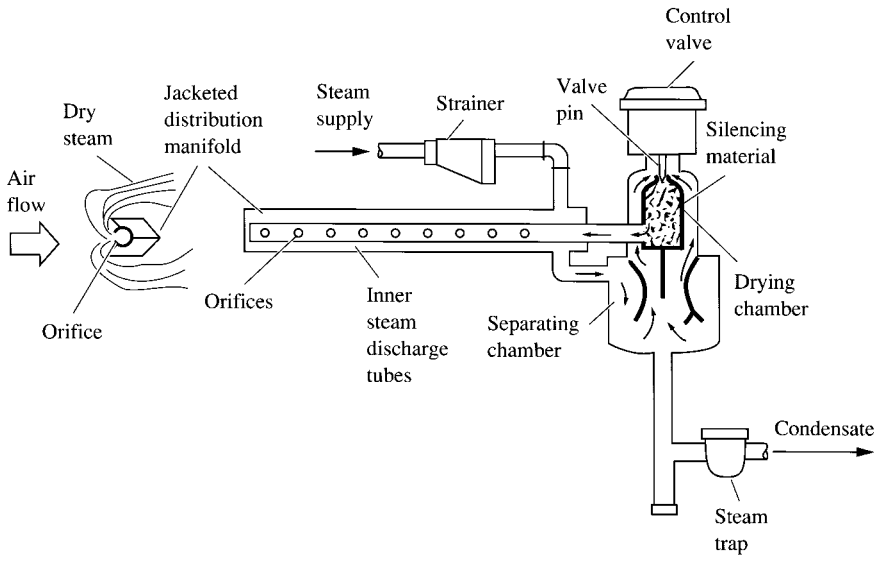


FIGURE 15.43 Steam humidifier with a separator. (*Adapted with permission from The Armstrong Humidification Handbook, 1991.*)

Heating Element Humidifiers

When a steam boiler is not available or is too expensive to install, steam humidification is still possible by using heating element humidifiers. A heating element humidifier has a water pan with electric or gas-fired heated elements installed at the bottom of the pan to evaporate the liquid water to water vapor and then add the vapor to the airstream flowing over the water pan, as shown in Fig. 15.44. The humidifying capacity of a heated element humidifier is limited by its heat-transfer surface and the characteristics of the heating element.

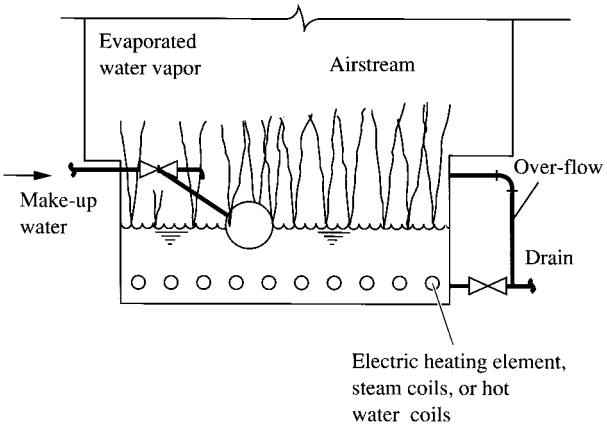


FIGURE 15.44 Heating element humidifier.

Characteristics and Requirements of Steam and Heating Element Humidifiers

Steam humidifiers are widely used in commercial and industrial applications. Both steam and heating element humidifiers need heat energy from gas, fossil fuel, or electricity to evaporate the liquid water in the steam generator. The temperature of the ambient air or airstream in which moisture has been added is approximately unchanged. Steam and heating element humidifiers are therefore often called isothermal humidifiers. For a conditioned space that has a cooling load year round and needs a cold air supply in winter, it is a waste of energy to evaporate the liquid water by using fossil fuel instead of the excess heat gains at the conditioned space.

Control is essential in steam and heating element humidifiers. The capacity of a steam humidifier is often controlled by a microprocessor-based controller by modulating the valve pin of the control valve according to the signal of a humidity sensor and, therefore, the steam flow rate. Morton (1996) reported an accuracy of humidity control of ± 5 to 7 percent for on/off control and ± 3 to 5 percent for modulation control. The size of the control valve should be carefully matched with the humidifying load with an adequate turndown ratio. The control valve should be integrated with the humidifier and steam-jacketed at supply pressure to prevent condensation. An interlocking control should be installed to drain all condensate before the steam humidifier is started.

For an electric heating element humidifier, or simply an electric humidifier, the input wattage can be adjusted when the space relative humidity is too high or too low. A gas-fired heating element humidifier, or simply a gas-fired humidifier, uses infrared burners located within a heat exchanger. A microprocessor-based controller stages and cycles the burner to vary the heat output and therefore the humidifying capacity during part-load operation.

The makeup water supply to the steam boiler and the water pan must be well treated so that the steam is free of mineral deposits and odor. For electric and gas-fired humidifiers, according to DeBat (1996), recent developments, such as ionic bed technology, overcome the problem of mineral deposits. Dissolved mineral salts begin to precipitate out of water when their concentration becomes too high. Ionic beds consist of inert fibers that attract these precipitated minerals. The mineral salts deposit on the ionic bed instead of on the walls of the heating chamber. When throughly encrusted with crystalline solids, the ionic bed cartridge is easily removed and discarded.

Steam humidifiers should be located where noise is not objectionable. Access to equipment for inspection and maintenance should be provided. Condensate drainage must be properly designed. If the steam humidifier is located in ductwork, install a duct high-limit humidistat and an interlocked airflow proving switch. The humidity sensor is located preferably in the return air duct or a representative location.

15.20 ATOMIZING AND WETTED ELEMENT HUMIDIFIERS

Humidification Process

Atomizing means producing a fine spray. When liquid water is atomized, the smaller the diameter of the water droplets, the greater the interfacial area between water and air and thus the higher the rate of evaporation and humidification.

When air flows through an atomized water spray from an atomizing humidifier, if the difference between the temperature of the water spray and the wet-bulb temperature of the ambient air is small, the result is an increase in the humidity ratio of air from the addition of evaporated water vapor and a corresponding drop in air temperature because of the absorbtion of required latent heat of vaporization from the ambient air. Such a humidification process is an adiabatic saturation or evaporative cooling process and follows the thermodynamic wet-bulb temperature line on the psychrometric chart.

Atomizing humidifiers can be classified, according to their configuration and mechanism of atomizing, as ultrasonic humidifiers, centrifugal atomizing humidifiers, pneumatic atomizing humidifiers, and air washers.

Ultrasonic Humidifiers

An ultrasonic humidifier uses a piezoelectric transducer to convert high-frequency mechanical electric signals to a high-frequency ocscillation. The transducer is submerged into a water pan in which water is well treated and demineralized. The negative oscillation produces a momentary vacuum that causes the water to cavitate into vapor at low pressure. The positive oscillation creates a high-compression wave which forces the water particles to leave the water surface and inject into the airstream. Water films are then broken into a fine mist. Air containing this mist can be blown into the space directly or through an air system by a fan.

If the water is well treated, it will be free of mineral deposits, and no filter is required downstream from the humidifier. Ultrasonic humidifiers create less equipment noise than other atomizing humidifiers. In the mid-1990s, ultrasonic humidification had been installed in telecommunication projects. Ultrasonic atomizing humidifiers with small humidifying capacities have been used directly in residences.

Case Study: White Plains Ultrasonic Humidification Project

Longo (1994) reported on a retrofit ultrasonic humidification project which is located in a 1950s building in White Plains, New York. The humidification system in a totally new air system serves space that houses solid-state complex multiswitching processors. Static charges created in a low-humidity environment can be detrimental to system components and cause failure.

The 100,000-cfm (47,200-L/s) air system consists of four supply fans (two standby), high-efficiency filters, and cooling coils with outside air economizer for free cooling. The ultrasonic humidifiers were placed after the filters and upstream from the supply fan. There are altogether 43 ultrasonic units mounted on stainless-steel racks, each rated at 39.6 lb/h (18 kg/h) and 960 W. A humidity sensor is installed in the return air duct. The controller turns the humidity system on one stage at a time up to a total of six stages. A high-limit sensor set at 80 percent relative himidity shuts down the system in case of overhumidification.

Water is treated by a series of prefilters, a reverse osmosis system, and deionization canisters. Water quality is controlled by a resistivity probe. When water resistance reaches 0.02 MΩ, the deionization canisters will be changed.

After completion, the actual running results in winter are excellent. As the equipment was installed gradually, the supply air temperature was rescheduled, and the space relative humidity was maintained precisely at 50 percent. The initial cost of the White Plains humidity system was $350,000. Annual operating cost was $23,000, which was less than one-tenth of the annal operating cost of an electric heating element humidity system.

Centrifugal Atomizing Humidifiers

These humidifiers use the centrifugal force produced by a rotating device such as a rotating cone, blades of an axial fan, rotating disk, or rotating drum to break the liquid water film into fine mist or to fling it into fine water droplets. When air is forced through an atomizing water spray, produced by a pulverizing fan, water vapor is evaporated and added to air as a result of the mass concentration difference between the saturated air film at the surface of the droplets and the ambient air. The rotating device receives liquid water either from a pressurized water supply or by dipping into the surface of a nonpressurized supply. The humidifying capacity of a pulverizing fan depends mainly on the volume flow rate of the fan. Because of the oversaturation characteristics of the pulverizing fan, $\dot{m}_{cap}$ may vary from 50 to 150 lb/h (23 to 68 kg/h). Oversaturation is the excess amount of water particles present in the moist air, and it is discussed in Chap. 20. Many different types of rotating humidifiers with limiting humidifying capacities are used directly in the conditioning space in residential and industrial applications. Most are portable.

Maintaining a predetermined space relative humidity with a centrifugal atomizing humidifier can be performed by modulating the flow rate of water supplied to the rotating devices.

Pneumatic Atomizing Humidifiers

A typical pneumatic atomizing nozzle consists of two concentric brass tubes, as shown in Fig. 15.45*a*. The inner tube has an outer diameter of $\frac{3}{32}$ in. (2.4 mm) and an inner diameter of $\frac{1}{32}$ in. (0.8 mm). It is connected to a water tank that can be moved to adjust the difference in water levels between the water tank and the centerline of the nozzle. The conical outer tube has a minimum inner diameter of $\frac{1}{8}$ in. (3.2 mm) and is connected to a compressed air line with a pressure at 15 psig (103 kPag).

When compressed air is discharged from the annular slot at a very high velocity, it extracts water from the inner tube and breaks the water into a very fine mist. A typical pneumatic atomizing nozzle may have a humidifying capacity $\dot{m}_{cap}$ from 6 to 10 lb/h (2.7 to 7.5 kg/h). The magnitude of $\dot{m}_{cap}$ is affected by the configuration of the nozzle, the pressure of the compressed air, and the difference in water levels. In a pneumatic humidifier, $\dot{m}_{cap}$ is usually controlled by adjusting the difference in water levels between the water tank and the centerline of the nozzle.

A pneumatic humidifier produces high-frequency noise. This type of humidifier is usually applied in industries with a certain sound level of machine noise and can be used for direct in-space humidification.

Wetted Element Humidifiers

Wetted element humidifiers include a wetted element, such as an evaporative pad, plastic, or impregnated cellulose, that is dipped with water from the top. Such humidifiers have been installed in air-handling units and packaged units to humidify the air. Characteristics of wetted element humidifiers are similar to those of wetted element evaporative coolers (see Chap. 27). Modulation of the water supply to the water dipping device varies the humidifying capacity and maintains the desirable space relative humidity.

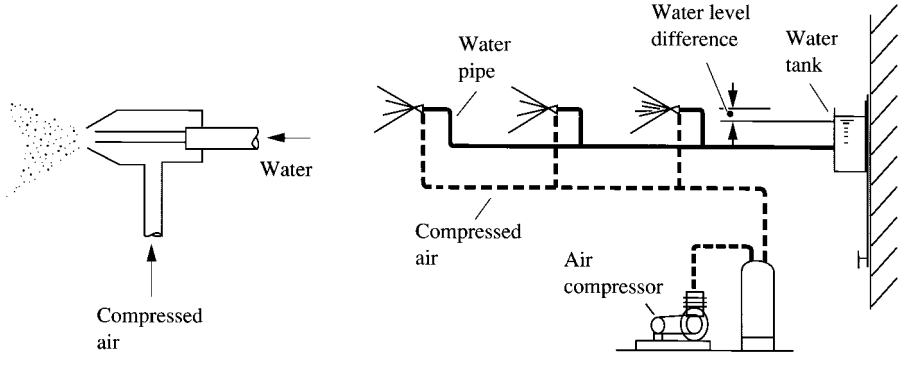


FIGURE 15.45 Pneumatic atomizing humidifier: (*a*) nozzle; (*b*) pneumatic atomizing humidifying system.

15.21 AIR WASHERS

The air washer was the first air conditioning equipment developed by Carrier in 1904, and it is still used today to humidify, cool, and clean the air in many factories.

Construction of an Air Washer

An air washer has an outer casing, shown in Fig. 15.46*a*. It is usually made of plastic or galvanized steel sheet with water-resistant paint for protection. All joints are well sealed by water-resistant resin. A water tank either forms a part of the casing at the bottom or is installed separately on the floor to collect or sometimes to mix the recirculating and incoming chilled water. A separately installed water tank is usually made of steel, stainless steel, or reinforced concrete with an insulation layer mounted on the outer surface if chilled water is used for spraying. A bank of guided baffles installed at the entrance provides an evenly distributed air-water contact. Eliminators in the shape of a sinusoidal curve at the exit are installed to remove entrained water droplets from the air. They are preferably made of plastic or stainless steel for convenient cleaning and maintenance. Access doors

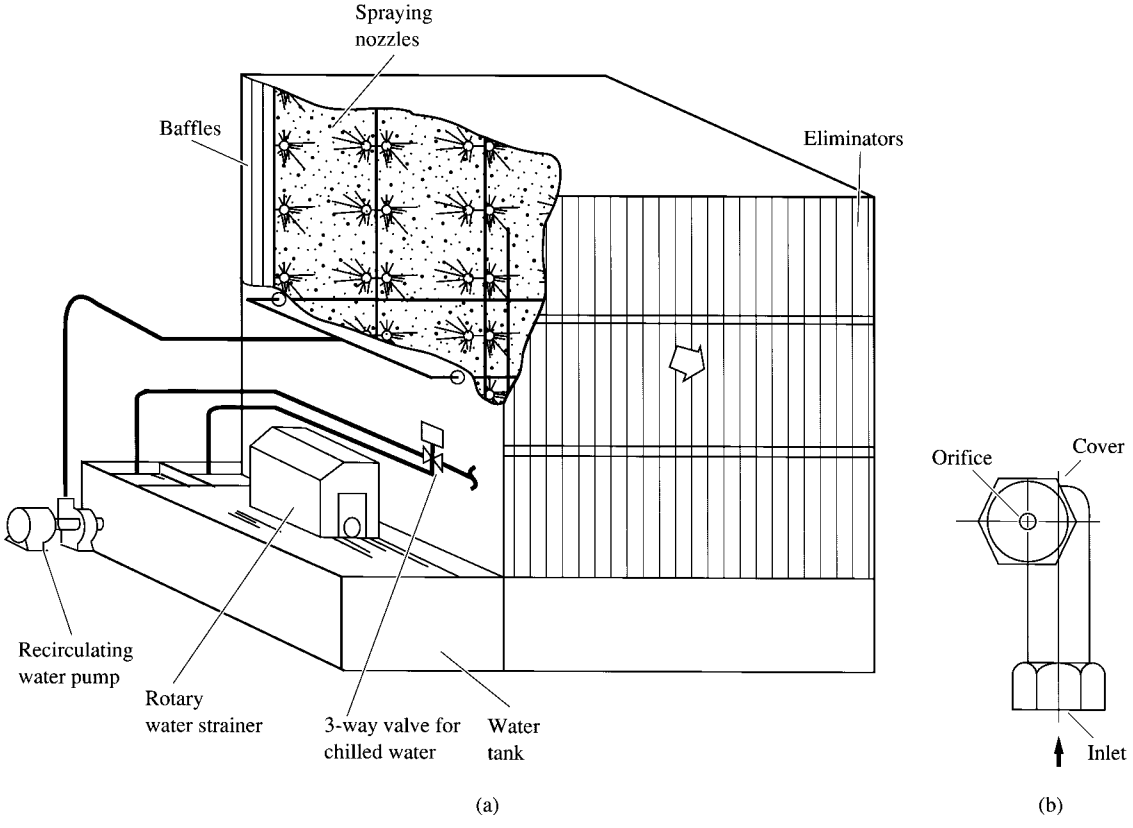


FIGURE 15.46 Schematic diagram of a typical air washer: (*a*) air washer; (*b*) spray nozzle.

mounted on the outer casing must be provided for inspection and maintenance. They should be well sealed to prevent water leakage.

In most air washers, two banks of spraying nozzles face each other. Nozzles are often made of brass, plastic, or nylon. The nozzle consists of a cover with an orifice diameter between $^1/_{16}$ and $^3/_{16}$ in. (1.6 and 4.8 mm), a discharge chamber, and an inlet with its centerline slightly eccentric from the centerline of the orifice, as shown in Fig. 15.46*b*. This offset causes the water stream to rotate inside the discharge chamber and breaks the water into fine droplets. A larger orifice diameter does not clog easily, and a smaller orifice can produce finer water sprays. The distance between two spraying banks is from 3 to 4.5 ft (0.9 to 1.35 m), and the total length of the air washer varies from 4 to 7 ft (1.2 to 2.1 m).

A centrifugal water pump is used to recirculate the spraying water. For air containing much dirt and lint, as in air conditioning systems in the textile industry, an automatic washing and collecting water strainer, using copper or brass fine-mesh screen, often precedes the pump. The opening of the fine-mesh screen should be smaller than the diameter of the hole of spraying nozzles to avoid clogging.

Functions of an Air Washer

Currently, air washers are used to perform one or more of the following functions:

- Cooling and humidification
- Cooling and dehumidification
- Washing and cleaning

Whether it is a cooling and humidification or cooling and dehumidification process is determined by the temperature of the spraying water. If recirculating water is used as the spraying water, the temperature of water then approaches the wet-bulb temperature of the air entering the air washer, and the air is humidified and evaporatively cooled.

If chilled water or a mixture of chilled water and recirculating water is used for spraying water, and its temperature is lower than the dew point of the entering air, the air is cooled and dehumidified.

Water spraying is sometimes practiced for the purpose of washing and cleaning air when an objectionable gas is known to be soluble in water and is to be removed from the air. For example, exhaust gas from many industrial applications may be removed by water spraying.

For humidification purposes, nozzles must have a smaller orifice and a higher water pressure, such as a $^1/_{16}$-in. (1.6-mm) diameter and a pressure of 40 psig (275 kPag). For cooling and dehumidification, a larger orifice and lower water pressure, such as a $^3/_{16}$-in. (4.8-mm) orifice and 25 psig (172 kPag), are often used.

Performance of an Air Washer

For a humidification process along the thermodynamic wet-bulb temperature line, the performance of an air washer can be illustrated by the saturation efficiency ε_{sat} and calculated as

$$\varepsilon_{sat} = \frac{w_2 - w_1}{w_s^* - w_1} \approx \frac{T_2 - T_1}{T_s^* - T_1} \tag{15.54}$$

where T_1, T_2 = temperature of air entering and leaving air washer, °F (°C)
w_1, w_2 = humidity ratio of air entering and leaving air washer, lb/lb (kg/kg)
w_s^*, T_s^* = humidity ratio and temperature of saturated air at thermodynamic wet-bulb temperature, lb/lb (kg/kg) and °F (°C), respectively

For a cooling and dehumidification process, the performance of an air washer is better described by a factor called the *performance factor* F_p as follows:

$$F_p = \frac{h_{\text{en}} - h_{\text{lv}}}{h_{\text{en}} - h_s^*} \tag{15.55}$$

where $h_{\text{en}}, h_{\text{lv}}$ = enthalpy of air entering and leaving air washer, Btu/lb (kJ/kg)
h_s^* = enthalpy of saturated air at chilled water spraying temperature, Btu/lb (kJ/kg)

The performance of an air washer for a humidification process, indicated by the saturation efficiency ε_{sat}, and for a cooling and dehumidification process, expressed by F_p, depends mainly on the contact time between air and water and the contact surface area, i.e., the water-air ratio $\dot{m}_w/\dot{m}_a$ (which indicates the mass flow rate of the spraying water to the mass flow rate of the air flowing through the air washer, in same units), the length of the air washer, and the direction of spray with respect to airflow, (whether it is opposing or following the airflow).

An air washer is usually designed at an air velocity between 500 and 800 fpm (2.5 and 4 m/s) with respect to its cross-sectional area at the water sprayers. Its greater washer length and the attached water tank result in a bulkier volume than that of DX coils and water cooling coils. Total pressure loss of the airstream flowing through an air washer $\Delta p_{t,w}$, in in. WC (Pa), depends mainly on the configuration of the eliminators and the air velocity flowing through them. Usually $\Delta p_{t,w}$ varies from 0.25 to 1 in. WC (62.5 to 250 Pa), and typically it is 0.5 in. WC (125 Pa). For humidification and evaporative cooling, the water-air ratio $\dot{m}_w/\dot{m}_a$ is usually 0.3 to 0.6. For a 5-ft (1.5-m) air washer with $\dot{m}_w/\dot{m}_a = 0.45$, that is, 4 gpm of water per 1000 cfm of air (0.534 L/s of water per 1000 L/s of air), $\varepsilon_{\text{sat}} = 0.85$ to 0.9. For cooling and dehumidification, the water-air ratio is usually 0.5 to 1.2. For a 5-ft- (1.5-m-) long air washer with $\dot{m}_w/\dot{m}_a = 0.9$, that is, 8 gpm/1000 cfm (1 L/s of water per 1000 L/s of air), $F_p = 0.5$ to 0.75.

For either the humidification process or the cooling and dehumidification process in an air washer, oversaturation always exists at the exit. Oversaturation ($w_o - w_s^*$) usually varies between 0.0002 and 0.001 lb/lb (0.0002 and 0.001 kg/kg). It depends mainly on the construction of the eliminator and the flow-through air velocity. Oversaturation is beneficial to a humidification process with space heat gains and interferes with a cooling and dehumidification process. Because of the existence of oversaturation, the saturation effectiveness is no longer the dominant index that affects the performance of the air washer during humidification. Oversaturation is discussed in Sec. 20.13.

Bypass Control

Bypass control is defined as modulating the face damper and the bypass damper of the air washer to vary the proportion of humidifying air and bypass air in order to maintain the space relative humidity within predetermined limits. Such a control is often used in an air washer to provide better humidity control than modulating the water pressure or the water flow rate to the spraying nozzles.

Single-Stage or Multistage

A group of nozzles that are supplied with water at the same pressure and produce spray at the same temperature are said to be in the same stage. A stage may have one or two spraying banks. A two-stage air washer is illustrated in Fig. 15.47.

For humidification processes or for cooling and dehumidification processes using chilled water from the refrigeration plant, it is more economical to operate a single-stage air washer, usually with two water-spraying banks. When low-temperature deep well or groundwater is used for cooling and dehumidification, it is often advantageous to use two-stage, or even three-stage, water spraying to make full use of the cooling capacity of the groundwater.

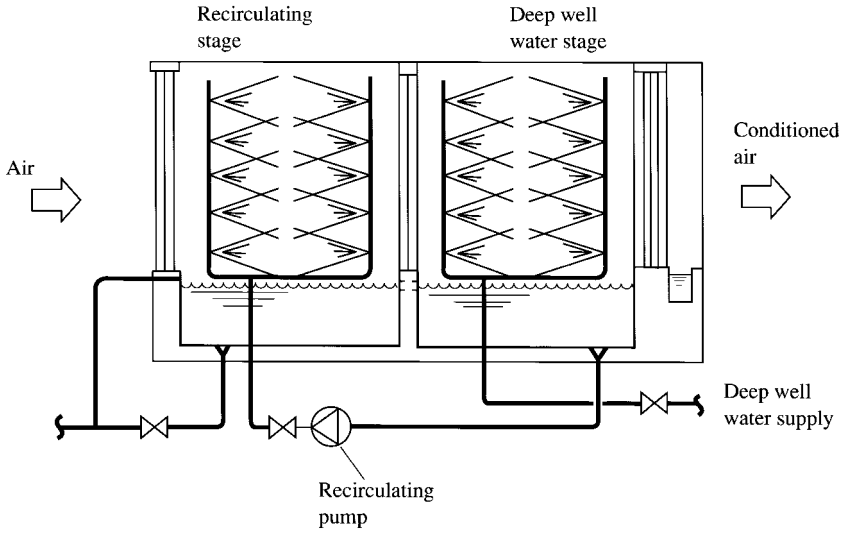


FIGURE 15.47 Two-stage air washer using deep well water.

Groundwater can be used again as condenser water, or as flushing water in plumbing systems after its discharge from the air washer following necessary water treatment. Used groundwater should be drained to another well (recharge well) in the vicinity to prevent subsidence and damage to the soil, which may affect the foundation of the building in the vicinity of the deep well.

The use of groundwater is regulated by local and federal environmental protection agencies.

15.22 CHARACTERISTICS OF ATOMIZING AND WETTED ELEMENT HUMIDIFIERS

Atomizing and wetted element humidifiers, including air washers for humidification purposes have the following operating characteristics:

- Conditioned air follows the thermodynamic wet-bulb temperature line on a psychrometric chart. The humidity ratio of the air increases while it is evaporatively cooled. Atomizing humidifiers are often called nonisothermal humidifiers. This characteristic indicates that atomizing and wetted element humidifiers are especially suitable for conditioned spaces that need a cold air supply to offset a space cooling load during or after the humidification process. The heat energy required to evaporate the liquid water, as in steam or heating element humidifiers, has been saved. For a conditioned space which needs a warm air supply during winter operation, the temperature of evaporatively cooled air after an atomizing or wetted element humidification process should be raised before supplying it to the conditioned space, so as to offset the transmission loss through the building envelope. The heat energy required is approximately equal to that required to evaporate water in a steam boiler or in a water pan when a steam humidifier or a heating element humidifier is used.
- For atomizing humidifiers that need a cold air supply during winter, when the humidifying capacity is modulated, both the space relative humidity and temperature vary accordingly. Many

industrial manufacturing processes need to maintain a stable space relative humidity (RH). RH is often the controlled variable, and temperature is usually maintained within predetermined limits.

- Oversaturation exists at the exit of the atomizing humidifier even if an eliminator is used. Oversaturation is often advantageous for humidifiers installed in conditioned spaces having excess heat gains because it increases the humidifying capacity.
- The size of the water droplets, the local humidifying capacity, and the excessive heat gains determine whether liquid water droplets may fall on the floor, products, or equipment. According to field measurements, the diameter of water droplets suspended in the airstream discharged from atomizing humidifiers is as follows:

  Pneumatic atomizing humidifiers 30 to 80 μm

  Centrifugal atomizing humidifiers 50 to 300 μm

  When the diameter of the water droplets exceeds 80μm and the local humidifying capacity is excessive, water may drip. Water dripping must be avoided, even if reducing the humidifying capacity is necessary. It causes product damage and corrosion of appliances, instruments, and equipment.
- In the water tank and water pan of atomizing and wetted element humidifiers, algae, bacteria, and other microorganisms may grow. Water spray and the airstream discharged from the atomizing and wetted element humidifiers may distribute the bacteria and sometimes odors. Water treatments, regular cleaning and flushing, and periodic inspection and blowdown help to minimize these problems. The designer must include convenient access for such maintenance.

15.23 SELECTION OF HUMIDIFIERS AND DESIGN

The following factors should be considered in selecting a humidifier:

- *Energy consumption and operating cost.* For a space that needs a cold air supply even in winter, atomizing and wetted element humidifiers save energy and, therefore, have lower operating costs. Among the atomizing humidifiers, pneumatic atomizing humidifiers require compressed air to atomize the liquid water and, therefore, have a higher operating cost than other atomizing humidifiers.
- *Quality of humidification.* Humidified air should be clean and free of odor, bacteria, particulate matter, and water droplets to prevent microbial growth due to wetted surfaces or standing water and improving IAQ.
- *Humidifying capacity.* Is a large, medium, or small humidifying capacity required? Is oversaturation beneficial in the conditioned space?
- *Capacity control.* Is this an on/off control or a modulation control, i.e., a stepless control? Does the control have a fast or slow response? What is the turndown ratio of the control valve?
- Morton (1996) recommended that the best location of a humidifier in an AHU or PU is in the first section of supply main duct downstream from the supply fan. The next choice is between the coil section and the supply fan if there is no oversaturation in the airstream.
- *Equipment noise*. For in-space humidifiers such as pneumatic humidifiers, equipment noise is a primary concern.
- *Initial cost.* For a steam humidifier, the initial cost should also include the installation costs of the steam boiler and the corresponding steam piping system.
- *Maintenance.* This includes the amount and cost of required maintenance work for the humidifiers.
- *Space occupied*. This is the volume occupied by the humidifier per unit humidifying capacity.

TABLE 15.4 Comparison of Various Types of Humidifiers

Description	Steam	Heating element	Air washer	Ultrasonic	Pneumatic atomizing
Humidifying capacity	Small to large	Small	Large	Small to large	Medium and large
Oversaturation	No	No	Yes	Possible	Possible
Quality of humidification	Good	Good	Good	Good	Good
Dripping of water in conditioned space	No	No	No	No	No
Energy consumption and operation cost	Need more energy if cold air supply is needed, so operating cost is higher	Same as steam	Saving energy if cold air supply is needed in winter	Saving energy if cold air supply is needed in winter	Saving energy if cold air supply is needed in winter
Initial cost	High	High	Medium	Medium	High
Maintenance work and maintenance cost	Low	Medium	High	Medium	Medium
Capacity control	Modulation	On/off or modulation	Modulation	Multiple stages	Modulation
Response to control	Fast	Slow	Medium	Fast	Fast
Space occupied	Small	Medium	Large	Small	Medium
Equipment noise	Medium	Low	Medium	Low	High

Various types of humidifiers are compared in Table 15.4. For commercial buildings that need a warm air supply during winter and for which a steam supply is available, a steam humidifier is often the most suitable choice. For a small packaged unit serving a health care facility, a heating element humidifier may be selected. For a telecommunications center with an appreciable amount of space heat gain, an ultrasonic humidifier is often the right choice. Ultrasonic humidifiers are also often used in residences for their comparatively low equipment noise. For a factory in which space air contains lint and other dirt that should be cleaned, or for which a cold air supply is needed to offset the cooling load in winter, an air washer may be the suitable choice.

When air washers or atomizing humidifiers are selected as the humidifiers, oversaturation may result in the supply air discharged from the air washer or atomizing humidifier. As oversaturation causes wetted surfaces and dampness along the air distribution passage, the following remedies are recommended to prevent IAQ problems:

- All-metal (fiber-free) silencer or active silencer should be used, if required.
- Interior surfaces of ducts that contact supply air must be galvanized metal sheets.
- Duct heat gain may be beneficial to eliminate oversaturation.
- Periodic monitoring of the moisture and dirt conditions of the ducts and silencers should be provided.

ASHRAE/IESNA Standard 90.1-1999 mandates that when a zone is served by a system or systems with both humidification and dehumidification, means (such as limit switches, mechanical stops, software program for DDC control) shall be provided to prevent simultaneous operation of humidification and dehumidification equipment. Exceptions are zones served by desiccant dehumidification systems, which are used with direct evaporative cooling in series, or zones where specific humidity levels are required such as computer rooms, museums, and hospitals. Standard 90.1-1999 also mandates that humidifiers with preheating jackets mounted in the airstream shall have an automatic valve to shut off preheat operation when humidification is not in use.

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CHAPTER 16
AIR SYSTEMS: EQUIPMENT—AIR-HANDLING UNITS AND PACKAGED UNITS

16.1 AIR-HANDLING UNITS 16.1
Functions of Air-Handling Units 16.1
Classifications of Air-Handling Units 16.2
16.2 MAIN COMPONENTS 16.4
Casing 16.4
Fans 16.4
Coils 16.5
Filters 16.5
Humidifiers 16.5
Outdoor Air Intake, Mixing, and Exhaust Section 16.6
Controls 16.6
Component Layout 16.6
Coil Face Velocity 16.8
16.3 SELECTION OF AIR-HANDLING UNITS 16.9
16.4 PACKAGED UNITS 16.12
Types of Packaged Unit 16.12
Rooftop Packaged Units 16.12
Indoor Packaged Units 16.15
Split Packaged Units 16.16
16.5 PERFORMANCE AND SELECTION OF PACKAGED UNITS 16.17
Indoor Environmental Control 16.17
Indoor Air Quality 16.18
Scroll Compressors and Evaporative Condensers 16.18
Controls 16.18
Minimum Performance 16.19
Selection of Packaged Units 16.19
16.6 FAN ROOM 16.24
Types of Fan Room 16.24
Layout Considerations 16.25
REFERENCES 16.28

16.1 AIR-HANDLING UNITS

Functions of Air-Handling Units

An air-handling unit (AHU) is the primary equipment in an air system of a central hydronic system; it handles and conditions the air and distributes it to various conditioned spaces. In an AHU, the required amounts of outdoor air and recirculating air are often mixed and conditioned. The temperature of the discharge air is then maintained within predetermined limits by means of control systems. After that, the conditioned supply air is provided with motive force and is distributed to various conditioned spaces through ductwork and space diffusion devices.

Many air-handling units are modular so that they have the flexibility to add components as required. An AHU basically consists of an outdoor air intake and mixing box section, a fan section including a supply fan and a fan motor, a coil section with a water cooling coil, a filter section, and a control section. A return or relief fan, a heating coil, a precooling coil, and a humidifier may also be included depending on the application. Supply volume flow rates of AHUs vary from 2000 to 63,000 cfm (945 to 29,730 L/s).

Whether a return fan or a relief fan should be added to an air system depends on the construction and operating characteristics of the air system and the total pressure loss of the return system (see Chap. 22). A heating coil is mainly used in the air-handling unit that serves the perimeter zone, or for morning warm-up in the heating season. The use of a precooling coil, to draw

cooling water from the cooling tower as a water economizer is discussed in Chap. 21. Humidifiers are employed for processing air conditioning and health care facilites where space humidity must be controlled.

Classifications of Air-Handling Units

Air-handling units may be classified according to their structure, location, and conditioning characteristics.

Horizontal or Vertical Unit. In a horizontal unit, the supply fan, coils, and filters are all installed at the same level, as shown in Fig. 16.1*a*. Horizontal units need more floor space for installation, and they are mainly used as large AHUs. Most horizontal units are installed inside the fan room. Occasionally, small horizontal units may be hung from the ceiling inside the ceiling plenum. In such a circumstance, fan noise and vibration must be carefully controlled if the unit is adjacent to the conditioned space.

In a vertical unit, the supply fan is not installed at the same level as the coils and filters but is often at a higher level, as shown in Fig. 16.1*b*. Vertical units require less floor space. They are usually smaller, so that the height of the coil section plus the fan section, and the height of the ductwork that crosses over the AHU under the ceiling, is less than the head room (the height from the floor to the ceiling or the beam of the fan room). The fan room is the room used to house AHUs and other mechanical equipment.

Draw-Through Unit or Blow-Through Unit. In a draw-through unit, the supply fan is located downstream from the cooling coil section, and the air is drawn through the coil section, as shown in Fig. 16.1*a* and *b*. In a draw-through unit, conditioned air is evenly distributed over the entire surface of the coil section. Also the discharge air from the AHU can be easily connected to a supply duct of similar higher velocity. Draw-through units are the most widely used AHUs.

In a blow-through unit, the supply fan is located upstream from the coil section, and the air blows through the coil section, as shown in Fig. 16.1*c*. Usually, a multizone air-handling unit adopts a blow-through unit. In a multizone AHU, the coil section is divided into the hot deck and the cold deck. The heating coil is installed in the hot deck just above the cold deck, where the cooling coil is located. The hot deck is connected to ductwork that supplies warm air to the perimeter zone through the warm duct. The cold deck is connected to a cold duct that supplies cold air to both the perimeter and interior zones.

A blow-through unit also has the advantage of treating the supply fan heat gain as part of the coil load and thus reduces the supply system heat gain.

Outdoor Air (or Makeup Air) AHU or Mixing AHU. Most mixing AHUs can be used to condition either outdoor air only or a mixture of outdoor air and recirculating air, whereas an outdoor air AHU is used only to condition 100 percent outdoor air, as shown in Fig. 16.1*d*. An outdoor air, or makeup air, AHU is a once-through unit; there is no return air and mixing box. It may be a constant-volume system or a variable-air-volume (VAV) system if the number of occupants varies. In an outdoor-air AHU, the cooling coil is usually a six- to eight-row depth coil because of the greater enthalpy difference during cooling and dehumidification in summer. Freeze protections for water coils are necessary in locations where the outdoor temperature may be below 32°F (0°C) in winter. A heat recovery coil or a water economizer precooling coil is often installed in makeup air AHUs for energy savings.

Single-Zone AHU or Multizone AHU. A single-zone AHU serves only a single zone. A multizone AHU serves two or more zones, as shown in Fig. 16.1*c*. A zone can be a large perimeter or an interior zone or one of the many control zones which connect to a multizone AHU through ducts and terminals.

A multizone AHU with a hot and cold deck is now often used for a dual-duct VAV system (see Chap. 21). Another kind of multizone unit which has many separate warm air ducts and cold air

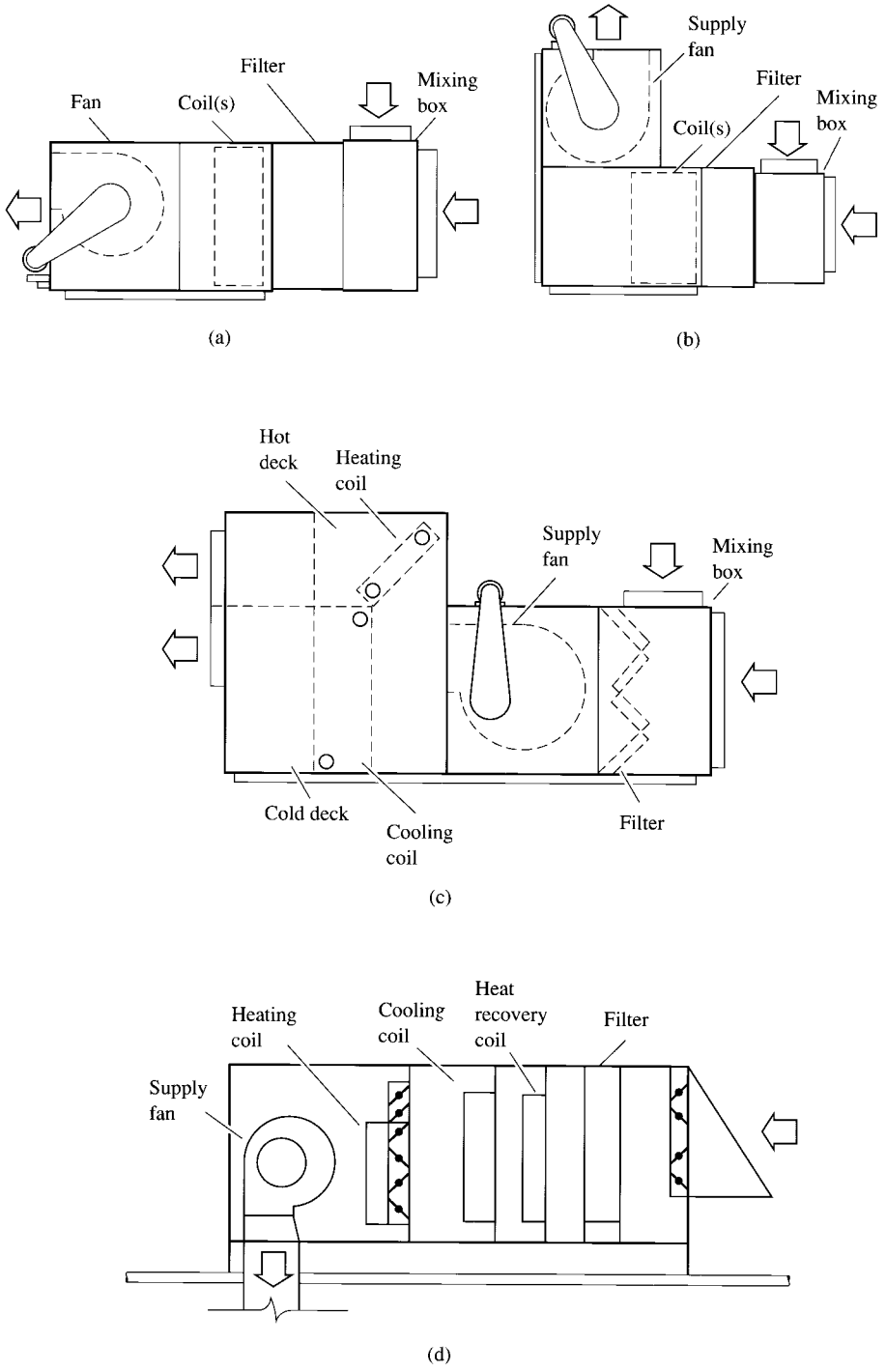


FIGURE 16.1 Type of air-handling units (AHUs): (*a*) horizontal, draw-through unit; (*b*) vertical draw-through unit; (*c*) blow-through unit, multizone AHU; (*d*) makeup air AHU, custom-built, rooftop unit.

ducts with associated warm and cold air dampers for each of the zones became obsolete because this kind of multizone unit wastes energy, needs complicated control, and is expensive.

Factory-Fabricated AHU or Field-Built AHU, Custom-Built or Standard Fabrication. One important reason to use factory-fabricated AHUs or standard fabrications is their lower cost and higher quality. Factory labor and controlled manufacturing techniques provide more efficient and better-quality construction than field labor and assembly.

Custom-built and field-built AHUs provide more flexibility in structure, system component arrangements, dimensions, and specialized functions than standard fabricated products. Custom-built and field-built AHUs also need more comprehensive, detailed specifications. Standard fabricating products are usually less expensive and can be delivered in a shorter time.

Rooftop AHU or Indoor AHU. A rooftop AHU is an outdoor penthouse, as shown in Fig. 16.1*d*. It is usually curb-mounted on the roof and should be completely weathertight. The outside casing is usually made of heavy-gauge galvanized steel or aluminum sheets with corrosion-resistant coating and sealant at the joints, both inside and outside. The fan motor, water valves, damper actuator linkages, and controls are all installed inside the casing. Access doors are necessary for service and maintenance of fans, coils, and filters. An indoor AHU is usually located in the fan room. Small AHUs are sometimes ceiling-hung.

16.2 MAIN COMPONENTS

Because of the impact of the indoor air quality (IAQ), the design and the construction of the AHU have been affected in many ways, as discussed in Gill (1996).

Casing

Two kinds of casings are more widely used for new AHUs today: (1) a double-wall sheet-metal casing in which the insulation material is sandwiched between two sheet-metal panels of 1- to 2-in. (25- to 50-mm) thickness with a U value from 0.12 to 0.25 $Btu/h \cdot ft^2 \cdot °F$ (0.68 to 1.42 $W/m^2 \cdot °C$) and (2) single sheet-metal panel with inner insulation layer and perforated metal liners.

Although insulating materials such as glass fibers and mineral wool are inert, when they become wet and collect dirt, both the glass fiber and the glass fiber liner provide the site and source of microbial growth. In addition, glass fiber liner is susceptible to deterioration and erosion over time. With a double-wall sheet-metal casing, glass fibers are not exposed to the moisture of the ambient air. Its inner surface can also be cleaned easily. Perforated metal liners cannot isolate the isulating material from the ambient moisture, but they are helpful to attenuate the fan noise.

The outside surface of the casing is often coated with an ultraviolet-resistant epoxy paint. The interior surface is better coated with a light color paint which increases the ability to spot the debris and microbial growth. Hinged access panels to the fan, coils, and filter sections must be provided for inspection and maintenance. Thermal break construction employs a resin bridge between the exterior and interior panels to interrupt the through-metal path heat transfer. A well-sealed double-wall metal panel should have an air leakage of only 1 cfm/ft^2 (5.078 $L/s \cdot m^2$) panel at a 4 in. WC (1000 Pa) of outer and inner static pressure difference.

Fans

A double-inlet airfoil, backward-inclined centrifugal fan is often used in large AHUs with greater cfm (L/s) and higher fan total pressure for its higher efficiency and lower noise. Vane-axial fans with carefully designed sound-absorptive housings, sound attenuators at inlet and outlet, and other

attenuations can now provide a sound rating of NC 55 in the fan room. Although the forward-curved centrifugal fan has a lower efficiency at full load, it is more compact and its part-load operating characteristics are better than those of a backward-inclined centrifugal fan. It is often used in small AHUs and where cfm (L/s) and fan total pressure are lower. For VAV systems, a dedicated outdoor air injection fan is sometimes used to provide outdoor ventilation air according to demand at both full and part load. An axial relief fan or an unhoused centrifugal return fan may be added as an optional system component. A return fan is used when the total pressure loss of the return system is considerable. This is discussed in Chap. 22. Large fans are usually belt-driven. Only small fans are sometimes direct-driven.

As mentioned in Sec. 15.4, an adjustable-frequency variable-speed drive saves more energy than inlet vanes for VAV systems during part-load operation. It is often cost-effective for large centrifugal fans although a variable-speed drive is expensive. Inlet vanes are not suitable for small airfoil or backward-inclined centrifugal fans because they block the air passage at the fan inlet.

Generally, a centrifugal fan has a higher efficiency and at the same time a lower noise. Given two fans of the same model, both with the same cfm (L/s) and fan total pressure, a centrifugal fan that is greater in size and slower in speed creates less noise.

Coils

In AHUs, the following types of coil are often used: water cooling coils, water heating coils, electric heating coils, and water precooling coils. The construction and characteristics of these coils are discussed in Chap. 15. Electric heating coils are made with nickel-chromium wire as the heating element (see Sec. 8.4). Ceramic bushes float the heating elements, and vertical brackets prevent the elements from sagging. In a finned tubular element sheathed construction, the electric heating coil is usually made with a spiral fin brazed to a steel sheath. An electric heating coil is usually divided into several stages for capacity control.

When an electric heating coil is installed inside an AHU, the manufacturer should have the assembly tested by the Underwriters' Laboratory (UL) to ensure that its requirements are met. Otherwise, the heater may only be installed outside the AHU as a duct heater, with a minimum distance of 4 ft (1.2 m) from the AHU. Various safety cutoffs and controls must be provided according to the National Electrical Codes and other related codes.

Cooling and heating coils need periodic cleaning and freeze-up protection in locations where the outdoor air temperature may drop below 32°F (0°C) in winter. Condensate pan and condensate drain line must be properly designed and installed. All these are discussed in Chap. 15.

Filters

Air filtration is an important component to achieve an acceptable indoor air quality. In AHUs, earlier low-efficiency filters of the panel type are giving way to the medium- and high-efficiency bag type and cartridge type of filters. Carbon-activated gaseous absorption filters are also used to remove objectionable odors or volatile organic compounds (VOCs) in buildings. Newly developed air filters are more efficient at removing air contaminants of particle size between 0.3 and 5 μm which are lung-damaging dust.

Humidifiers

Usually, there is no humidifier installed in the AHU for comfort air conditioning systems; but the outdoor climate is very cold in winter so that if a humidifier is not employed, the winter indoor relative humidity may be too low. Humidifiers are necessary for health care facilities and processing systems in pharmaceutical, semiconductor, textile, communication centers, and computer rooms.

Steam grid or electric heating element humidifiers are widely used in AHUs where a warm air

supply and humidity control are needed in winter. Ultrasonic humidifiers are often used for buildings in which a cold air supply and humidity control are required. For industrial applications such as textile mills where humidity control, air washing, and cold air supply are needed all year round, an air washer is often used for these purposes.

Outdoor Air Intake, Mixing, and Exhaust Section

An outdoor air intake, mixing, and exhaust section includes an outdoor air intake, an exhaust outlet, dampers, a mixing box, and a return fan or a relief fan. The location of outdoor air intake has a direct impact on space IAQ.

- The outdoor air intake for each AHU should install with wind shield and louvers to prevent rain and birds. If the AHU is located on the roof, the bottom of the outdoor intake louvers should be at least 3 ft above the the roof.
- The outdoor air intake must be located as far away from the exhaust outlets and plumbing vent stacks (horizontally and vertically) as possible, to prevent the intake of exhaust contaminants, condensation, and freezing which may provide a means of growth of microorganisms. Codes and local regulations should be followed.
- The outdoor air intake should reflect the influence of the prevailing winds.
- An outdoor intake system should be provided with air filters or even air cleaners in locations where outdoor air contaminants exceed the National Primary Air Quality Standard, as discussed in Sec. 4.10.

For better outdoor ventilation air control, an outdoor damper should split into two dampers: a minimum outdoor ventilation damper and an economizer damper of 100 percent outdoor air free cooling except in small AHUs. Both should provide a short ducted outside path for air balancing and install with airflow measuring station, or a minimum outdoor ventilation air injection fan and control if necessary.

Poor mixing, such as parallel outdoor and recirculating airstreams in the mixing box, causes stratification of the mixture, as discussed in Sec. 15.11. This is one of the important causes of coil freezing in locations where the outdoor air temperature drops below 32°F (0°C). For good mixing, airstreams should meet at a 90° angle or opposite each other, as shown in Fig. 16.2.

Recently, there is a trend to use an unhoused plug/plenum fan as a return fan in many AHUs and PUs. Usually, a return fan is located nearly in the center of an exhaust compartment, as shown in Fig. 16.3. The exhaust and recirculating dampers form two sides of the exhaust compartment. A plug/plenum fan has the advantage of discharging from both exhaust and recirculating dampers and is quieter. After the recirculating damper, the mixture enters the coil section.

Controls

As discussed in Sec. 5.14, AHU controls include generic, specific, safety, and diagnostic controls. The economizer control, discharge air temperature control, duct static pressure control, outdoor ventilation air control, humidity, and warm-up or cool-down controls are discussed in Chap. 21, 22, and 23.

Component Layout

In a typical horizontal, drawn-through AHU, the layout of the components in serial order is usually as follows:

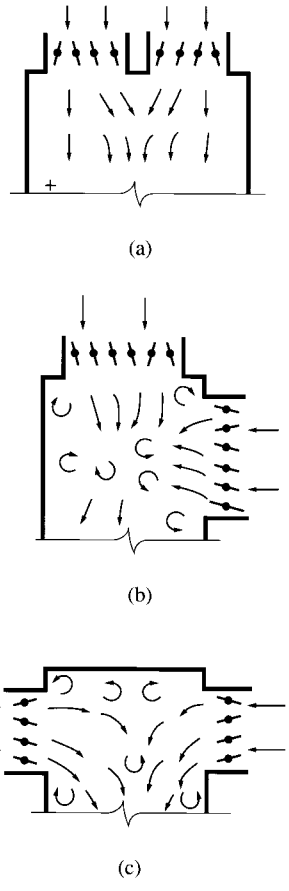


FIGURE 16.2 Mixing of outdoor and recirculating airstreams: (*a*) parallel airstreams (poor mixing); (*b*) airstreams at 90° (good mixing); (*c*) opposite airstreams (good mixing).

1. Return or relief fan, exhaust air passage and damper (optional)
2. Mixing box with outdoor air and recirculating air dampers
3. Filters: (prefilter, optional) medium-efficiency filters
4. Preheating coil (optional)
5. Precooling coil (optional)
6. Cooling coil
7. Heating coil (optional)
8. Supply fan
9. Humidifier (optional)
10. High- or ultrahigh-efficiency filters (optional)

If there is a relief fan, it should be located on one side of the exhausting compartment or in the relief or exhaust passage. If there is an unhoused return fan, it should be located nearly in the center

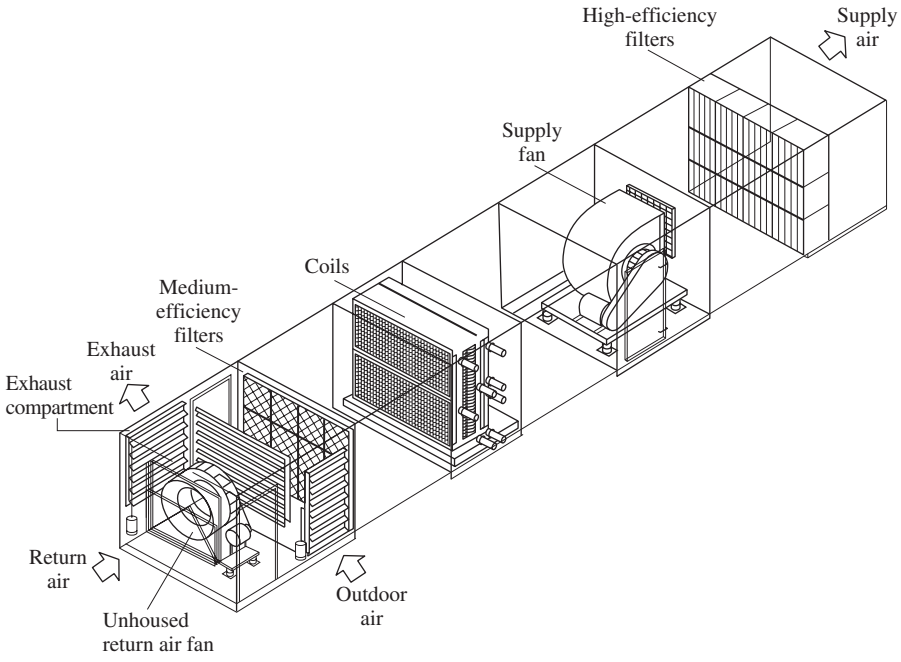


FIGURE 16.3 A typical AHU with unhoused plug/plenum return fan.

of the exhaust compartment. The volume flow and fan total pressure of the return fan should be carefully determined so that the pressure inside the exhaust compartment is positive; and the pressure inside the mixing box must be negative in order to extract outdoor air, except an outdoor air projected fan is installed in the outdoor air passage (see Chaps. 21 and 23).

In an AHU, to condition a mixture of outdoor and recirculating air is often simpler and less expensive than to condition the outdoor air and recirculating air separately. Therefore, the mixing box is usually located before the filters and the coils. A preheating coil is always located before the water heating and cooling coils for the sake of freeze-up protection. A precooling coil is always located before a cooling coil for a greater temperature difference between the air and water. A steam grid humidifier is usually located after a heating coil because humidification is more effective at a higher air temperature.

If there are ultrahigh-efficiency filters, they should be located as near to the clean room or clean space as possible to prevent pollution from the ductwork or elsewhere. Low- or medium-efficiency prefilters must be installed ahead of the coils.

Coil Face Velocity

The size, or more accurately, the width and height of a horizontal AHU, is mainly determined by the face velocity of the coil. A higher coil face velocity results in a smaller coil, a higher heat-transfer coefficient, a greater pressure drop across the coil and filter, and a smaller fan room, which directly affects the space required. On the contrary, a lower coil face velocity has a larger coil, a lower heat-transfer coefficient, and a smaller pressure drop and fan energy use.

The maximum face velocity is usually determined according to the value required to prevent carryover of water droplets due to condensate from a cooling coil. As mentioned in Secs. 10.2 and

15.10, the cooling coil face velocity should not exceed 500 fpm (2.5 m/s) for a smooth fin coil and 550 fpm (2.75 m/s) for corrugated fins. The lower limit of the coil face velocity depends mainly on the initial and energy cost analysis, including cost of the AHU, fan room size, the total number of annual operating hours, and the unit rate of electric power. A lower limit of face velocities between 400 and 450 fpm (2 and 2.25 m/s) at design conditions may be considered appropriate under many specific circumstances.

For a fan room of adequate head room, it is preferable to use a high coil to reduce the face velocity and the pressure drop of coils and filters. Use of a high coil has little influence on the floor area of the fan room except that when a coil is higher than 42 in. (1070 mm), a cooling coil should split into two coils vertically and use two separate condensate pans vertically to prevent condensate carryover.

16.3 SELECTION OF AIR-HANDLING UNITS

Table 16.1 lists general data of the supply fan and coil of a typical horizontal draw-through AHU; fan B means a class II fan. In Table 16.2, the volume flow–fan static pressure performance of this horizontal draw-through modular AHU (unit size 30) with inlet vanes is presented. A backward-inclined centrifugal fan of $22\frac{1}{4}$-in. (565-mm) diameter is used. In Table 16.2, the following items are also listed:

- Supply volume flow rate $\dot{V}_s$, in cfm of standard air.
- Air velocity at fan outlet, fpm.
- Fan static pressure, in in. WC. Fan total pressure can be obtained by adding the fan velocity pressure to the fan static pressure; velocity pressure p_v, in in. WC, can be calculated by $p_v = (v_{out}/4005)^2$ here v_{out} indicates the fan outlet velocity, in fpm.
- Revolutions per minute (rpm) of fan impeller.
- Brake horsepower (bhp) input to fan shaft.

The following are recommendations for selection of AHUs from a manufacturer's catalog:

- Draw-through AHUs are widely used. For a small AHU, a vertical unit saves floor space if the headroom of the fan room is sufficient except the AHU is ceiling-hung. For a large AHU, a horizontal unit is often the right choice.
- The size of the AHU is selected so that the face velocity of the cooling coil v_{coil} is optimum. For corrugated fins, maximum v_{coil} should not exceed 550 fpm (2.75 m/s).
- For large AHUs, choose a backward-curved centrifugal fan with airfoil blades or a backward-inclined fan for higher efficiency. Select the rpm of the supply fan or supply and return fans in order to meet the required system total pressure loss, i.e., external total pressure plus the total pressure loss of the AHU.
- For VAV systems, an adjustable-frequency variable-speed drive should be compared with inlet vanes via life-cycle cost analysis. Energy consumption also should be analyzed at part-load operation. For a large AHU, an adjustable-frequency variable-speed drive may be cost-effective. When an AHU is equipped with a small airfoil or backward-inclined centrifugal fan, inlet vanes for capacity control are not recommended because of the extremely high air velocity at the fan inlet.
- The required coil load is met through the variation of the number of rows of coil and the fin spacing. An even number of rows are often used so that the inlet and the outlet of the coils are on the same side. Four-row coils are often used for a mixing AHU, and a makeup AHU seldom uses a coil that exceeds eight rows.
- In locations where outdoor air temperatures go below 32°F (0°C), coil freeze-up protection attained by installing a preheating coil and the improvement of the mixing of airstreams in the mixing box should be considered.

TABLE 16.1 General Data on Supply Fans and Coils of a Typical Horizontal Draw-through AHU

Description			Unit size number									
			3	6	8	10	12	14	17	21	25	30
							Fan data					
Fan	Size		—	—	—	$13\frac{1}{2}$	15	$16\frac{1}{2}$	$18\frac{1}{4}$	20	$22\frac{1}{4}$	$22\frac{1}{4}$
BI	Shaft size (in.)		—	—	—	$1\frac{11}{16}$	$1\frac{15}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{7}{16}$	$2\frac{11}{16}$	$2\frac{11}{16}$
	Outlet area (ft^2)		—	1.41	1.90	2.31	2.79	3.39	4.14	5.05	6.30	6.30
	Max. rpm/static pressure (in. WC)		—	—	—	3536/8	3183/8	2894/8	2616/8	2387/8	2164/8	2164/8
	Motor hp range		—	—	—	1–10	1–15	1–15	1–20	1–25	1–25	1–30
							Coil data					
	Unit coils ($\frac{1}{2}$-in. tube)											
Cooling		Ft^2	3.32	5.86	7.54	9.64	12.3	14.2	16.8	20.8	24.4	29.0
		Dimensions (in.)	21 × 23	23 × 36	27 × 40	27 × 51	32 × 55	35 × 59	37 × 65	45 × 67	51 × 69	51 × 82
Heating		Ft^2	2.34	4.31	5.49	7.01	9.46	10.2	12.3	15.0	17.8	21.2
		Dimensions (in.)	15 × 23	17 × 36	20 × 40	20 × 51	25 × 55	25 × 59	27 × 65	32 × 67	37 × 69	37 × 82
	$\frac{5}{8}$-in. or 1-in. tube coils											
		Ft^2	2.75	4.38	6.50	8.33	11.3	12.1	14.7	16.5	19.8	23.6
		Dimensions (in.)	18 × 22	18 × 35	24 × 39	24 × 50	30 × 54	30 × 58	33 × 64	2−18 × 66*	1−18 × 68* 1−24 × 68	1−18 × 81* 1−24 × 81

BI: Backward-inclined centrifugal fan
*Two coils are used for this unit.
***Source*:** The Trane Company. Reprinted with permission.

TABLE 16.2 Volume Flow and Fan Static Pressure of a Typical Horizontal Draw-through Modular AHU (Size 30) with Inlet Vanes (Backward-Inclined Centrifugal Fan of 22.25-in. Diameter)

CFM std. air	Outlet velocity	Fan static pressure, in. WC													
		3.50		3.75		4.00		4.25		4.50		4.75		5.00	
		rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
10,000	1,587	1,524	10.39	1,561	10.91	1,597	11.40	1,632	11.87	1,667	12.32	1,701	12.77	1,734	13.23
11,000	1,746	1,570	11.81	1,607	12.54	1,643	13.23	1,678	13.86	1,712	14.44	1,745	14.99	1,777	15.51
12,000	1,905	1,620	13.21	1,656	14.00	1,691	14.81	1,725	15.62	1,759	16.41	1,792	17.16	1,824	17.85
13,000	2,063	1,678	14.90	1,710	15.67	1,742	16.46	1,774	17.28	1,806	18.14	1,838	19.02	1,870	19.90
14,000	2,222	1,741	16.82	1,771	17.61	1,800	18.41	1,830	19.23	1,860	20.08	1,889	20.94	1,919	21.84
15,000	2,381	1,809	18.98	1,836	19.80	1,864	20.63	1,891	21.47	1,919	22.32	1,946	23.20	1,974	24.09
16,000	2,540	1,880	21.36	1,906	22.23	1,931	23.10	1,957	23.97	1,982	24.85	2,008	25.75	2,034	26.65
17,000	2,698	1,953	23.97	1,978	24.88	2,002	25.80	2,026	26.72	2,050	27.64	2,074	28.57	2,098	29.71
18,000	2,857	2,028	26.83	2,052	27.78	2,075	28.74	2,098	29.71						
19,000	3,016	2,104	29.96												
CFM std. air	Outlet velocity	5.25		5.50		5.75		6.00		6.25		6.50			
		rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp		
10,000	1,587	1,766	13.70	1,798	14.19	1,830	14.71	1,861	15.26	1,892	15.82	1,923	16.41		
11,000	1,746	1,809	16.01	1,840	16.51	1,871	17.00	1,901	17.51	1,931	18.02	1,960	18.55		
12,000	1,905	1,854	18.48	1,885	19.09	1,915	19.68	1,944	20.24	1,973	20.79	2,002	21.33		
13,000	2,063	1,901	20.76	1,932	21.58	1,962	22.35	1,991	23.07	2,020	23.76	2,048	24.41		
14,000	2,222	1,949	22.77	1,979	23.71	2,008	24.66	2,037	25.60	2,066	26.50	2,094	27.36		
15,000	2,381	2,002	25.00	2,030	25.94	2,057	26.90	2,085	27.9	2,113	28.90	2,140	29.92		
16,000	2,540	2,060	27.57	2,086	28.51	2,112	29.46								

Outlet vel–outlet velocity, fpm
rpm–revolutions/minute
bhp–brake horsepower
Source: The Trane Company. Reprinted with permission.

- To improve indoor air quality, medium- and high-efficiency filters should be used to provide an acceptable IAQ and to protect coils and air distribution devices. Dirty coils and condensate pans significantly degrade the IAQ. A prefilter must be installed to extend the service life of high- and ultrahigh-efficiency filters, or gas absorbers.
- Use an air or water economizer to save energy. Use indirect and direct evaporative coolers to replace part of the refrigeration if they are applicable and economical. Evaporative cooling is discussed in Chap. 27.
- Select an AHU with adequate specific and safety controls and diagnostics.

16.4 PACKAGED UNITS

Types of Packaged Unit

A packaged unit (PU) is a unitary, self-contained air conditioner. It is also the primary equipment of a unitary packaged system. A packaged unit not only conditions the air and provides the motive force to supply the conditioned air to the space, but also provides gas heating, or electric heating, and refrigeration cooling from its own gas-fired furnace and refrigerating equipment or from its own heat pump. It is actually the primary equipment in an air conditioning system.

A packaged unit is always equipped with DX coil(s) for cooling. This characteristic is the primary difference between a packaged unit and an air-handling unit. The portion that handles conditioned air in a packaged unit is called an airhandler, the air system of the packaged system, to distinguish it from an air-handling unit, as shown in Fig. 16.4. Another portion is a condensing unit, the refrigerant plant. Refrigerants HCFC-22, HFC-134a, HFC-404A, HFC-410A, HFC-407A, and HFC-407C are now used in packaged units. Most PUs are factory-built standard fabrication units.

A packaged unit can be either enclosed in a single package or split into two units: an indoor air handler and an outdoor condensing unit. A packaged unit can also be a packaged heat pump; most are air-source heat pumps. In a packaged heat pump, in addition to the fan, DX coil, filters, compressors, condensers, expansion valves and controls, there are four-way reversing valves to reverse the refrigerant flow when cooling mode operation is changed to heating mode operation.

The construction and size of a packaged unit depend mainly on its model and cooling capacity, in refrigeration tons or Btu/h (W). Packaged units can be classified according to their place of installation as rooftop packaged units, indoor packaged units, and split packaged units. Among these units, the rooftop packaged units are most widely used in commercial buildings.

Rooftop Packaged Units

A rooftop packaged unit is mounted on the roof of the conditioned space, as shown in Fig. 16.4. It is usually enclosed in a weatherproof outer casing. The mixture of outdoor air and recirculating air is often conditioned in the rooftop packaged unit and supplied to the conditioned space on the floors below. Based on the types of heating and cooling sources, rooftop packaged units can be subdivided into the following:

- *Gas/electric rooftop packaged unit*. In this unit, heating is provided by a gas-fired furnace, and cooling is provided by electric-driven reciprocating or scroll compressors.
- *Electric/electric rooftop packaged unit.* In this unit, heating is provided by electric heating coils and cooling by reciprocating or scroll compressors.
- *Rooftop packaged heat pump*. In this unit, heating and cooling are provided by the heat pump, with auxiliary electric heating if necessary.

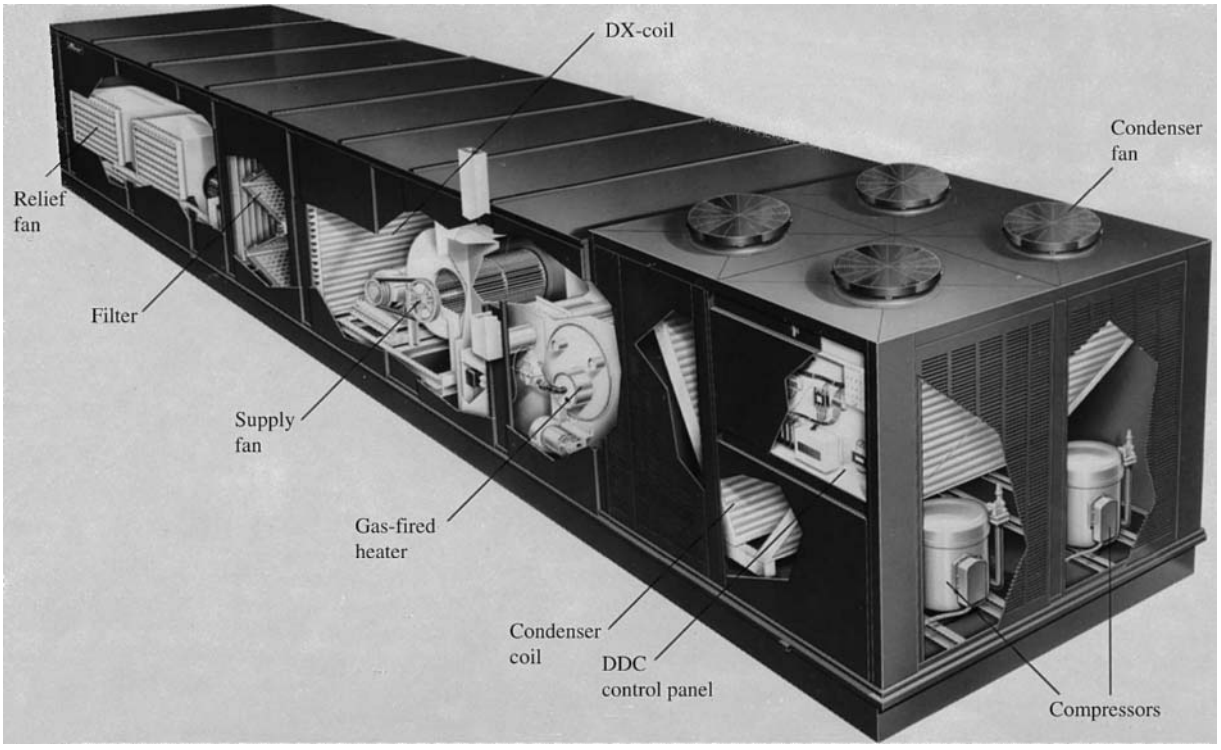


FIGURE 16.4 Cutaway view of a typical rooftop packaged unit. (*Source: the Trane Company. Reprinted with permission.*)

A rooftop packaged unit is a single packaged unit composed of two main components: an air handler and a condensing unit. Its cooling capacity may vary from 3 to 220 tons (10 to 774 kW), and its supply volume flow rate may vary from 1200 to 80,000 cfm (565 to 37,750 L/s). An air handler of a typical rooftop packaged unit consists of mainly a casing, an indoor fan, DX coils, filters, a mixing box, and controls; a gas-fired heater, a relief or return fan, and a humidifier are optional. The construction and characteristics of the casing, fans, filters, and outdoor air intake and mixing box are similar to those discussed in AHUs.

Curb. A rooftop packaged unit is mounted on a curb which is a perimeter frame supporting the unit. A curb is often made of galvanized or aluminized steel sheets or angle iron and sits on a deck. It usually has additional structural support at or beneath the deck. On the top of the curb, there is always a mating flange that matches the size of the rooftop unit with a sealing gasket to provide a weatherproof joint. Curbs are either factory-prefabricated or field-assembled.

Curbs should be tall enough above the structural support for a sloped roof. The manufacturer and the structural engineer should be consulted to ensure that the unit can resist the local anticipated wind pressure. The requirements of National Roofing Contractors Association (NRCA) should be followed.

DX Coils. For a specific model and size of rooftop packaged unit, the coil surface area is a fixed value. DX coils are usually of two, three, and four rows (except makeup units) with a fin spacing of 12 to 17 fins/in. (1.5- to 2.1-mm fin spacing). For large units, two separate refrigerant circuits with their own coils, associated expansion valves, and distributors are often used for a better capacity control.

The cooling capacity of a rooftop packaged unit should include the additional solar heat absorbed by the unit on the rooftop. Because a DX coil is a wet coil, the same as the water cooling coil, condensate drain pans, condensate traps, and the condensate drain line should be properly designed and installed. Since refrigerant is used as the coolant instead of chilled water, coil freeze-up protection is no longer necessary.

Supply, Return, and Relief or Exhaust Fans. For a rooftop packaged unit with a cooling capacity of 10 tons (35 kW) or less, there is often only a supply (indoor) fan. For a rooftop packaged unit of cooling capacity of 15 to 30 tons (53 to 105 kW), there is often a supply fan and a relief (exhaust) fan. For a rooftop packaged unit of cooling capacity of 60 tons (210 kW) and greater, some manufacturers offer a supply fan and a return fan. As in AHUs, an unhoused plug/plenum return fan is also often located nearly in the center of an exhaust compartment in a rooftop packaged unit.

Supply and return fans in rooftop packaged units are usually belt-driven. For each refrigeration ton (3.5 kW) of cooling capacity, a rooftop packaged unit usually has a nominal supply volume flow rate of 350 to 450 cfm/ton (47 to 60 L/s·kW). However, a rooftop packaged unit can vary its speed of supply fan and return fan to provide various volume flow rates and fan total pressure for a specific model and size. External pressure is the pressure loss of the duct system and terminals. Most of the rooftop packaged units can vary their supply volume flow rate at a range between 200 and 500 cfm/ton (27 and 67 L/s·kW). A maximum of 6-in. (1500-Pa) fan total static pressure or 4-in. (1000-Pa) external pressure can be provided by a supply fan in rooftop packaged units of 30 tons (105 kW) and greater.

There are also manufacturers that offer inlet vanes, inlet cone, or adjustable frequency variable-speed drive for rooftop packaged units of 20 tons (70 kW) and greater to modulate volume flow rates of supply and return fans at part-load operation in variable-air-volume systems. For variable-speed drives, cooling air must be supplied to the elecronic drive mechanism to prevent encountered high temperature because of the solar heat on the rooftop.

Gas-Fired Furnace and Electric Heating Coil. The gas burners in a rooftop packaged unit of a heating capacity of 40,000 Btu/h (11,720 W) and greater are power burners of the induced combustion type. A centrifugal blower is used to extract combustion air and combustion products to a vent. There are often two tubular heat exchangers: a 16-gauge (1.6-mm-thick) stainless-steel or aluminum-silicon alloy coated steel primary heat exchanger, and an 18-gauge (1.3-mm-thick) stainless-steel or corrosion-resistant alloy coated steel secondary exchanger. Both are of free-floating design to eliminate expansion and contraction stresses and noises.

Firing rate is often controlled by modulating the gas valve through a DDC controller. Safety controls for gas-fired furnace include the proving of combustion air supply prior to ignition and continuous electronic flame supervision.

An electric heater is often made of internally wired heavy-duty nickel-chromium elements. It is usually divided into multiple stages or units for capacity control.

Humidifiers. A humidifier is optional. Indoor packaged units for computer room and data processing systems are often installed with steam or heating element humidifiers in a position between the coil section and the supply fan inlet.

As in the AHU, the outdoor intake of a rooftop unit should be shielded from the wind effect, located as far away as possible from the contaminated air source, and the bottom of the louvers should be kept at least 3 ft (0.9 m) from the roof.

In a rooftop packaged unit, the condensing unit mainly consists of compressors; an air-cooled, evaporatively cooled, and water-cooled condenser; and controls.

Compressors. Semihermetic and hermetic reciprocating compressors and scroll compressors are often used. Scroll compressors are more energy-efficient than reciprocating compressors. They need fewer parts and are quieter. Scroll compressors are gradually replacing reciprocating compressors in packaged units. For medium-size and large rooftop packaged units, two, three, or four

compressors of equal or sometimes unequal horsepower input are preferable for better capacity control in steps.

Condensers. Multirow, $^3/_8$-in. (10-mm) copper tube and aluminum fin air-cooled condensing coil connected with subcooling coils are used. Condensing coil may cover the two sides of the condenser or be in a V shape at the middle. The ratio of the face area of the condensing and subcooling coils to the cooling capacity is often 1.5 to 2 ft^2/ton_{ref} (0.04 to 0.053 m^2/kW_{ref}). Propeller fans are used for the induced-draft condenser fan. The ratio of volume flow of cooling air to the cooling capacity is between 600 and 900 cfm/ton_{ref} (81 and 121 $L/s \cdot kW_{ref}$). Air-cooled condensers are most widely used in rooftop packaged units today for their lower initial cost and simple operation and maintenance.

An evaporative condenser has the advantage of low condensing temperature, similar to a water-cooled condenser with cooling towers, as well as lower energy consumption but at a lower initial cost. Today, an evaporatively cooled condenser has become one of the optional alternatives in a rooftop packaged unit and is used more and more frequently than before.

Heat Pump. A rooftop packaged heat pump is a packaged unit installed with four-way reversing valves to change the refrigerant flow after the compressor. In an air-source heat pump, the DX coil is often called an indoor coil, and the air-cooled condensing coil the outdoor coil. During cooling mode operation, hot gas from the compressor is first discharged to the outdoor coil for condensing and subcooling. The liquid refrigerant then enters the expansion valve and indoor coil to produce refrigeration. During the heating mode operation, the reversing valve changes its connections and the direction of refrigerant flow. Hot gas from the compressor now enters the indoor coil to release its condensing heat first. The liquid refrigerant is then discharged to the outdoor coil to absorb heat from the ambient air for evaporation. Detailed operation and system performance are discussed in Sec. 12. 2.

Indoor Packaged Units

An indoor packaged unit is also a single packaged, factory-built unit. It is usually installed indoors inside a fan room or a machinery room, as described in Sec. 9.21. A small or medium-size indoor packaged unit may sometimes be floor-mounted directly inside the conditioned space with or without connected ductwork, such as the indoor packaged unit in computer rooms, as shown in Fig. 16.5. The cooling capacity of the indoor packaged unit may vary from 3 to 100 tons (10 to 350 kW), and its supply volume flow rate from 1200 to 40,000 cfm (565 to 18,880 L/s).

Indoor packaged units can be classified as follows:

- *Indoor packaged cooling units*. Only cooling is provided by the DX coil.
- *Indoor packaged heating/cooling units.*These units not only provide cooling from the DX coil, but also provide heating from a hot water coil, steam heating coil, or electric heater.
- *Indoor packaged heat pump.* When an indoor packaged unit is connected to an air-cooled condenser and equipped with reversing valves, the change of refrigerant flow also causes the change of cooling mode and heating mode operation and provides heating and cooling as required.

In indoor packaged units, usually only a supply fan is installed in small units. Because of the compact size of the units, generally a forward-curved centrifugal fan is used. For large indoor packaged units, an additional return fan section is added to extract recirculating air from the conditioned space through the return duct.

For computer room indoor packaged units, return air entering the unit at high level and supply air discharged at low level are common. Medium-efficiency filters or sometimes high-efficiency filters are usually used. A steam humidifier or other type of humidifier is always an integral part of computer room units, so as to maintain a required space relative humidity in winter and prevent static electricity. Multiple semihermetic and hermetic reciprocating compressors or scroll compres-

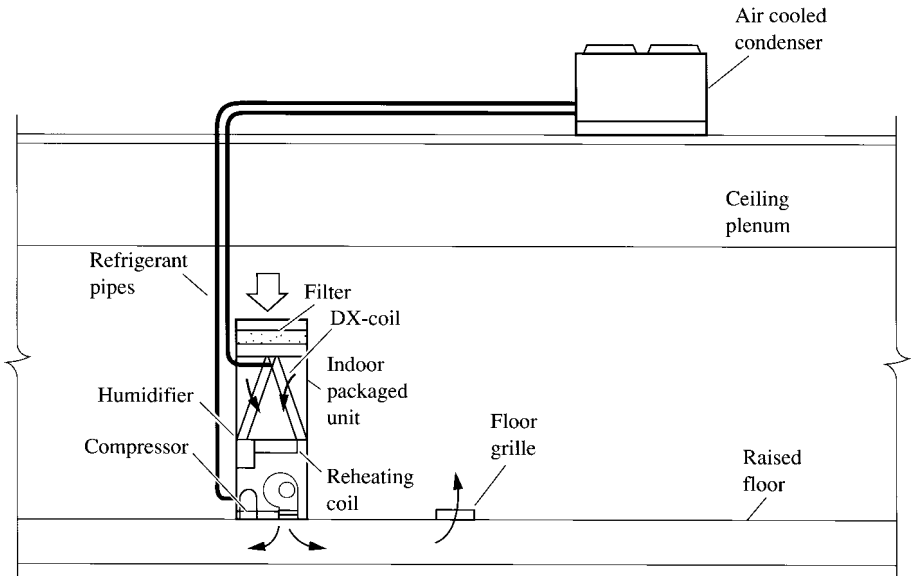


FIGURE 16.5 A typical indoor packaged unit.

sors—with dual refrigerant circuits—are used in medium-size and large units for capacity control. A microprocessor-based DDC control is always used for new and retrofit indoor packaged units.

An indoor packaged unit differs from a rooftop packaged unit in condensing arrangements. Usually there are two alternatives in indoor packaged units:

- With an air-cooled condenser, hot gas from the compressor is discharged to an air-cooled condenser through the discharge line located on the rooftop. Liquid refrigerant is returned to the DX coil from the air-cooled condenser through the liquid line.
- A shell-and-tube or a double-tube water-cooled condenser is installed inside the unit, and the condenser water is supplied from the cooling tower or from other sources.

An economical analysis based on the local conditions should be made to determine whether an air-cooled or a water-cooled condenser should be installed. If a water-cooled condenser using condenser water from the cooling tower is used, a precooling coil is sometimes installed in the indoor packaged unit as a component of the water-side economizer.

Split Packaged Units

A split packaged unit, sometimes called a split system, splits the packaged unit into an indoor air handler and an outdoor condensing unit, which is most probably mounted outdoors, on the rooftop, on a podium, or in some other adjacent place, as shown in Fig. 16.6. The indoor air handler and outdoor condensing unit are connected by refrigerant pipes.

An air handler in a split packaged unit is similar to the air handler in a rooftop unit except that a large air handler in a split packaged unit is usually installed inside the fan rooms, whereas the small air handler may be hung under the ceiling. An air handler for split packaged units usually has a cooling capacity from 3 to 80 tons (10 to 280 kW), a supply volume flow rate of 1200 to 32,000 cfm (565 to 15,100 L/s), and a maximum fan total pressure of 5.0 in. WC (1250 Pa) for medium-size and large units.

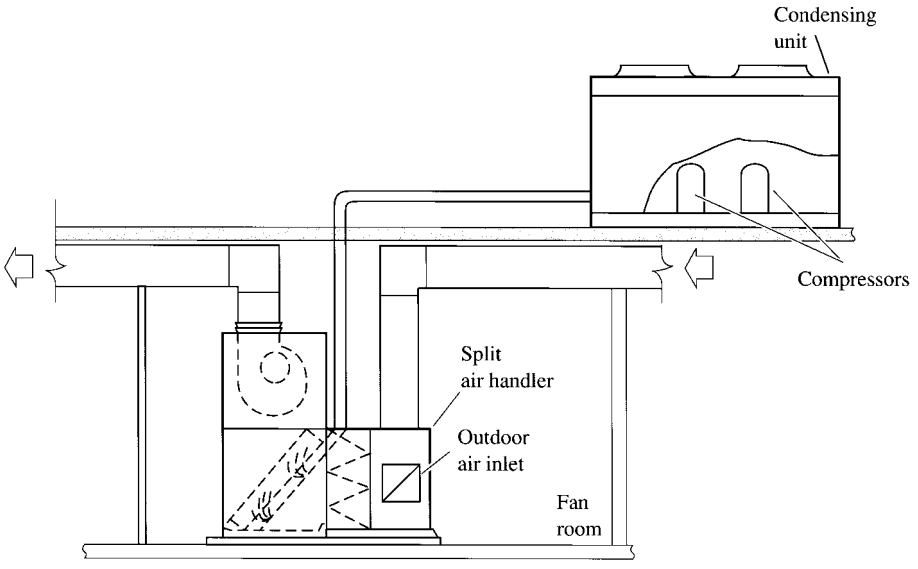


FIGURE 16.6 A typical split packaged unit.

Reciprocating and scroll compressors are used in split packaged units. The condenser in the outdoor condensing unit can be either air-cooled or water-cooled. Compared with indoor packaged units, a split packaged unit always has its compressors in its outdoor condensing unit, whereas an indoor packaged unit has its compressor, water-cooled condenser indoors.

A split packaged heat pump is also a kind of split packaged unit. In such a unit, additional reversing valves or changeover arrangements force the refrigerant flow from the compressor to the outdoor air-cooled condensing coil during the cooling mode operation, and change to the indoor DX coil during the heating mode operation. The cooling capacity of a split packaged heat pump varies from 10 to 30 tons (35 to 105 kW), and the heating capacity ranges from 100,000 to 400,000 Btu/h (29,300 to 117,200 W) at rated conditions.

16.5 PERFORMANCE AND SELECTION OF PACKAGED UNITS

A packaged unit is the primary equipment of a packaged air conditioning system. Its function and performance represent the quality and capability of the packaged system. Because of the development of microprocessor-based DDC control systems in the 1980s, and the effort toward energy conservation since the energy crisis in 1973, the difference in indoor environmental control and indoor air quality between a custom-built central system and a packaged system becomes smaller in the late 1990s, according to the following analyses.

Indoor Environmental Control

Today, there are so many types, models, and sizes of packaged units, such as rooftop, indoor, and split units with gas-fired, electric, hot water, and steam heating at a volume flow from 1200 to 48,000 cfm (565 to 22,650 L/s) and a fan total pressure from 2.0 to 6.0 in. WC (500 to 1500 Pa)

to meet various requirments of HVAC&R systems. Factory-fabricated and -assembled packaged units usually have a higher quality than the field-assembled central system of nearly the same construction.

Many manufacturers offer variable-air-volume packaged units of cooling capacity of 20 tons (70 kW_{ref}) and greater. A VAV packaged unit maintains a required zone temperature by varying the volume flow rate of supply air using proportional-integral control mode through each terminal. Such a control is a modulation control, offset-free, fast response, and energy-efficient.

Some packaged units provide humidifiers to control the space relative humidity as required. Most of the packaged units offer simultaneously operated air economizer and refrigeration compressor at a cooling capacity of 5 tons (18 kW) and greater.

Indoor Air Quality

Adequate outdoor ventilation air, higher-efficiency air filters, and a clean and effective air system directly affect the indoor air quality. A packaged unit is as effective at providing adequate outdoor ventilation air for the occupants in the conditioned space as an AHU in a central system. With the increase of the fan total pressure (external pressure), most medium-size and large packaged units can now be installed with high-efficiency filters.

Scroll Compressors and Evaporative Condensers

Many packaged units now use scroll compressors instead of reciprocating compressors. In 1997, one scroll compressor manufacturer in the United States announced that the energy efficiency ratio (EER) of its product was 11.5 Btu/h·W (3.37 COP). A rooftop packaged unit using an evaporative condenser can lower the condensing temperature, approaching a value that only a water-cooled condenser can achieve, and therefore is energy-efficient.

Controls

Microprocessor-based specific controls, safety controls, and diagnostics for 1997 manufactured rooftop packaged units with a cooling capacity between 20 to 130 tons (70 to 455 kW_{ref}) are as follows:

1. Discharge air temperature control
   - Cooling: air economizer control
   - Cooling: staging of scroll compressors
   - Heating: modulation of gas valve—continuous flame supervision
   - Reset based on outdoor or zone temperature
2. Duct static pressure control
   - Inlet vanes/variable-speed drive control
   - Maximum duct static pressure control
3. Space pressure control
4. Morning warm-up—full or cycling capacity
5. Minimum ventilation control
   - Space positive pressurization, exhaust mode, and purge mode
6. DX coil frost protection
7. Occupied/Unoccupied switching—night setback
8. Low ambient compressor lockout

9. Compressor lead/lag control—a more balanced run time among compressors
10. Humam interface—to monitor all temperatures, pressures, humidities, inputs and outputs; to edit set points; and to select services
11. Diagnostics—to detect faults and diagnostics (a total of 49 different diagnostics can be read from the display)
12. Emergency stop input

A satisfactory indoor environment surely will be maintained, and at the same time the rooftop packaged unit is energy-efficient with all these controls.

Minimum Performance

To reduce energy use in the packaged units, ASHRAE/IESNA Standard 90.1-1999 mandates minimum efficiency requirements of various PUs as listed in Table 11.6, and also specifies the minimum efficiency requirements for various air source heat pumps as listed in Sec. 12.2. Almost all PUs and air source heat pumps are driven and operated electrically.

In Table 11.6, EER indicates the energy efficiency ratio, SEER the seasonal energy efficiency ratio, IPLV the integrated part-load value, and HSPF the heating seasonal performance factor. All of these energy index are defined in Sec. 9.20.

When using the efficiency ratings listed in Table 11.6 and Sec. 12.2 to compare different types of HVAC&R equipment, the following conditions must be considered:

- The efficiency ratings for water-cooled equipment cannot be compared directly to those for air-cooled equipment. Water-cooled equipment does not include the energy use of required condenser water pumps and cooling tower fans whereas air-cooled packaged unit ratings include the energy use of condenser fan.
- The ratings for condensing units cannot be directly compared with single or split packaged units. Condensing unit ratings do not include the energy use of fans in indoor air handlers.
- The efficiency ratings of a water chiller cannot be compared with a packaged unit using DX coil, as the efficiency of a water chiller does not include the energy use of chilled-water pumps.

Selection of Packaged Units

The procedure of selection of PUs is different from that for an AHU because the number of rows and fin spacing of a DX coil are fixed for a specific model and size of PU. The size of a PU is determined primarily by the required cooling capacity of the DX coil at various operating conditions. The cooling capacity of a typical rooftop PU is listed in Table 16.3, and Table 16.4 presents the supply fan performance of this typical rooftop PU. The selection procedure for PUs, based on data from manufacturers' catalogs, can be outlined as follows:

1. Calculate the cooling coil load and sensible cooling coil load (refer to Chap. 6) of the conditioned space that is served by the PU. For a unitary packaged system using DX coils, coil load is equal to the refrigeration load of the refrigeration system.

Select the model and size of the PU based on the cooling capacity that is equal to or greater than the required coil load and sensible coil load at the specified design conditions. These include the dry- and wet-bulb temperatures of air entering the DX coil and the outdoor air entering the air-cooled condenser or evaporatively cooled condenser.

The equipment capacity may exceed the design load only when the equipment selected is the smallest size needed to meet the load.

TABLE 16.3 Cooling Capacity of a Typical Rooftop PU, Nominal Capacity 55 Tons; Volume Flow Rate, Standard Air, of 19,250 cfm

	Ambient temperature, °F											
	85						95					
	Entering wet-bulb temperature, °F											
	61		67		73		61		67		73	
Entering dry-bulb, °F	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR
75	502	81	559	55	621	33	473	83	528	56	587	33
80	506	95	559	69	620	46	479	97	528	71	586	47
85	530	100	559	83	620	59	505	100	528	85	586	60
90	558	100	564	95	619	71	533	100	536	96	585	73
	105°F						115°F					
	61°F		67°F		73°F		61°F		67°F		73°F	
Entering dry-bulb, °F	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR	MBH	SHR
75	443	86	496	57	552	33	413	89	463	58	516	33
80	453	98	495	73	551	47	427	100	462	75	516	48
85	480	100	496	88	551	61	454	100	464	91	515	63
90	508	100	507	98	550	75	481	100	481	100	515	78

MBH: 1000 Btu/h
SHR: Sensible heat ratio
Source: The Trane Company. Reprinted with permission.

TABLE 16.4 Supply Fan Performance of a Typical Rooftop PU, Nominal Capacity 55 Tons

CFM standard air	Total static pressure, in. WC															
	2.250		2.500		2.750		3.000		3.250		3.500		3.750		4.000	
	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
10,000	842	8.09	887	9.18	927	10.28	965	11.37	1,000	12.47	1,034	13.59	1,067	14.74	1,099	15.90
12,000	843	9.17	887	10.31	931	11.50	972	12.75	1,012	14.03	1,050	15.34	1,085	16.65	1,118	17.96
14,000	857	10.72	897	11.85	936	13.01	974	14.25	1,012	15.54	1,050	16.90	1,087	18.31	1,123	19.76
16,000	876	12.59	914	13.81	952	15.05	988	16.31	1,023	17.60	1,057	18.93	1,091	20.30	1,124	21.74
18,000	900	14.80	936	16.08	972	17.40	1,006	18.76	1,040	20.14	1,073	21.54	1,105	22.96	1,137	24.40
20,000	926	17.31	962	18.75	996	20.14	1,029	21.56	1,061	23.01	1,093	24.50	1,124	26.01	1,154	27.54
22,000	946	19.85	986	21.57	1,022	23.22	1,055	24.79	1,086	26.33	1,116	27.88	1,146	29.46		
22,500	950	20.50	991	22.28	1,027	23.97	1,061	25.64	1,093	27.24	1,123	28.84				
24,000	963	22.44	1,004	24.40												

rpm: revolutions per minute
bhp: brake horsepower
Source: The Trane Company. Reprinted with permission.

If the required supply air volume flow rate deviates from the nominal rated value in percentage, as shown below, then the cooling capacity and sensible cooling capacity can be roughly multiplied by a multiplier because of the change of the heat-transfer coefficient as follows:

Volume flow	− 20%	− 10%	0%	+10%	+20%
Cooling capacity	0.965	0.985	1.0	1.015	1.025
Sensible cooling	0.94	0.97	1.0	1.03	1.06

2. Calculate the heating coil load at the winter design condition or at the warm-up period. Determine the capacity of the gas-fired furnace, electric heater, water or steam heating coil. For a packaged heat pump, calculate the supplementary heating capacity of the gas-fired or electric heater.

3. Evaluate the external total pressure loss of the duct system, terminal, and space diffusion devices (see Chaps. 17 and 18). Determine the speed of the supply fan, relief fan, exhaust fan, or return fan, such that the volume flow and the fan total pressure of the supply fan, supply fan/return fan combination is equal to or greater than the sum of the external total pressure loss and the total pressure loss in the PU.

For fans for which only a fan static pressure is given, roughly a 0.4-in. WC (100-Pa) of velocity pressure can be added to the fan static pressure for purposes of rough estimation of fan total pressure.

Check that the face velocity of the DX coil does not exceed 550 fpm (2.75 m/s) so that condensate will not carry over.

4. For small PUs, a medium-efficiency air filter of lower pressure drop, such as a final pressure drop between 0.4 and 0.6 in. WC (100 to 150 Pa), should be selected.

Example 16.1. Select an AHU or a rooftop PU for a typical floor in a commercial building with the following operating characteristics:

Supply volume flow rate	16,000 cfm (7550 L/s)
Cooling coil load	520,000 Btu/h or 43 tons (152,360 W)
Sensible cooling coil load	364,000 Btu/h (106,650 W)
Outdoor air temperature	95°F (35°C)
Entering coil dry-bulb temperature	80°F (26.7°C)
Entering coil wet-bulb temperature	67°F (19.4°C)
External pressure drop	2.0 in. WC (500 Pa)
Total pressure loss of AHU or PU	2.25 in. WC (563 Pa)

For a 4-row, 12-fins/in. water-colling coil in an AHU, chilled water enters the coil at 45°F (7.2°C) and is expected to leave the coil at 55°F (12.7°C).

Solution:

1. Divide the supply volume flow rate by 500 fpm, that is, 16,000/500 = 32. From Tables 16.1 and 16.2, select an AHU of size 30, which gives a maximum static pressure up to 8 in. WG and a face area of cooling coil of 29 ft^2. The actual face velocity of cooling coil is

$$v_{coil} = \frac{16{,}000}{29} = 551 \text{ fpm } (2.75 \text{ m/s})$$

There will be no carryover of condensate water droplets; therefore, it is acceptable.

2. From Table 16.2, select a size-30 horizontal draw-through unit with a backward-inclined centrifugal fan at an impeller diameter of $22\frac{1}{4}$ in. and a fan speed of 1931 rpm. The fan static pressure is then 4.00 in. WC at a volume flow of 16,000 cfm. From Table 16.2, because the fan outlet velocity is 2540 fpm, its velocity pressure is then

$$p_v = \left(\frac{2540}{4005}\right)^2 = 0.40 \text{ in. WC}$$

And the fan total pressure provided by this fan is

$$4.00 + 0.40 = 4.40 \text{ in. WC (1100 Pa)}$$

This value is greater than the required fan total pressure

$$2.0 + 2.25 = 4.25 \text{ in. WC (1063 Pa)}$$

From the manufacturer's catalog, the heights of the fan and coil modules are both 4 ft 6 in. (1.37 m). For such a height, the headroom of the fan room is usually sufficient.

3. As in Example 15.2, the cooling and dehumidifying capacity per ft^2 of coil face area of the water cooling coil given by the manufacturer's catalog is 19.1 MBtu/h·ft^2, and the sensible cooling capacity is 13.89 MBtu/h·ft^2. For a coil face area of 29 ft^2, the total cooling and dehumidifying capacity is

$$Q_c = 19{,}100 \times 29 = 553{,}900 \text{ Btu/h (162,290 W)}$$

and the sensible cooling capacity is

$$Q_{cs} = 13{,}890 \times 29 = 402{,}810 \text{ Btu/h (118,020 W)}$$

These capacities both are greater than the required cooling capacity of 520,000 Btu/h (152,360 W) and the sensible cooling capacity of 364,000 Btu/h (106,650 W).

4. From Table 16.3, select a rooftop PU with a cooling capacity of 55 tons and a sensible heat ratio $SHR_c = 0.71$. When the supply volume flow rate of this PU is 19,250 cfm, at an air entering coil dry-bulb temperature of 80°F and a wet-bulb temperature of 67°F, and an outdoor air temperature of 95°F, the cooling capacity is 528 MBtu/h, and the sensible cooling capacity is $0.71 \times 528 = 375$ MBtu/h. Both are greater than the required values. The selected 55-ton PU is suitable. From the manufacturer's catalog, the face area of this 55-ton DX coil is 37.9 ft^2.

If the supply volume flow rate is reduced to 17,625 cfm $(19{,}250 - 17{,}625)/16{,}000 = 0.10$, or 10 percent greater than the required volume flow rate of 16,000 cfm, then the cooling capacity is reduced to

$$528{,}000 \times 0.985 = 520{,}080 \text{ Btu/h (152,380 W)}$$

and the sensible cooling capacity will be reduced to

$$375{,}000 \times 0.97 = 363{,}750 \text{ Btu/h (106,580 W)}$$

Only the sensible cooling capacity is slightly less than the required value of 364,000 Btu/h.

5. Assume that the supply fan outlet velocity of the rooftop PU is 2000 fpm (10 m/s); that is, the velocity pressure at fan outlet is

$$p_v = \left(\frac{2000}{4005}\right)^2 = 0.25 \text{ in. WC}$$

From Table 16.4, we see that at a supply volume flow of 17,625 fpm and a fan speed of about 1135 rpm, this rooftop PU can provide a fan total pressure of

$$\Delta p_t = 4.0 + 0.25 = 4.25 \text{ in. WC (1060 Pa)}$$

This fan total pressure can meet the required value. Check the DX coil's face velocity

$$\frac{17{,}625}{37.9} = 465 \text{ fpm (2.32 m/s)}$$

Coil face velocity is less than 550 fpm (2.75 m/s); therefore, there is no condensate droplet carryover.

16.6 FAN ROOM

Types of Fan Room

A fan room is an enclosure in which an AHU, an air handler, an indoor PU, and other accessories and air-handling equipment are located. According to ASHRAE Standard 15-1994, for refrigerating systems of 100 hp (74.6 kW) or less, a fan room may contain refrigeration machinery if

- The fan room is a separated, tight construction with tight fitting doors
- Access by authorized personnel is controlled
- Detectors (refrigerant, oxygen, etc.) are located in refrigerant leaking areas

Refer to Standard 15-1994 for details.

In low-rise buildings of three stories and less, a fan room may be used to serve up to three floors. In high-rise buildings of four stories and more, a fan room may be used to serve one or more floors, usually up to 20 floors, depending on the characteristics of the air system, its initial cost, and its operating cost.

Fan rooms can be classified according to their pressure characteristics as open or isolated.

Open Fan Room. An open fan room is open to the filter end of the AHU or air handler, as shown in Fig. 16.7. The return ceiling plenum is directly connected to the fan room through an inner-lined return duct or a return duct with a sound attenuator. Outdoor air is often forced to the fan room by an outdoor air fan or a makeup AHU, or extracted to the fan room by the supply fan in the AHU. The fan room becomes the mixing box of the AHU or air handler. Its pressure is often lower than the static pressure in the return plenum and the outdoor atmosphere. Return air is then extracted to the fan room through the return duct. An exhaust fan is often installed on the external wall of the fan room to maintain this pressure difference.

The advantages of this kind of fan room are a positive outdoor air supply and less ductwork in the fan room. The disadvantages include the following:

- There may be infiltration of uncontrolled outdoor air if the fan room is not airtight.
- The fan room is entirely exposed to outdoor air.
- Sufficient sound attenuation must be provided in the transfer duct or in the air passage that transfers the return air from the return ceiling plenum to the fan room.

Isolated Fan Room. In this kind of fan room, shown in Fig. 16.8, the outdoor air, the return air, and the exhaust air are all isolated from the fan room air because of the ductwork. The static pressure in the fan room depends mainly on the air leakage from or to the AHU or air handler, and the

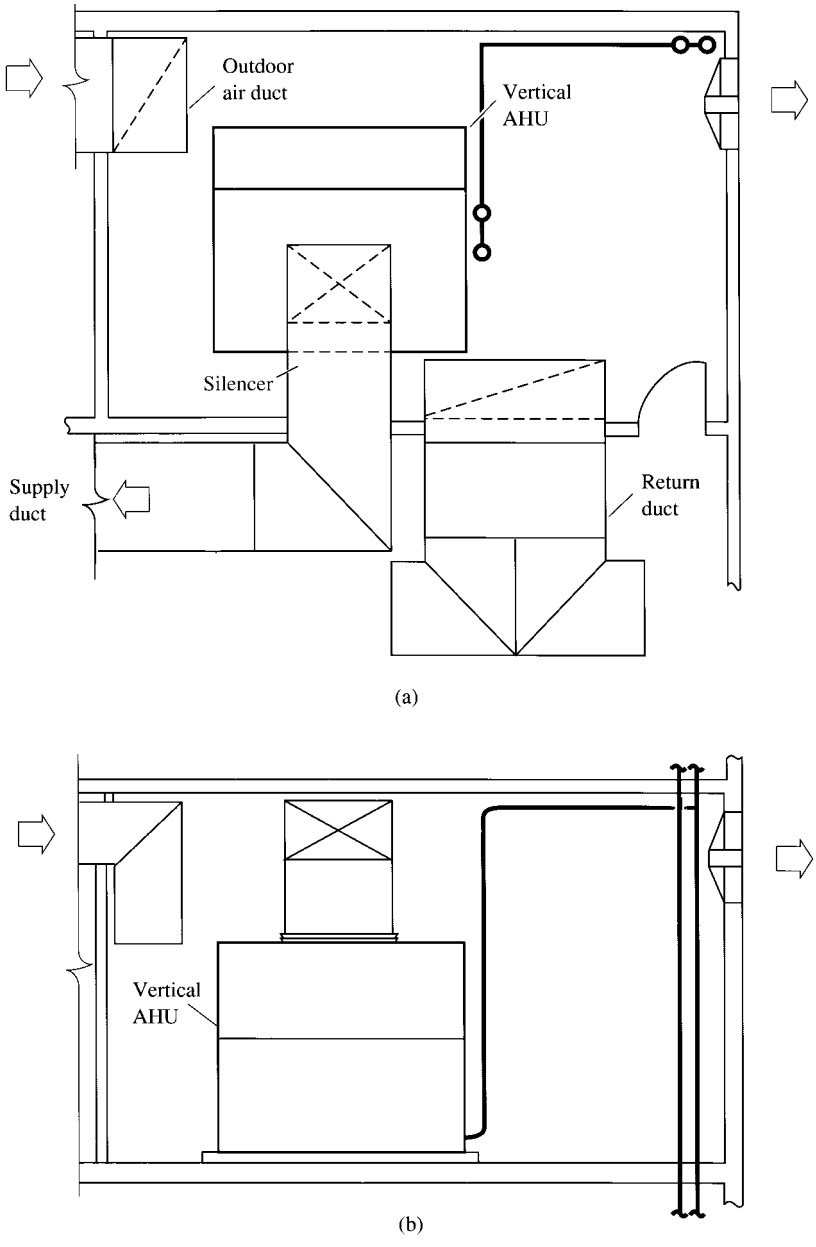


FIGURE 16.7 Open fan room: (*a*) plan view; (*b*) sectional view.

air passage connecting the fan room and outside atmophere. This kind of fan room is most widely used in commercial buildings.

Fan rooms located at the perimeter of the building often have direct access from the outside walls. They are more convenient to provide the outdoor air intake and exhaust. For fan rooms

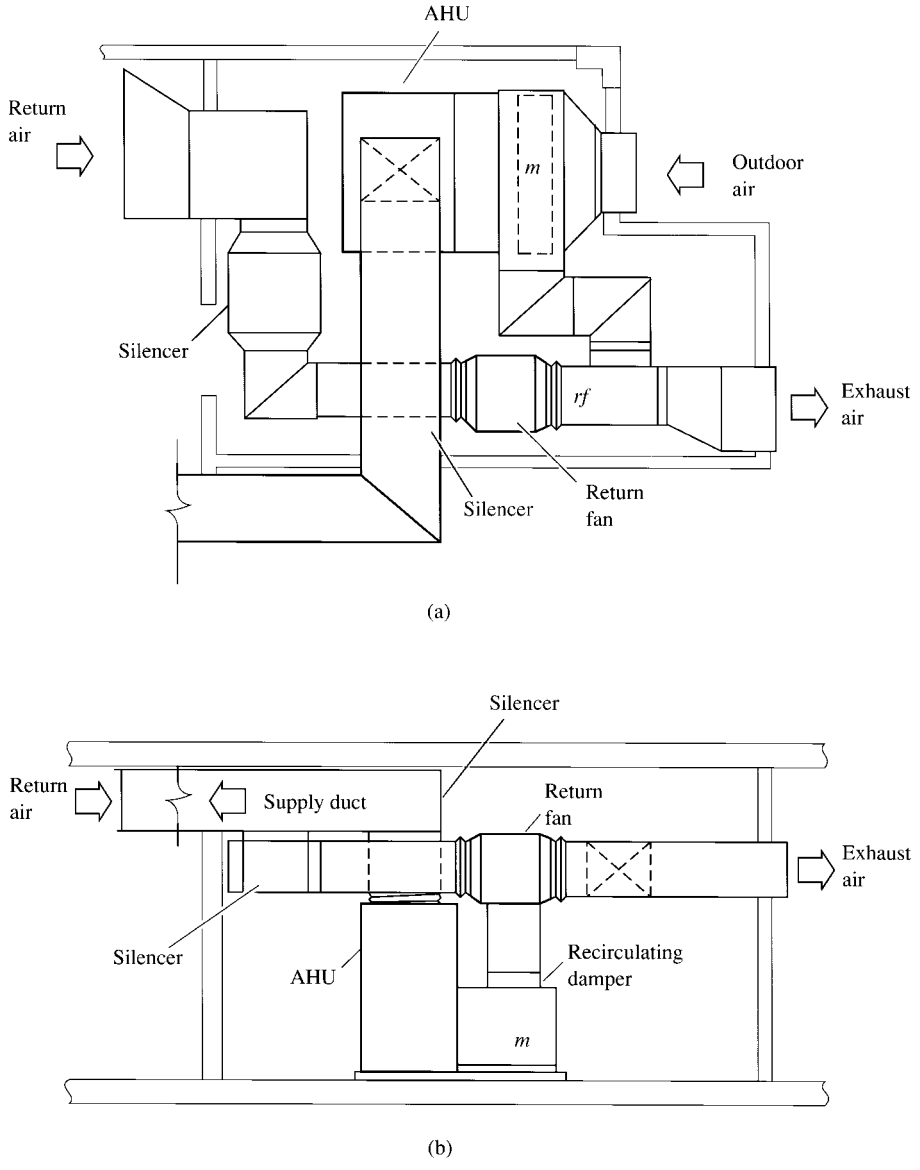
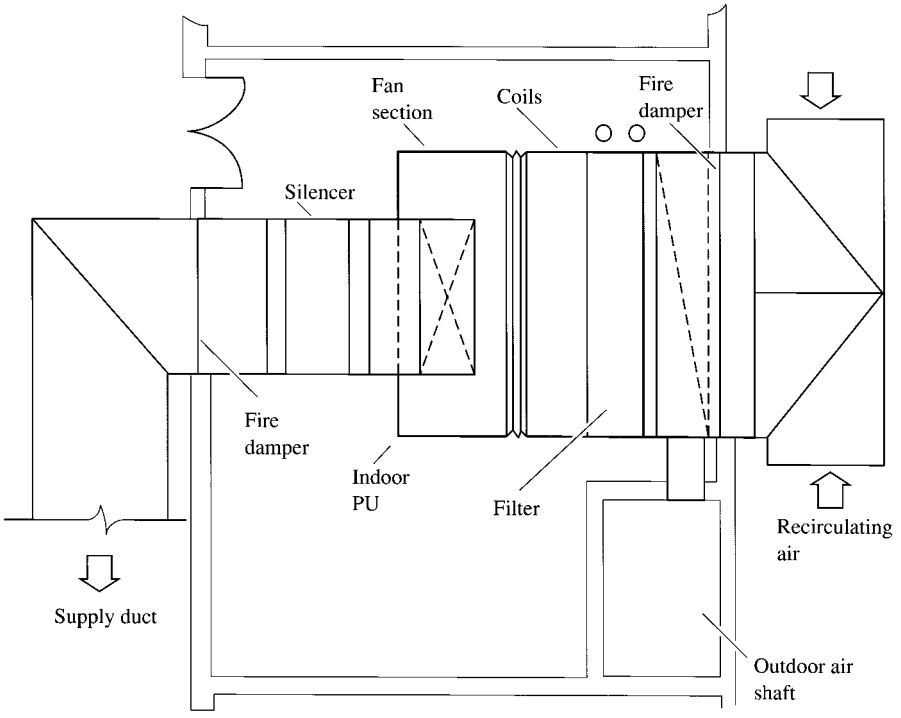


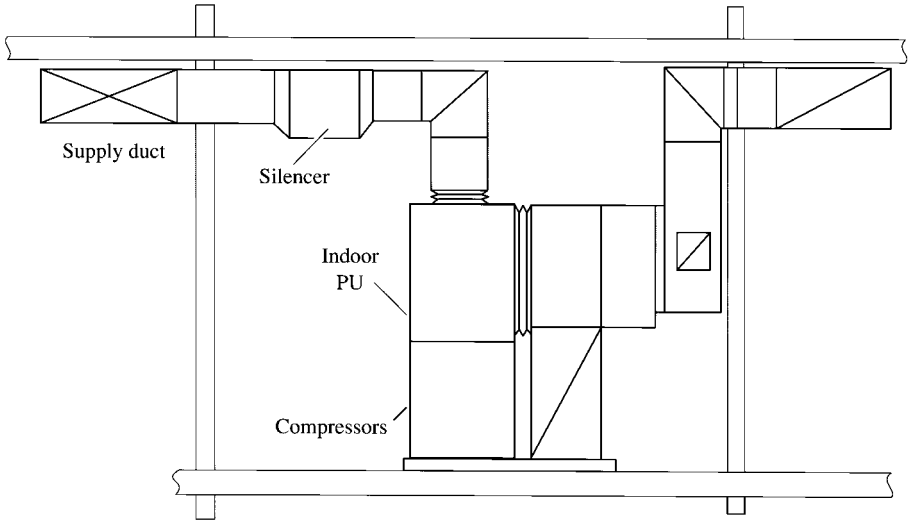
FIGURE 16.8 Isolated fan room: (*a*) plan view; (*b*) sectional view.

located in the interior core, large outdoor and exhaust air risers are required in multistory buildings when an air economizer cycle is used.

Figure 16.9 shows the plan and sectional views of an isolated fan room for an indoor packaged unit located in the interior core of the building. The unit is also equipped with a coil module for water cooling and heating coils and water economizer precooling coils.



(a)



(b)

FIGURE 16.9 Interior core fan room: (*a*) plan view; (*b*) sectional view.

Layout Considerations

A satisfactory fan room layout should meet the following requirements:

- It should be compact yet provide sufficient space for the maintenance workers to pull out fan shafts, coils, and filters. Piping connections on the same side of the AHU or PU are preferable.
- Outdoor intake and exhaust outlets are located either on outside walls perpendicular to each other or on different walls with a certain distance between them.
- Fire dampers should be installed to separate the fan room and the fire compartment according to the fire codes. Fan room ventilation and exhaust should be provided to meet the requirements of the codes.
- Inner-lined square elbows or elbows with 2- or 3-in. -(50- or 75-mm) thick duct liners are used for better sound attenuation at low frequencies. Sound attenuating devices are required for both the supply and return sides of the fan.
- A vertical AHU occupies less floor space than a horizontal unit; therefore, it is often the first choice if the headroom is sufficient.
- An unhoused centrifugal fan located in the exhaust compartment may be the suitable choice of return fan for its lower noise, and it has the advantage of discharging on both sides.

In Fig. 16.8*a*, at point *rf* after the return fan, the pressure is positive. At point *m* in the mixing box, the pressure must be negative in order to extract outdoor air. There must be a damper and an appropriate pressure drop between point *rf* and *m* to guarantee such a positive-to-negative pressure conversion.

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CHAPTER 17
AIR SYSTEMS: AIR DUCT DESIGN

17.1 BASICS OF AIRFLOW IN DUCTS 17.2
- Bernoulli Equation 17.2
- Steady Flow Energy Equation 17.2
- Static Pressure, Velocity Pressure, and Total Pressure 17.3
- Stack Effect 17.5
- Laminar Flow and Turbulent Flow 17.6
- Velocity Distribution 17.7
- Equation of Continuity 17.7

17.2 CHARACTERISTICS OF AIRFLOW IN DUCTS 17.8
- Types of Air Duct 17.8
- Pressure Characteristics of the Airflow 17.8
- System Pressure Loss 17.10
- Criteria of Fan Energy Use 17.10

17.3 DUCT CONSTRUCTION 17.12
- Maximum Pressure Difference 17.12
- Material 17.12
- Rectangular Ducts 17.13
- Round Ducts 17.17
- Flat Oval Ducts 17.17
- Flexible Ducts 17.18
- Fiberglass Ducts 17.18

17.4 DUCT HEAT GAIN, HEAT LOSS, AND DUCT INSULATION 17.19
- Temperature Rise or Drop due to Duct Heat Gain or Loss 17.19
- Duct Insulation 17.19
- Temperature Rise Curves 17.21

17.5 FRICTIONAL LOSSES 17.22
- Darcy-Weisbach Equation 17.22
- Friction Factor 17.22
- Duct Friction Chart 17.24
- Roughness and Temperature Corrections 17.25
- Circular Equivalents 17.27

17.6 DYNAMIC LOSSES 17.31
- Elbows 17.31
- Converging and Diverging Tees and Wyes 17.34
- Entrances, Exits, Enlargements, and Contractions 17.38

17.7 FLOW RESISTANCE 17.38
- Flow Resistances Connected in Series 17.40
- Flow Resistances Connected in Parallel 17.41
- Flow Resistance of a Y Connection 17.42
- Flow Resistance of a Duct System 17.42

17.8 PRINCIPLES AND CONSIDERATIONS IN AIR DUCT DESIGN 17.43
- Optimal Air Duct Design 17.43
- Design Velocity 17.45
- System Balancing 17.46
- Critical Path 17.48
- Air Leakage 17.48
- Shapes and Material of Air Ducts 17.50
- Ductwork Installation 17.50
- Fire Protection 17.50

17.9 AIR DUCT DESIGN PROCEDURE AND DUCT LAYOUT 17.51
- Design Procedure 17.51
- Duct System Characteristics 17.52
- Duct Layout 17.52

17.10 DUCT SIZING METHODS 17.53
- Equal-Friction Method 17.53
- Constant-Velocity Method 17.53
- Static Regain Method 17.54
- T Method 17.55

17.11 DUCT SYSTEMS WITH CERTAIN PRESSURE LOSSES IN BRANCH TAKEOFFS 17.56
- Design Characteristics 17.56
- Cost Optimization 17.56
- Condensing Two Duct Sections 17.59
- Local Loss Coefficients for Diverging Tees and Wyes 17.60
- Return or Exhaust Duct Systems 17.63

17.12 DUCT SYSTEMS WITH NEGLIGIBLE PRESSURE LOSS AT BRANCH DUCTS 17.66
- Supply Duct Systems 17.66
- Pressure Characteristics of Airflow in Supply Ducts 17.66
- Return or Exhaust Duct Systems 17.71

17.13 REQUIREMENTS OF EXHAUST DUCT SYSTEMS FOR A MINIMUM VELOCITY 17.72

17.14 COMPUTER-AIDED DUCT DESIGN AND DRAFTING 17.72
- Drafting 17.72
- Schedules and Layering 17.72
- Design Interface 17.73
- Running Processes 17.73

17.15 DUCT LINER AND DUCT CLEANING 17.74
- Duct Liner 17.74
- Duct Cleaning 17.74

17.16 PRESSURE AND AIRFLOW MEASUREMENTS 17.75
- Equal-Area versus Log Tchebycheff Rule 17.77

REFERENCES 17.78

17.1 BASICS OF AIRFLOW IN DUCTS

Bernoulli Equation

The Bernoulli equation relates the mean velocity v, in ft/s (m/s), the pressure p, in lb_f/ft^2 absolute (abs.) or psia (Pa abs.), and the elevation z, in ft (m), of a frictionless or ideal fluid at steady state. When a fluid motion is said to be in steady state, the variables of the fluid at any point along the fluid flow do not vary with time. Assuming constant density, the Bernoulli equation can be expressed in the following form:

$$\frac{p}{\rho} + \frac{v^2}{2g_c} + \frac{gz}{g_c} = \text{constant} \tag{17.1}$$

where p = static pressure, lb_f/ft^2 abs. (Pa abs.)
ρ = fluid density, lb_m/ft^3 (kg/m^3)
g = gravitational acceleration, ft/s^2 (m/s^2)
g_c = dimensional constant, 32.2 $lb_m \cdot ft/lb_f \cdot s^2$ (1)

For convenience, lb = lb_m (mass).

Steady Flow Energy Equation

For a real fluid flowing between two cross sections in an air duct, pipe, or conduit, energy loss is inevitable because of the viscosity of the fluid, the presence of the mechanical friction, and eddies. The energy used to overcome these losses is usually transformed to heat energy. If we ignore the kinetic energy difference between the value calculated by the mean velocity of the cross section and the value calculated according to the velocity distribution of the cross section, then the steady flow energy equation for a unit mass of real fluid is given as

$$\frac{p_1'}{\rho_1} + u_1 J + \frac{v_1^2}{2g_c} + \frac{gz_1}{g_c} + qJ = \frac{p_2'}{r_2} + u_2 J + \frac{v_2^2}{2g_c} + \frac{gz_2}{g_c} + W \tag{17.2}$$

where u = internal energy, Btu/lb (J/kg)
J = Joule's equivalent, 778 $ft \cdot lb_f/Btu$ (1)
q = heat supplied, Btu/lb (J/kg)
W = work developed, $ft \cdot lb_f/lb$ (J/s)

In Eq. (17.2), subscripts 1 and 2 indicate the cross section 1 and 2, respectively, and p_1' and p_2' denote the absolute static pressure at cross section 1 and 2. Signs of q and W follow the convention in thermodynamics, i.e., when heat is supplied to the system, q is positive and when heat is released from the system, q is negative. When work is developed by the system, W is positive; and for work input to the system, W is negative.

Multiply both sides of Eq. (17.2) by ρ, ignore the difference in densities, and rearrange the terms. Then each term has the unit of pressure, in lb_f/ft^2 abs. (Pa abs.), or

$$p_1' + \frac{\rho_1 v_1^2}{2g_c} + \frac{\rho_1 g z_1}{g_c} = p_2' + \frac{\rho_2 v_2^2}{2g_c} + \frac{\rho_2 g z_2}{g_c} + \rho W + \rho J(u_2 - u_1 - q) \tag{17.3}$$

For an air duct or piping work without a fan, compressor, and pump, $\rho W = 0$.

Let the pressure loss from viscosity, friction, and eddies between cross sections 1 and 2 be $\Delta p_f = \rho J(u_2 - u_1 - q)$; then each term of Eq. (17.3) can be expressed in the form of pressure

$$p_1' + \frac{\rho_1 v_1^2}{2g_c} + \frac{\rho_1 g z_1}{g_c} = p_2' + \frac{\rho_2 v_2^2}{2g_c} + \frac{\rho_2 g z_2}{g_c} + \Delta p_f \tag{17.4}$$

If both sides of Eq. (17.2) are multiplied by g_c/g and $W = 0$, then each term of the equation is expressed in the form of head, in ft or in. (m) of fluid column. That is,

$$\frac{g_c p_1'}{g\rho_1} + \frac{v_1^2}{2g} + z_1 = \frac{g_c p_2'}{g\rho_2} + \frac{v_2^2}{2g} + z_2 + \frac{g_c \Delta p_f}{\rho g} \tag{17.5}$$

Static Pressure, Velocity Pressure, and Total Pressure

Pressure is the force per unit area exerted by a fluid or solid. In an air duct system, a water piping system, or a refrigerant piping system, fluid pressure including air, water, refrigerant pressure at a surface or a level, or inside an enclosed vessel, or pressure difference between two surfaces is often measured under the following conditions:

- Fluid pressure is measured related to a datum of absolute vacuum. Such a measured fluid pressure is given as absolute pressure and is often represented by the pressure exerted at the bottom surface of a water column.
- Fluid pressure is often more conveniently measured related to a datum of atmospheric pressure. Such a measured fluid pressure is given as gauge pressure. The measured gauge pressure that is greater than the atmosperic pressure is expressed as positive gauge pressure or simply gauge pressure. That part of measured gauge pressure which is less than the atmospheric pressure is expressed as negative gauge pressure or vacuum.
- Fluid pressure is measured as a pressure difference, pressure drop, or pressure loss between two surfaces, two levels, or two cross-sectional surfaces. The involved two measured pressures must be either both gauge pressure or both absolute pressure.

Consider a supply duct system in a multistory building, as shown in Fig. 17.1. In Eq. (17.4), since $p_1' = p_{at1} + p_1$ and $p_2' = p_{at2} + p_2$, where p_1 and p_2 represent the gauge static pressure and p_{at1} and p_{at2} the atmospheric pressure added on the fluid at cross sections 1 and 2. The relationship of fluid properties between cross sections 1 and 2 can be expressed as

$$p_{at1} + p_1 + \frac{\rho_1 v_1^2}{2g_c} + \frac{\rho_1 g z_1}{g_c} = p_{at2} + p_2 + \frac{\rho_2 v_2^2}{2g_c} + \frac{\rho_2 g z_2}{g_c} + \Delta p_f \tag{17.6}$$

If the air temperature inside the air duct is equal to the ambient air temperature, and if the stack effect because of the difference in air densities between the air columns inside the air duct and the ambient air does not exist, then

$$p_{at1} - p_{at2} = (\rho_2 z_2 - \rho_1 z_1)\frac{g}{g_c}$$

Therefore, Eq. (17.6) becomes

$$p_1 + \frac{\rho_1 v_1^2}{2g_c} = p_2 + \frac{\rho_2 v_2^2}{2g_c} + \Delta p_f \tag{17.7}$$

Equation (17.7) is one of the primary equations used to determine the pressure characteristics of an air duct system that does not contain a fan and in which the stack effect is negligible.

Static Pressure. In Eq. (17.7), static pressures p_1 and p_2 are often represented by p_s. In air duct systems, its unit can be either Pa (pascal, or newtons per square meter) in SI units, or the height of water

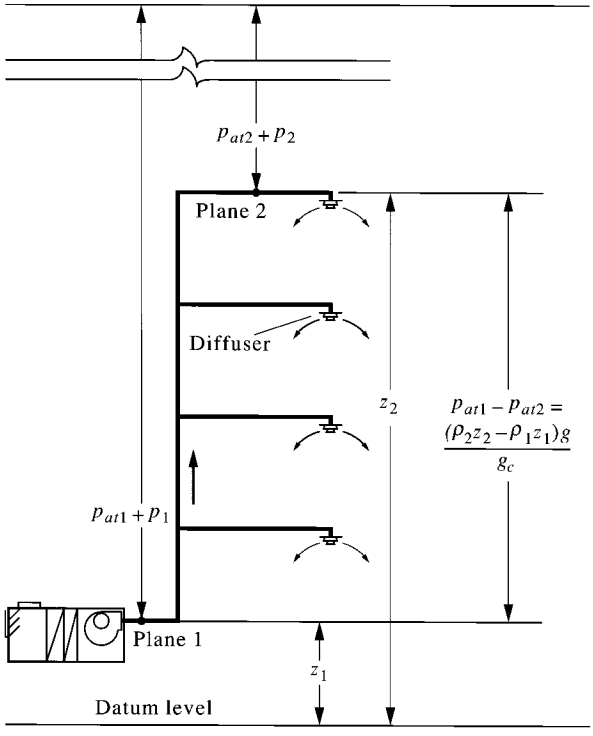


FIGURE 17.1 Pressure characteristics of an air duct system.

column, in inches, for I-P units. Either is expressed in gauge pressure or absolute pressure. The relationship between the static pressure p_s, in lb_f/ft^2, and the height of a water column H, in ft, is

$$p_s = \frac{\rho_w g H A}{g_c A} = \frac{\rho_w g H}{g_c} \tag{17.8}$$

where A = cross-sectional area of water column, ft^2
ρ = density of water, lb/ft^3

When the static pressure is expressed as the height of 1 in. of water column absolute pressure (1 in. WC) or 1 in. of water column gauge pressure (1 in. WG), for a density of water $\rho_w = 62.3\ lb/ft^3$ and numerically $g = g_c = 32.2$, we find from Eq. (17.8)

$$p_s = 1 \text{ in. WG} \qquad \text{and} \qquad \Delta p_s = 1 \text{ in. WC} = \frac{\rho_w g H}{g_c} = \frac{62.3 \times 32.2 \times 1}{32.2 \times 12} = 5.192\ lb_f/ft^2$$

That is 1 in. WC = 5.192 lb_f/ft^2. Because 1 lb_f/ft^2 = 47.88 Pa, 1 in. WC = 5.192 × 47.88 = 248.6 Pa.

Velocity Pressure. In Eq. (17.7), the term $\rho v^2/(2g_c)$ is called the velocity pressure, or dynamic pressure, and is represented by the symbol p_v, that is,

$$p_v = \frac{\rho v^2}{2g_c} \tag{17.9}$$

where ρ = air density, lb/ft^3. For air density $\rho = 0.075$ lb/ft^3, if the velocity pressure p_v is expressed in in. WC and the air velocity in fpm or ft/min, according to Eq. (17.9),

$$\left(\frac{v}{60}\right)^2 = \frac{5.192(2p_v g_c)}{\rho} = \frac{5.192 \times 2 \times 32.2\, p_v}{0.075} = 4458 p_v$$

and

$$v = 4005\sqrt{p_v} \qquad p_v = \left(\frac{v}{4005}\right)^2 \tag{17.10}$$

In SI units, if v is expressed in m/s and ρ in kg/m^3, then p_v is in Pa, and p_v can be calculated by Eq. (17.9). For SI units, $g_c = 1$.

Total Pressure. At any cross-sectional plane perpendicular to the direction of the airflow, the total pressure of the airstream p_t is defined as the sum of the static pressure p_s and the velocity pressure p_v, that is,

$$p_t = p_s + p_v \qquad p_v = p_t - p_s \tag{17.11}$$

From Eq. (17.11) velocity pressure p_v is also a kind of pressure difference. The units of p_t must be consistent with p_s and p_v. In I-P units, it is also indicated in inches WC or WG and in SI units in Pa absolute (abs.) or gauge (g). Substituting Eq. (17.11) into Eq. (17.7), we see that

$$p_{t1} = p_{t2} + \Delta p_f \tag{17.12}$$

Equation (17.12) is another primary equation that relates the pressure loss from friction and other sources, Δp_f, and the total pressure p_{t1} and p_{t2} at two cross sections of the air duct system.

Stack Effect

When an air duct system has an elevation difference and the air temperature inside the air duct is different from the ambient air temperature, the stack effect exists. It affects airflow at different elevations. During a hot summer day, when the density of the outdoor air is less than the density of the cold supply air inside the air duct, the pressure exerted by the atmospheric air column between z_1 and z_2, as shown in Fig. 17.1, is given as

$$p_{at1} - p_{at2} = \frac{\rho_o g(z_2 - z_1)}{g_c}$$

where ρ_o = mean density of the ambient air, lb/ft^3 (kg/m^3). And the pressure exerted by the air column inside the air duct between z_1 and z_2 is

$$\frac{\rho_2 g z_2 - \rho_1 g z_1}{g_c} = \frac{\rho_i g(z_2 - z_1)}{g_c}$$

where ρ_i = mean density of the supply air inside the air duct, in lb/ft^3 (kg/m^3). If the differences between the densities inside the air ducts, ρ_1 and ρ_i, and ρ_2 and ρ_i, are ignored, substituting these relationships into Eq. (17.6) yields

$$p_1 + \frac{\rho_1 v_1^2}{2g_c} + \frac{g(\rho_o - \rho_i)(z_2 - z_1)}{g_c} = p_2 + \frac{\rho_2 v_2^2}{2g_c} + \Delta p_f \tag{17.13}$$

and

$$p_{t1} + \frac{g(\rho_o - \rho_i)(z_2 - z_1)}{g_c} = p_{t2} + \Delta p_f \tag{17.14}$$

The third term on the left-hand side of Eq. (17.13) and the second term on the left-hand side of Eq. (17.14) are called the stack effect p_{st}, in lb_f/ft^2 (Pa),

$$p_{st} = \frac{g(\rho_o - \rho_i)(z_2 - z_1)}{g_c} \tag{17.15}$$

If p_{st} is expressed in in. WC (1 lb_f/ft^2 = 0.1926 in. WC), then

$$p_{st} = \frac{0.1926g(\rho_o - \rho_i)(z_2 - z_1)}{g_c} \tag{17.16}$$

For an upward supply duct system, $z_2 > z_1$. If cold air is supplied, $\rho_i > \rho_o$ and p_{st} is negative. If warm air is supplied, $\rho_o > \rho_i$ and p_{st} is positive. For a downward supply duct system with a cold air supply, p_{st} is positive; if there is a warm air supply, p_{st} is negative.

When supply air is at a temperature of 60°F (15.5°C) and a relative humidity of 80 percent, its density is 0.075 lb/ft^3 (1.205 kg/m^3). Also, if the space air has a temperature of 75°F (24°C) and a relative humidity of 50 percent, its density is 0.073 lb/ft^3 (1.171 kg/m^3). Numerically, $g = g_c$, for a difference of $z_2 - z_1 = 30$ ft (9.14 m), so

$$p_{st} = 0.1926\ (0.073 - 0.075)(30) = 0.0116 \text{ in. WC } (2.96 \text{ Pa})$$

For an air-handling unit or a packaged unit that supplies air to the same floor where it is located, the stack effect is usually ignored.

Laminar Flow and Turbulent Flow

Reynolds identified two types of fluid flow in 1883 by observing the behavior of a stream of dye in a water flow: laminar flow and turbulent flow. He also discovered that the ratio of inertial to viscous forces is the criterion that distinguishes these two types of fluid flow. This dimensionless parameter is now widely known as *Reynolds number* Re, or

$$\text{Re} = \frac{\rho v L}{\mu} = \frac{vL}{\nu} \tag{17.17}$$

where ρ = density of fluid, lb/ft^3 (kg/m^3)
v = velocity of fluid, ft/s (m/s)
L = characteristic length, ft (m)
μ = viscosity or absolute viscosity, $lb/ft \cdot s$ ($N \cdot s/m^2$)
ν = kinematic viscosity, ft^2/s (m^2/s)

Many experiments have shown that laminar flow occurs at Re $\leq$ 2000 in round ducts and pipes. A transition region exists between 2000 $<$ Re $\leq$ 4000. When Re $>$ 4000, the fluid flow is probably a turbulent flow.

At 60°F (15.5°C), the viscosity $\mu = 1.21 \times 10^5$ $lb/ft \cdot s$ (18.0×10^6 $N \cdot s/m^2$). For a round duct of 1-ft (0.305-m) diameter and an airflow through it of 3 ft/s (180 ft/min or 0.915 m/s), the Reynolds number is

$$\text{Re} = \frac{\rho v L}{\mu} = \frac{0.075 \times 3 \times 1}{1.21 \times 10^5} = 18{,}595$$

The Re of such an air duct is far greater than 4000. Therefore, the airflow inside the air duct is usually turbulent except within the boundary layer adjacent to the duct wall.

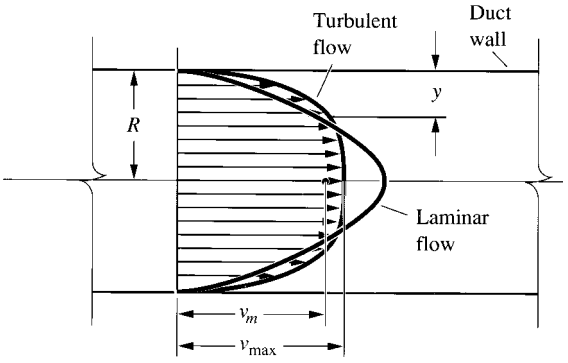


FIGURE 17.2 Velocity distribution in a circular duct.

Velocity Distribution

The velocity distributions of turbulent and laminar flow at a specific cross section in a circular duct that result from (1) the mechanical friction between the fluid particles and the duct wall and (2) the shearing stress of the viscous fluid are shown in Fig. 17.2. The difference between these two types of flow is significant.

For a fully developed turbulent flow, the air velocity v, in fpm (m/s), at various distances from the duct wall of a circular duct y, in ft (m), varies according to Prantdl's one-seventh power law as follows:

$$\frac{v}{v_{max}} = \left(\frac{v}{R}\right)^{1/7} \tag{17.18}$$

where v_{max} = maximum air velocity on centerline of air duct, fpm (m/s)
R = radius of duct, ft (m)

The mean air velocity v_m lies at a distance about $0.33R$ from the duct wall.

Equation of Continuity

For one-dimensional fluid flow at steady state, the application of the principle of conservation of mass gives the following equation of continuity:

$$\dot{m} = \rho_1 v_1 A_1 = \rho_2 v_2 A_2 \tag{17.19}$$

where $\dot{m}$ = mass flow rate, lb/s (kg/s)
A = cross-sectional area perpendicular to fluid flow, ft^2 (m^2)

Subscripts 1 and 2 indicate the cross sections 1 and 2 along the fluid flow.

If the differences in fluid density at various cross sections are negligible, then the equation of continuity becomes

$$\dot{V} = A_1 v_1 = A_2 v_2 \tag{17.20}$$

where $\dot{V}$ = volume flow rate of airflow, cfm (m^3/s)
v = mean air velocity at any specific cross section, fpm (m/s)

Theoretically, the velocity pressure p_{vt} should be calculated as

$$p_{vt} = \frac{\int [\rho v^2/(2g_c)]\, v\, dA}{\int v\, dA} \tag{17.21}$$

Its value is slightly different from p_v calculated from Eq. (17.9), which is based on mean velocity v. For a fully developed turbulent flow, $p_{vt} \approx 1.06 p_v$. Since most experimental results of pressure loss (indicated in terms of velocity pressure) are calculated by $p_v = \rho v_2/(2g_c)$, for the sake of simplicity p_v is used here instead of p_{vt}.

17.2 CHARACTERISTICS OF AIR FLOW IN DUCTS

Types of Air Duct

Air ducts can be classified into four types according to their transporting functions:

1. *Supply duct*. Conditioned air is supplied to the conditioned space.
2. *Return duct*. Space air is returned (1) to the fan room where the air-handling unit is installed or (2) to the packaged unit.
3. *Outdoor air duct*. Outdoor air is transported to the air-handling unit, to the fan room, or to the space directly.
4. *Exhaust duct*. Space air or contaminated air is exhausted from the space, equipment, fan room, or localized area.

Each of these four types of duct may also subdivide into headers, main ducts, and branch ducts or runouts. A header is that part of a duct that connects directly to the supply or exhaust fan before air is supplied to the main ducts in a large duct system. Main ducts have comparatively greater flow rates and size, serve a greater conditioned area, and, therefore, allow higher air velocities. Branch ducts are usually connected to the terminals, hoods, supply outlets, return grilles, and exhaust hoods. A vertical duct is called a riser. Sometimes, a header or a main duct is also called a trunk.

Pressure Characteristics of the Airflow

During the analysis of the pressure characteristics of airflow in a fan duct system such as the one in Fig. 17.3, it is assumed that the static pressure of the space air is equal to the static pressure of the atmospheric air, and the velocity pressure of the space air is equal to zero. Also, for convenient measurements and presentation, as previously mentioned, the pressure of the atmospheric air is taken as the datum, that is, $p_{at} = 0$, and pressure is expressed as gauge pressure. When $p > p_{at}$, p is positive; and if $p < p_{at}$, then p is negative. In a fan duct system, a fan or fans are connected to a ductwork and equipment.

At cross section R_1, as the recirculating air enters the return grille, both the total pressure p_t and static pressure p_s decrease as the result of the total pressure loss of the inlet. The velocity pressure p_v, indicated by the shaded section in Fig. 17.3, will gradually increase until it is equal to the velocity of the branch duct. Both p_t and p_s are negative so that air will flow from the conditioned space at a datum of 0 to a negative pressure. Because velocity pressure p_v is always positive in the direction of flow, from Eq. (17.11) $p_t = p_s + p_v$, so p_s is then smaller, or more negative, than p_t.

When the recirculating air flows through the branch duct segment R_1-1_1, elbow 1_1-1_2, branch duct segment 1_2-1_3, diffuser 1_3-1_4, and branch duct segment 1_4-1_5, both p_t and p_s drop because of the pressure losses. Velocity pressure p_v remains the same between cross sections R_1 and 1_3. It gradually decreases because of the diffuser 1_3-1_4 and remains the same between 1_4 and 1_5.

As the recirculating air flows through the converging tee 1_5-1, this straight-through stream meets with another branch stream of recirculating air from duct section R_2-1, at node or junction 1. The

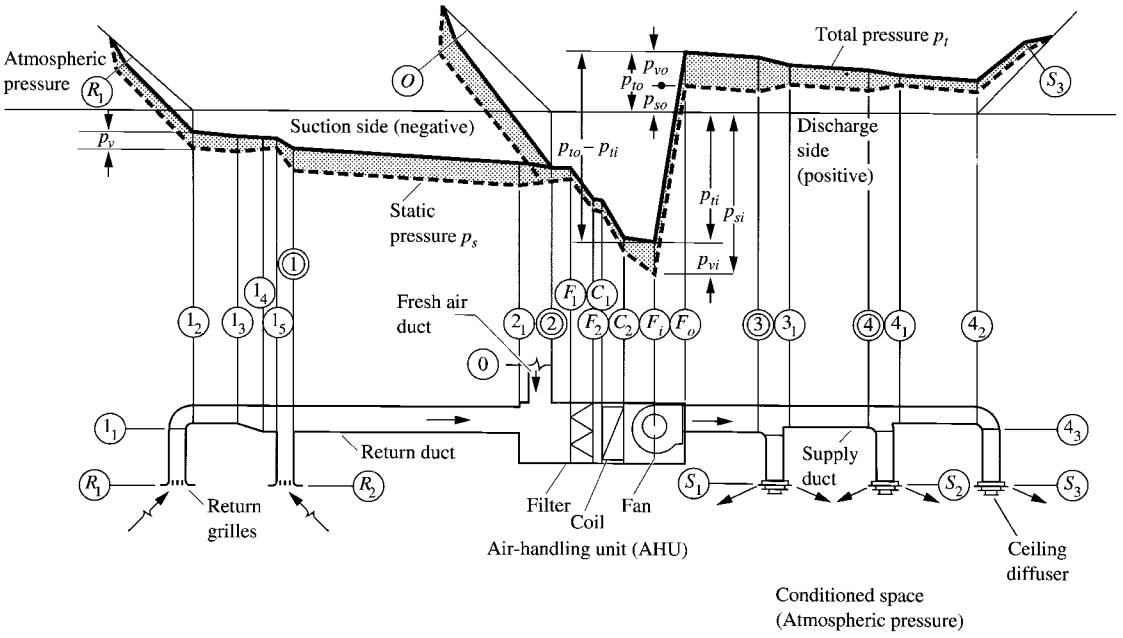


FIGURE 17.3 Pressure characteristics of a fan duct system.

combined airstream then becomes the main stream. Total pressure p_t usually decreases when the straight-through stream flows through the converging tee. It may increase if the velocity of the branch stream is much higher than that of the straight-through stream. However, the sum of the energies of the straight-through and the branch streams at the upstream side of the converging tee is always higher than the energy at the downstream side because of the pressure loss of the converging tee.

In the main duct section 1-2_1, p_t and p_s drop further while p_v increases because of the higher air velocity. This is mainly because of the greater volume flow rate in main duct section 1-2_1. When the recirculating air enters the air-handling unit, its velocity drops sharply to a value between 400 and 600 fpm (2 and 3 m/s). The air is then mixed with the outdoor airstream from the fresh air duct at node 2. After the mixing section, p_t and p_s drop sharply when the mixture of recirculating and outdoor air flows through the filter and the coil. Total pressure p_t and static pressure p_s drop to their minimum values before the inlet of the supply fan F_i.

At the supply fan, p_t is raised to its highest value at the fan outlet F_o. Both p_t and p_s decrease in duct section F_o-3. At junction 3, the airstream diverges into the main stream or straight-through stream and the branch stream. Although there is a drop in p_t after the main stream passing through the diverging tee 3-3_1, p_s increases because of the smaller p_v after 3_1. The increase in p_s due to the decrease of p_v is known as the *static regain* $\Delta p_{s,r}$ in in. WC (Pa). It can be expressed as

$$\Delta p_{s,r} = \left(\frac{v_3}{4005}\right)^2 - \left(\frac{v_{31}}{4005}\right)^2 = \frac{\rho(v_3^2 - v_{31}^2)}{2} \tag{17.22}$$

where v_3, v_{31} = air velocity at cross section 3 and 3_1, respectively, fpm (m/s). In duct sections 3-4 and 4-S_3, p_t decreases gradually along the direction of airflow. Finally, p_t, p_s, and p_v all drop to 0 after the supply air is discharged to the conditioned space.

The pressure characteristics along the airflow can be summarized as follows:

- In most sections, p_t of the main airstream decreases along the airflow. However, p_t of the main airstream may increase because of the higher velocity of the combined branch airstream.
- When air flows through the supply fan, mechanical work is done on the air so that p_t and p_s are raised from a minimum negative value at the fan inlet to a maximum positive value at the fan outlet.
- The pressure characteristics between any two cross sections of a duct system are governed by the change of p_t and the pressure loss Δp_f between these two cross sections $p_{t1} = p_{t2} + \Delta p_f$. Static pressure is always calculated as $p_s = p_t - p_v$.
- In a constant-volume air system, the airflow inside an air duct is considered steady and continuous. Because the change in p_s in a fan duct system is small when compared with p_{at}, the airflow is also considered incompressible.

System Pressure Loss

For an air system (a fan duct system), system pressure loss Δp_{sy}, in in. WC (Pa), is the sum of the total pressure losses of the return air system $\Delta p_{r,s}$, section R_1-2_1 in Fig. 17.3; the air-handling unit Δp_{AHU}, section 2_1-F_o, or the packaged unit Δp_{PU}; and the supply air system $\Delta p_{s,s}$, section F_o-S_3; all are expressed in in. WC (Pa). That is,

$$\begin{aligned}\Delta p_{sy} &= \Delta p_{r,s} + \Delta p_{AHU} + \Delta p_{s,s} \\ &= \Delta p_{r,s} + \Delta p_{PU} + \Delta p_{s,s}\end{aligned} \tag{17.23}$$

Both $\Delta p_{r,s}$ and $\Delta p_{s,s}$ may include the pressure losses of duct sections (including duct segments); duct fittings such as elbows, diffusers, and converging and diverging tees; components; and equipment.

The sum of $\Delta p_{r,s}$ and $\Delta p_{s,s}$ is called the external total pressure loss, or external pressure loss $\Delta p_{t,ex}$, as opposed to the pressure loss in the air-handling unit or packaged unit. The external pressure loss $\Delta p_{t,ex} = \Delta p_{r,s} + \Delta p_{s,s}$, in in. WC (Pa).

In commercial buildings, most air systems have a system pressure loss of 2.5 to 6 in. WC (625 to 1500 Pa). Of this, Δp_{AHU} usually has a value of 1.5 to 3 in. WC (375 to 750 Pa), and $\Delta p_{s,s}$ is usually less than 0.6 Δp_{sy} except in large auditoriums and indoor stadiums.

Criteria of Fan Energy Use

ASHRAE/IESNA Standard 90.1-1999 specifies that each air system having a total fan power exceeding 5 hp (3.7 kW) shall meet the following specified allowable fan system power (AFSP) in order to encourage heat recovery, relief fan, and other energy efficient means:

$$\text{(AFSP)} = \text{(fan power limitation)(temperature ratio)} + \text{(pressure credit)} + \text{(relief fan credit)} \tag{17.24}$$

Standard 90.1-1999 specifies fan power limitation (including supply, return, relief, and exhaust fans) as follows:

For constant-volume systems

$$\frac{P_{sy}}{\dot{V}_{s,d}} \le 0.0012 \text{ hp/cfm } (0.0019 \text{ W} \cdot \text{s/L}) \qquad \text{when } \dot{V}_{s,d} < 20{,}000 \text{ cfm}$$

$$\frac{P_{sy}}{\dot{V}_{s,d}} \le 0.0011 \text{ hp/cfm } (0.00174 \text{ W} \cdot \text{s/L}) \qquad \text{when } \dot{V}_{s,d} \ge 20{,}000 \text{ cfm} \tag{17.24a}$$

where P_{sy} = each air system total power input to the fan motors, hp (kW)
$\dot{V}_{s,d}$ = volume flow rate of air system at design conditions, cfm (L/s)

For variable-air-volume (VAV) system: nameplate fan motor power is 0.0017 hp/cfm (0.0027 W per L/s); when supply volume flow ≥ 20,000 cfm (9440 L/s), allowable nameplate fan motor power is 0.0015 hp/cfm (W per L/s). That is:

$$\frac{P_{sy}}{\dot{V}_{s,d}} \le 0.0017 \text{ hp/cfm } (0.0027 \text{ W} \cdot \text{s/L}) \qquad \text{when } \dot{V}_{s,d} < 20{,}000 \text{ cfm}$$

$$\frac{P_{sy}}{\dot{V}_{s,d}} \le 0.0015 \text{ hp/cfm } (0.0024 \text{ W} \cdot \text{s/L}) \qquad \text{when } \dot{V}_{s,d} \ge 20{,}000 \text{ cfm} \tag{17.24b}$$

P_{sy} can be calculated as

$$P_{sy} = \frac{\Delta p_{sy,d} \dot{V}_{s,d}}{6350 \eta_{f,d} \eta_{d,d} \eta_{m,d}} \tag{17.25a}$$

where $\Delta p_{sy,d}$ = system total pressure loss at design conditions, in. WC (Pa)
$\eta_{f,d}$, $\eta_{d,d}$, $\eta_{m,d}$ = fan total efficiency, drive efficiency, and motor efficiency at design conditions

Substituting Eq. (17.25*a*) into Eq. (17.24), for constant-volume systems, the system total pressure loss is

$$\frac{\Delta p_{sy}}{\eta_{f,d} \eta_{d,d} \eta_{m,d}} \le 7.6 \text{ in. WC (Pa)} \qquad \text{when } \dot{V}_{s,d} < 20{,}000 \text{ cfm}$$

$$\frac{\Delta p_{sy}}{\eta_{f,d} \eta_{d,d} \eta_{m,d}} \le 7.0 \text{ in. WC (Pa)} \qquad \text{when } \dot{V}_{s,d} \ge 20{,}000 \text{ cfm} \tag{17.25b}$$

For VAV systems, the system total pressure loss is

$$\frac{\Delta p_{sy}}{\eta_{f,d} \eta_{d,d} \eta_{m,d}} \le 10.8 \text{ in. WC (Pa)} \qquad \text{when } \dot{V}_{s,d} < 20{,}000 \text{ cfm}$$

$$\frac{\Delta p_{sy}}{\eta_{f,d} \eta_{d,d} \eta_{m,d}} \le 9.5 \text{ in. WC (Pa)} \qquad \text{when } \dot{V}_{s,d} \ge 20{,}000 \text{ cfm} \tag{17.25c}$$

In Eq. (17.24), temperature ratio can be calculated as:

$$\text{Temperature ratio} = \frac{T_{r,\text{set}} - T_{s,d}}{20} \tag{17.26}$$

Pressure credit, in hp (kW), can be calculated as:

$$\text{Pressure credit} = \frac{\sum_{n=1} \dot{V}_{s,d} (\Delta p_{\text{fil}} - 1.0)}{3718} + \frac{\sum_{m=1} \dot{V}_{s,d} \Delta p_{\text{HR}}}{3718} \tag{17.27a}$$

Relief fan credit, in hp (kW), can be calculated as:

$$\text{Relief fan credit} = P_{r,f} \left(1 - \frac{1 - \dot{V}_{r,f}}{\dot{V}_{s,d}} \right) \tag{17.27b}$$

where $T_{r,\text{set}}$ = room set point temperature, °F (°C)
$T_{s,d}$ = design supply air temperature for the zone in which the thermostat is located, °F(°C)
$\Delta p_{\text{fil}}, \Delta p_{\text{HR}}$ = total pressure loss of the filters and of the heat recovery coils, in WC (Pa)
$p_{r,f}$ = name plate relief fan motor power, hp (kW)
$\dot{V}_{r,f}$ = volume flow rate of relief fan at cooling design operation, cfm (L/s)

Consider a VAV system with a design supply volume flow rate $\dot{V}_{s,d} > 20{,}000$ cfm (9440 L/s). If a rooftop packaged system is used with a combined fan, drive, and motor efficiency $\eta_{f,d}\eta_{d,d}\eta_{m,d} = 0.45$, from Eq. (17.25*c*), the allowable system pressure loss for this rooftop package system (including supply and return fans) is

$$\Delta p_{\text{sy}} \leq \eta_{f,d}\eta_{d,d}\eta_{m,d}\,(9.5) \leq 0.45 \times 9.5 \leq 4.3 \text{ in. WC } (1070 \text{ Pa})$$

If the design system pressure loss $\Delta p_{\text{sy}} > 4.3$ in. WC (1070 Pa), the HVAC&R system designer is recommended either to use a relief fan instead of a return fan for relief fan credit, or to take into account the pressure credit if filter's total pressure loss is greater than 1 in. WC (250 Pa) or if there is heat recovery coil, increase the supply temperature differential $\Delta T_s = (T_{r,\text{set}} - T_s)$ to a value greater than 20°F (11.1°C) if they are cost effective to meet the fan power limitation in ASHRAE/IESNA Standard 90.1–1999.

For a VAV reheat central system of $\dot{V}_{s,d} \leq 20{,}000$ cfm (9440 L/s), if combined efficiency is 55 percent, from Eq. (17.25*c*), the allowable system pressure loss is

$$\Delta p_{\text{sy}} \leq \eta_{f,d}\eta_{d,d}\eta_{m,d}\,(9.5) \leq 0.55 \times 9.5 \leq 5.2 \text{ in. WC } (1300 \text{ Pa})$$

If the design system total pressure loss > 5.2 in. WC, the same as for the rooftop packaged system, means of relief fan credit, pressure credit due to filters and heat recovery coils, and the increase of the supply temperature differential greater than 20°F (11.1°C) should be considered. Refer to ASHRAE Standard 90.1-1999 for details.

17.3 DUCT CONSTRUCTION

Maximum Pressure Difference

Duct systems can be classified according to the maximum pressure difference between the air inside the duct and the ambient air (also called the static pressure differential) as ± 0.5 in. WC (± 125 Pa), ± 1 in. WC (± 250 Pa), ± 2 in. WC (± 500 Pa), ± 3 in. WC (± 750 Pa), ± 4 in. WC (± 1000 Pa), ± 6 in. WC (± 1500 Pa), and ± 10 in. WC (± 2500 Pa). In actual practice, the maximum pressure difference of the supply or return duct system in commercial buildings is usually less than ± 3 in. WC (± 750 Pa).

In commercial buildings, a low-pressure duct system has a static pressure differential of 2 in. WC (500 Pa) or less, and the maximum air velocity inside the air duct is usually 2400 fpm (12 m/s). A medium-pressure duct system has a static pressure differential of 2 to 6 in. WC (500 to 1500 Pa) with a maximum air velocity of about 3500 fpm (17.5 m/s). In industrial duct systems, including mechanical ventilation, mechanical exhaust, and industrial air pollution control systems, the pressure difference is often higher. In residential buildings, the static pressure differential of the duct systems is classified as ± 0.5 in. WC (± 125 Pa) or ± 1 in. WC (± 250 Pa).

Material

Underwriters' Laboratory (UL) classifies duct systems according to the flame spread and smoke developed of the duct material during fire as follows:

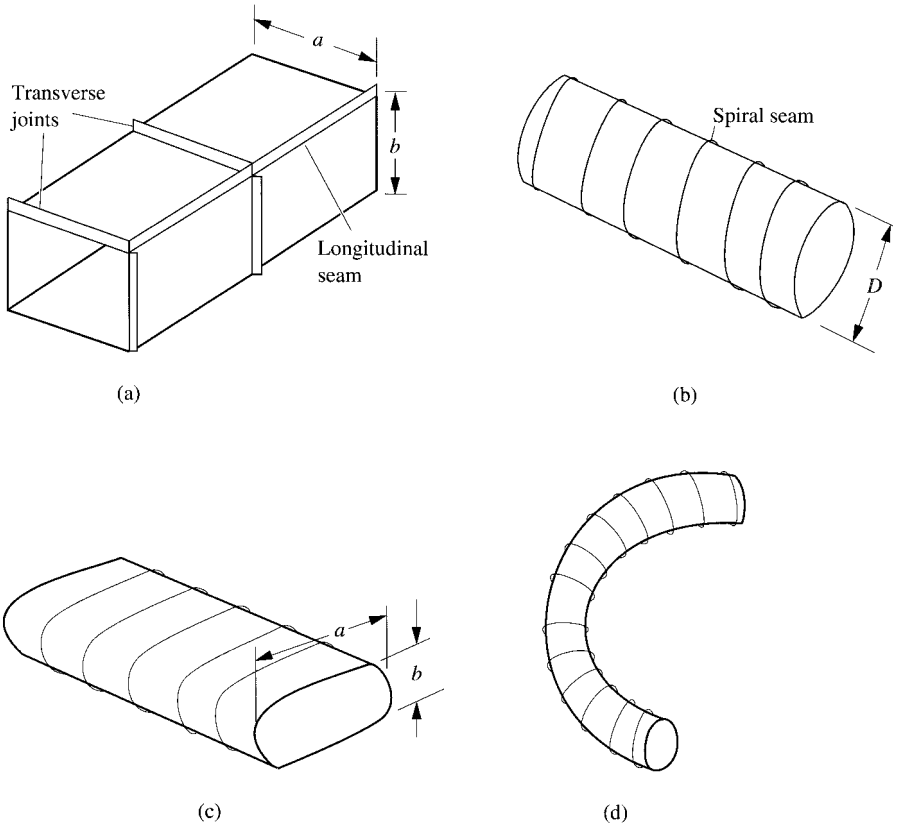


FIGURE 17.4 Various types of air duct: (*a*) rectangular duct; (*b*) round duct with spiral seam; (*c*) flat oval duct; (*d*) flexible duct.

Class 0. Zero flame spread, zero smoke developed.

Class 1. A flame spread rating of not more than 25 without evidence of continued progressive combustion and a smoke developed rating of not more than 50.

Class 2. A flame speed of 50 and a smoke developed rating of 100.

National Fire Protection Association (NFPA) Standard 90A specifies that the material of the ducts be iron; steel including galvanized sheets, aluminum, concrete, masonary; or clay tile. Ducts fabricated by these materials are listed as class 0. UL Standard 181 allows class 1 material to be used for ducts when they do not serve as risers for more than two stories or are not used in temperatures higher than 250°F (121°C). Fibrous glass and many flexible ducts that are factory-fabricated are approved by UL as class 1.

Ducts can be classified according to their shapes into rectangular, round, flat oval, and flexible, as shown in Fig. 17.4.

Rectangular Ducts

For the space available between the structural beam and the ceiling in a building, rectangular ducts have the greatest cross-sectional area. They are less rigid than round ducts and are more easily

TABLE 17.1 Thickness of Galvanized Sheet for Rectangular Ducts

Gauge	Thickness, in.		Nominal weight, lb/ft^2
	Nominal	Minimum	
30	0.0157	0.0127	0.656
28	0.0187	0.0157	0.781
26	0.0217	0.0187	0.906
24	0.0276	0.0236	1.156
22	0.0336	0.0296	1.406
20	0.0396	0.0356	1.656
18	0.0516	0.0466	2.156
16	0.0635	0.0575	2.656
14	0.0785	0.0705	3.281
13	0.0934	0.0854	3.906
12	0.1084	0.0994	4.531
11	0.1233	0.1143	5.156
10	0.1382	0.1292	5.781

Note: Minimum thickness is based on thickness tolerances of hot-dip galvanized sheets in cut lengths and coils (per ASTM Standard A525). Tolerance is valid for 48- and 60-in.-wide sheets.

Source: *ASHRAE Handbook 1988, Equipment.* Reprinted with permission.

fabricated on-site. The joints of rectangular ducts have a comparatively greater percentage of air leakage than factory-fabricated spiral-seamed round ducts and flat oval ducts, as well as fiberglass ducts. Unsealed rectangular ducts may have an air leakage from 15 to 20 percent of the supply volume flow rate. Rectangular ducts are usually used in low-pressure systems.

The ratio of the long side a to the short side b in a rectangular duct is called the *aspect ratio* R_{as}. The greater R_{as}, the higher the pressure loss per unit length as well as the heat loss and heat gain per unit volume flow rate transported. In addition, more labor and material are required.

Galvanized sheet or, more precisely, galvanized coated steel sheet, and aluminum sheet are the materials most widely used for rectangular ducts. To prevent vibration of the duct wall by the pulsating airflow, transverse joints and longitudinal seam reinforcements are required in ferrous metal ducts.

The galvanized sheet gauge and thickness for rectangular ducts are listed in Table 17.1. Table 17.2 gives specifications for rectangular ferrous metal duct construction for commercial systems based on the publication of the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) titled *HVAC Duct Construction Standards—Metal and Flexible*. For design and construction of an economical duct system, it is recommended to select an optimum combination of minimum galvanized sheet thickness, type of transverse joint reinforcement, and its maximum spacing for a specific duct dimension at a specific pressure differential between the air inside the duct and the ambient air.

For rectangular ducts, one uses the same metal thickness for all sides of the duct and evaluates duct reinforcement on each side separately. In Table 17.2, for a given duct dimension and thickness, letters indicate the type of duct reinforcement (rigidity class) and numbers indicate maximum spacing, in ft. Blanks indicate that reinforcement is not required, and dashes denote that such a combination is not allowed.

Transverse joint reinforcements, abridged from SMACNA's publication *HVAC Duct Construction Standard—Metal and Flexible* and *ASHRAE Handbook 1988, Equipment* are presented in Table 17.3. These must be matched with the arrangements in Table 17.2. Duct hangers should be installed at right angles to the centerline of the duct. Habjan (1984) recommended the following maximum duct hanger spacing:

TABLE 17.2 Rectangular Ferrous Metal Duct Construction for Commercial Buildings

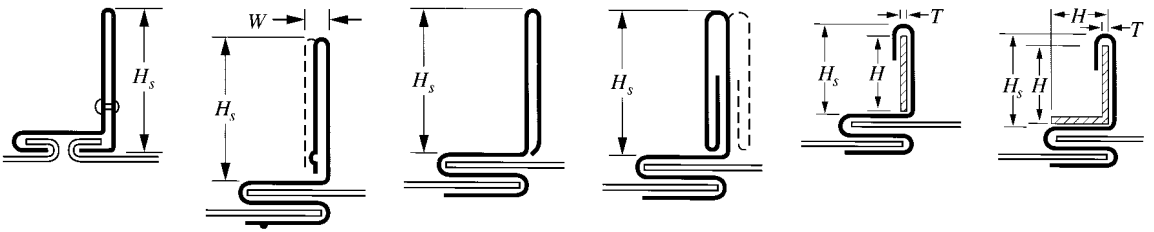
Duct dimensions, in.	Minimum galvanized steel thickness, in. (gauge)																		
	0.0575 (16)		0.0466 (18)			0.0356 (20)			0.0296 (22)				0.0236 (24)				0.0187 (26)		
	Pressure, in. WG																		
	±2	±3	±1	±2	±3	±1	±2	±3	±0.5	±1	±2	±3	±0.5	±1	±2	±3	±0.5	±1	±2
Up through 10																			
12																A-8			A-8
14												A-8			A-8	A-5		A-10	A-5
16								A-8			A-10	A-8		A-10	A-8	A-5		A-8	A-5
18								A-8			A-10	A-8		A-10	A-8	A-5		A-8	A-5
20					B-10		B-10	B-8		A-10	B-8	A-5		A-10	A-5	A-5	A-10	A-8	A-5
22				B-10	C-10	A-10	B-10	B-8		A-10	B-8	A-5	A-10	A-10	A-5	B-5	A-10	A-5	A-5
24				C-10	C-10	B-10	C-10	B-5		B-10	C-8	B-5	A-10	B-10	B-5	B-5	A-10	A-5	B-5
26	C-10	D-10		C-10	D-10	B-10	C-10	C-5	A-10	B-10	C-8	C-5	A-10	B-8	B-5	C-5	A-10	A-5	B-5
28	C-10	D-10	C-10	C-10	D-10	C-10	C-8	C-5	B-10	C-10	C-5	C-5	B-10	C-8	C-5	C-4	B-8	B-5	B-4
30	D-10	D-10	C-10	D-10	D-8	C-10	D-8	C-5	B-10	C-10	C-5	C-5	B-10	C-8	C-5	C-4	B-8	B-5	C-4
36	E-10	E-8	D-10	E-8	E-5	D-10	D-5	E-5	C-10	D-8	D-5	D-4	C-8	C-5	D-4	D-4	C-5	C-5	—
42	E-8	E-5	E-10	E-5	E-5	D-8	E-5	E-5	D-8	D-5	E-5	E-4	D-8	D-5	E-4	E-3	D-5	D-4	—
48	G-8	G-5	E-8	F-5	G-5	E-5	F-5	F-4	D-8	E-5	E-4	E-3	D-5	E-5	E-3	E-2.5	D-5	D-4	—
54	G-5	H-5	F-8	G-5	H-5	E-5	F-4	G-3	D-5	E-5	F-3	G-3	D-5	E-4	F-3	E-2.5	D-5	—	—
60	H-5	H-5	G-8	H-5	H-4	F-5	G-4	G-3	E-5	F-5	G-3	G-2.5	E-5	F-4	G-2.5	G-2.5	E-4	—	—
72	I-5	I-4	H-5	H-4	H-3	G-4	H-3	H-3	F-5	G-4	H-3	H-2.5	F-4	H-2	H-2	H-2	—	—	—
84	J-4	J-3	I-5	J-4	J-3	H-4	I-3	J-2.5	H-5	—	—	—	G-4	—	—	—	—	—	—
96	K-4	L-3	I-4	K-3	K-2.5	I-3	J-2.5	J-2	H-4	—	—	—	—	—	—	—	—	—	—
Over 96	H-2.5 plus rods	H-2.5 plus rods	H-2.5 plus rods	H-2.5 plus rods	H-2.5 plus rods	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Note: For a given duct thickness, letters indicate type (rigidity class) of duct reinforcement (see Table 17.3); numbers indicate maximum spacing (ft) between duct reinforcement. Use the same metal duct thickness on all duct sides.

Source: Adapted with permission from SMACNA, *HVAC Duct Construction Standards—Metal and Flexible*. Refer to this standard for complete details.

TABLE 17.3 Transverse Joint Reinforcement

Pressure, 4 in. WG maximum*



Minimum rigidity class	Standing drive slip	Standing *S*		Standing *S*	Standing *S*	Standing *S* (bar-reinforced)	Standing *S* (angle-reinforced)
	$H_s \times T$ (min), in.	*W*, in.	$H_s \times T$ (min), in.	$H_s \times T$ (min), in.	$H_s \times T$ (min), in.	$H_s \times T$ (min) plus reinforcement ($H \times T$), in.	
A	Use class B		Use class C	$\frac{1}{2} \times 0.0187$			
B	$1\frac{1}{8} \times 0.0187$			$\frac{1}{2} \times 0.0296$	Use class D		
C	$1\frac{1}{8} \times 0.0296$	—	1×0.0187	1×0.0187		Use class F	
D	—	—	1×0.0236	1×0.0236	$1\frac{1}{8} \times 0.0187$		
E	—	$\frac{3}{16}$	$1\frac{1}{8} \times 0.0356$	—	$1\frac{1}{8} \times 0.0466$		
F	—	$\frac{3}{16}$	$1\frac{5}{8} \times 0.0296$	—	$1\frac{1}{2} \times 0.0236$	$1\frac{1}{2} \times 0.0236$ plus $1\frac{1}{2} \times \frac{1}{8}$ bar	
G	—	$\frac{3}{16}$	$1\frac{5}{8} \times 0.0466$	—	$1\frac{1}{2} \times 0.0466$	$1\frac{1}{2} \times 0.0296$ plus $1\frac{1}{2} \times \frac{1}{8}$ bar	
H	—	—	—	—	—	$1\frac{1}{2} \times 0.0356$ plus $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{3}{16}$ angle	
I	—	—	—	—	—	2×0.0356 plus $2 \times 2 \times \frac{1}{8}$ angle	
J	—	—	—	—	—	2×0.0356 plus $2 \times 2 \times \frac{3}{16}$ angle	

* Acceptable to 36 in. length at 3 in. WG and to 30 in. length at 4 in. WG.

Source: Adapted with permission from SMACNA, *HVAC Duct Construction Standards—Metal and Flexible* and *ASHRAE Handbook 1988, Equipment*. Refer to SMACNA Standard for complete details.

TABLE 17.4 Round Ferrous Metal Duct Construction for Duct Systems in Commercial Buildings

	Minimum galvanized steel thickness, in.						
	Pressure, − 2 in. WG			Pressure, + 2 in. WG			
Duct diameter, in.	Spiral seam duct	Longitudinal seam duct	Fittings	Spiral seam duct	Longitudinal seam duct	Fittings	Suggested type of joint
Up through 8	0.0157	0.0236	0.0236	0.0157	0.0157	0.0187	Beaded slip
14	0.0187	0.0236	0.0236	0.0157	0.0187	0.0187	Beaded slip
26	0.0236	0.0296	0.0296	0.0187	0.0236	0.0236	Beaded slip
36	0.0296	0.0356	0.0356	0.0236	0.0296	0.0296	Beaded slip
50	0.0356	0.0466	0.0466	0.0296	0.0356	0.0356	Flange
60	0.0466	0.0575	0.0575	0.0356	0.0466	0.0466	Flange
84	0.0575	0.0705	0.0705	0.0466	0.0575	0.0575	Flange

Source: Adapted with permission from SMACNA, *HVAC Duct Construction Standards—Metal and Flexible*. Refer to SMACNA Standard for complete details.

Duct area	Maximum spacing, ft (m)
Up to 4 ft^2 (0.37 m^2)	8 (2.4)
Between 4 and 10 ft^2 (0.37 and 0.93 m^2)	6 (1.8)
Larger than 10 ft^2 (0.93 m^2)	4 (1.2)

Round Ducts

For a specified cross-sectional area and mean air velocity, a round duct has less fluid resistance against airflow than rectangular and flat oval ducts. Round ducts also have better rigidity and strength. The spiral- and longitude-seamed round ducts used in commercial buildings are usually factory-fabricated to improve the quality and sealing of the ductwork. The pressure losses can be calculated more precisely than for rectangular ducts, and result in a better balanced system. Air leakage can be maintained at about 3 percent as a result of well-sealed seams and joints. Round ducts have much smaller radiated noise breakout from the duct than rectangular and flat oval ducts.

The main disadvantage of round ducts is the greater space required under the beam for installation. Factory-fabricated spiral-seamed round ducts are the most widely used air ducts in commercial buildings. The standard diameters of round ducts range from 4 to 20 in. in 1-in. (100 to 500 mm in 25-mm) increments, from 20 to 36 in. in 2-in. (500 to 900 mm in 50-mm) increments, and from 36 to 60 in. in 4-in. (900 to 1500 mm in 100-mm) increments. The minimum thickness of galvanized sheet and fittings for round ducts in duct systems in commercial buildings is listed in Table 17.4.

Many industrial air pollution control systems often require a velocity around 3000 fpm (15 m/s) or higher to transport particulates. Round ducts with thicker metal sheets are usually used in such applications.

Flat Oval Ducts

Flat oval ducts, as shown in Fig. 17.4, have a cross-sectional shape between rectangular and round. They share the advantages of both the round and the rectangular duct with less large-scale air turbulence and a small depth of space required during installation. Flat oval ducts are quicker to install and have lower air leakage because of the factory fabrication.

Flat oval ducts are made in either spiral seam or longitudinal seam. The minimum thickness of the galvanized sheet and fittings for flat oval duct systems used in commercial buildings is presented in Table 17.5.

TABLE 17.5 Flat Oval Duct Construction for Positive-Pressure Duct Systems in Commercial Buildings

Major axis, in.	Minimum galvanized steel thickness, in.			Suggested type of joint
	Spiral seam duct	Longitudinal seam duct	Fittings	
Up through 24	0.0236	0.0356	0.0356	Beaded slip
36	0.0296	0.0356	0.0356	Beaded slip
48	0.0296	0.0466	0.0466	Flange
60	0.0356	0.0466	0.0466	Flange
70	0.0356	0.0575	0.0575	Flange
Over 70	0.0466	0.0575	0.0575	Flange

Source: Adapted with permission from SMACNA, *HVAC Duct Construction Standards—Metal and Flexible*. Refer to SMACNA Standard for complete details.

Flexible Ducts

Flexible ducts are often used to connect the main duct or the diffusers to the terminal box. Their flexibility and ease of removal allow allocation and relocation of the terminal devices. Flexible ducts are usually made of multiple-ply polyester film reinforced by a helical steel wire core or corrugated aluminum spiral strips. The duct is often insulated by a fiberglass blanket 1 or 2 in. (25 to 50 mm) thick. The outer surface of the flexible duct is usually covered with aluminum foil or other types of vapor barriers to prevent the permeation of water vapor into the insulation layer.

The inside diameter of flexible ducts may range from 2 to 10 in. in 1-in. (50 to 250 mm in 25-mm) increments and from 12 to 20 in. in 2-in. (300 to 500 mm in 50-mm) increments. The flexible duct should be as short as possible, and its length should be fully extended to minimize flow resistance.

Fiberglass Ducts

Fiberglass duct boards are usually made in 1-in. (25-mm) thickness. They are fabricated into rectangular ducts by closures. A fiberglass duct with a 1.5-in. (38-mm) thickness may be used in the Gulf area of the United States where the climate is hot and humid in summer, to minimize duct heat gain. Round molded fiberglass ducts are sometimes used.

Fiberglass ducts have a good thermal performance. For a 1-in. (25-mm) thickness of duct board, the U value is 0.21 $\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ at 2000 fpm (1.192 $\text{W/m}^2 \cdot {}^\circ\text{C}$ at 10 m/s) air velocity, which is better than a galvanized sheet metal duct with a 1-in. (25-mm) inner liner. Fiberglass duct has good sound attenuation characteristics. Its air leakage is usually 5 percent or less, which is far less than that of a sheet-metal rectangular duct that is not well sealed. Another important advantage of fiberglass duct is its lower cost.

The closures, also called taping systems, are tapes used to form rectangular duct sections from duct boards and to join the sections and fittings into an integrated duct system. The improved acrylic pressure-sensitive tapes provide a better bond than before. Heat-sensitive solid polymer adhesive closures show themselves to be good sealing tapes even if dust, oil, or water is present on the surface of the duct board. Mastic and glass fabric closures are also used in many fiberglass duct systems.

Tests show that the ongoing emission of fiberglass from the duct board was less than that contained in outdoor air. Fiberglass ducts have a slightly higher friction loss than galvanized sheet duct (0.03 in. WC or 7.5 Pa greater for a length of 100 ft or 30.5 m). They are also not as strong as metal sheet duct. They must be handled carefully to prevent damage during installation.

Fiberglass ducts are used in duct systems with a pressure differential of $\pm$ 2 in. WC ($\pm$ 500 Pa) or less. Many codes restrict the use of fiberglass in sensitive areas such as operating rooms and maternity wards.

17.4 DUCT HEAT GAIN, HEAT LOSS, AND DUCT INSULATION

Temperature Rise or Drop due to Duct Heat Gain or Loss

The temperature rise or drop from duct heat gain or loss is one of the parameters that affect the supply air temperature as well as the supply volume flow rate in the air conditioning system design. Heat gain or loss through the duct wall of a rectangular duct section with a constant-volume flow rate q_d, in Btu/h (W), can be calculated as

$$q_d = UPL\left(T_{\text{am}} - \frac{T_{\text{en}} + T_{\text{lv}}}{2}\right) \tag{17.28}$$

For a round duct section

$$q_d = U\pi D_d L\left(T_{\text{am}} - \frac{T_{\text{en}} - T_{\text{lv}}}{2}\right) \tag{17.29}$$

where U = overall heat transfer coefficient of duct wall, Btu/h·ft²·°F (W/m²·°K)
P, L = perimeter and length of duct, ft (m)
D_d = diameter of round duct, ft (m)
T_{am} = temperature of ambient air, °F (°C)
$T_{\text{en}}, T_{\text{lv}}$ = temperature of air entering and leaving duct section, °F (°C)

The temperature increase or drop of the air flowing through a duct section is given as

$$T_{\text{lv}} - T_{\text{en}} = \frac{q_d}{60A_d v\rho_s c_{pa}} \tag{17.30}$$

where A_d = cross-sectional area of duct, ft² (m²). In Eq. (17.30), the mean air velocity v is expressed in fpm [m/(60 s)].

We substitute Eq. (17.28) into Eq. (17.30). For rectangular duct let

$$y = \frac{120\,A_d v\rho_s c_{pa}}{UPL} \tag{17.31}$$

For round duct, let

$$y = \frac{30D_d v\rho_s c_{pa}}{UL} \tag{17.32}$$

Then the temperature of air leaving the duct section is

$$T_{\text{lv}} = \frac{2T_{\text{am}} + T_{\text{en}}(y - 1)}{y + 1} \tag{17.33}$$

Duct Insulation

Duct insulation is mounted or inner-lined to reduce heat loss and heat gain as well as to prevent the condensation on the outer surface of the duct. It is usually in the form of duct wrap (outer surface), duct inner liner, or fiberglass duct boards. Duct liner provides both thermal insulation and sound attenuation. The thickness of an insulation layer is based on economical analysis.

ASHRAE/IESNA Standard 90.1-1999 mandates that all supply and return ducts and plenums installed as part of an HVAC&R air distribution system shall be thermally insulated as listed in Table 17.6. The insulated R-values in h·ft²·°F/Btu (m²·°C/W), are for the insulation installed and

TABLE 17.6 Minimum Duct Insulation R-Value,* Cooling and Heating Supply Ducts and Return Ducts

Climate zone			Duct location						
Envelop criteria table	HDD65	CDD50	Exterior	Ventilated attic	Unvented attic with backloaded ceiling	Unvented attic with roof insulation	Unconditioned space†	Indirectly conditioned space§	Buried
			Heating ducts only						
5-1 to 5-7	0-1800	All	None	None	None	None	None	None	None
5-8 to 5-12	1801-3600	All	R-3.5	None	None	None	None	None	None
5-13 to 5-15	3601-5400	All	R-3.5	None	None	None	None	None	None
5-16 to 5-18	5401-7200	All	R-6	R-3.5	None	None	None	None	R-3.5
			Cooling only ducts						
5-15, 18, 20, 22 to 26	All	0-1800	R-1.9	R-1.9	R-1.9	R-1.9	R-1.9	None	None
5-12, 14, 17, 19, 21	All	1801-3600	R-3.5	R-1.9	R-3.5	R-1.9	R-1.9	None	None
5-7, 9, 11, 13, 16	All	3601-5400	R-3.5	R-3.5	R-6	R-1.9	R-1.9	None	None
5-4, 6, 8, 10	All	5401-7200	R-6	R-6	R-6	R-3.5	R-1.9	None	None
			Combined heating and cooling ducts						
5-9	1801-2700	0-5400	R-6	R-3.5	R-6	R-1.9	R-1.9	None	
5-10	2701-3600	5401 +	R-6	R-6	R-6	R-3.5	R-3.5	None	
5-11	2701-3600	3601-5400	R-6	R-6	R-6	R-3.5	R-3.5	None	
5-12	2701-3600	0-3600	R-3.5	R-3.5	R-3.5	R-1.9	R-1.9	None	
5-13	3601-5400	3601 +	R-6	R-6	R-6	R-3.5	R-3.5	None	
5-14	3601-5400	1801-3600	R-6	R-3.5	R-6	R-1.9	R-3.5	None	
5-15	3601-5400	0-1800	R-3.5	R-3.5	R-3.5	R-1.9	R-1.9	None	
			Return ducts						
5-1 to 5-26	All climates		R-3.5	R-3.5	R-3.5	None	None	None	None

*Insulation R-values, measured in $(h \cdot ft^2 \cdot °F)/Btu$, are for the insulation as installed and do not include film resistance. The required minimum thicknesses do not consider water vapor transmission and possible surface condensation. Where exterior walls are used as plenum walls, wall insulation shall be as required by the most restrictive condition of 6.2.4.2 or Section 5. Insulation resistance measured on a horizontal plane in accordance with ASTM C518 at a mean temperature of 75°F at the installed thickness.

†Includes crawl spaces, both ventilated and nonventilated.

§Includes return air plenums with or without exposed roofs above.

Source: ASHRAE/IESNA Standard 90.1-1999. Reprinted by permission.

do not include air film resistances. The required minimum thickness does not consider water vapor transmission and possible surface condensation. The recommended thickness of insulation layer, or duct wrap, for *R*-3.5 $h \cdot ft^2 \cdot °F/Btu$ (0.62 $m^2 \cdot °C/W$) is 1 to 1.5 in. (25 to 38 mm), and for *R*-6 $h \cdot ft^2 \cdot °F/Btu$ (1.06 $m^2 \cdot °C/W$) is 2 to 3 in. (50 to 75 mm).

Exceptions of duct and plenum insulation include

- Factory-installed plenums, casings, and ductwork as part of the equipment
- Ducts or plenums located in heated spaces, semiheated spaces, or cooled spaces
- For runout less than 10 ft (3 m) in length to air terminals or air outlets, *R*-value of insulation need not exceed *R*-3.5.
- Backs of air outlets and outlet plenums exposed to unconditioned or indirectly conditioned space with face area exceeding 5 ft^2 (0.5 m^2) need not exceed *R*-2, for those 5 ft^2 (0.5 m^2) or smaller need not be insulated.

If the temperature of the ambient air is 80°F (26.7°C) with a relative humidity of 50 percent, its dew point is 59°F (18.3°C). Only when the outer surface temperature of the duct $T_{sd} > 59$°F (15°C) will condensation not occur. Refer to ASHRAE Standard 90.1-1999 for details.

Temperature Rise Curves

Duct heat gain or loss and the temperature rise or drop of the air inside the duct depend on air velocity, duct dimensions, and duct insulation. Figure 17.5 shows curves for the temperature rise in round ducts. These curves are calculated according to Eqs. (17.30) to (17.33) under these conditions:

- Thickness of the insulation layer of duct wrap is 1.5 in. (38 mm), and the thermal conductivity of the insulating material $k = 0.30$ Btu·in/h·ft^2·°F (0.043 W/m·°C).
- Heat transfer coefficient of the outer surface of the duct

$$h = 1.6 \text{ Btu/h·ft}^2\text{·°F } (9.1 \text{ W/m}^2\text{·°C}) \tag{17.34}$$

- Convective heat-transfer coefficient of the inside surface h_c, in Btu/h·ft^2·°F, can be calculated by

$$h_c = 0.023 \left(\frac{k}{D_d}\right) \text{Re}_D^{0.8} \text{Pr}^{0.4} \tag{17.35}$$

where Re_D = Reynolds number based on duct diameter as characteristic length
Pr = Prandtl number

- Air temperature inside the air duct is assumed to be 55°F (12.8°C).
- Temperature difference between the air inside the duct and the ambient air surrounding the duct is 25°F (13.9°C).

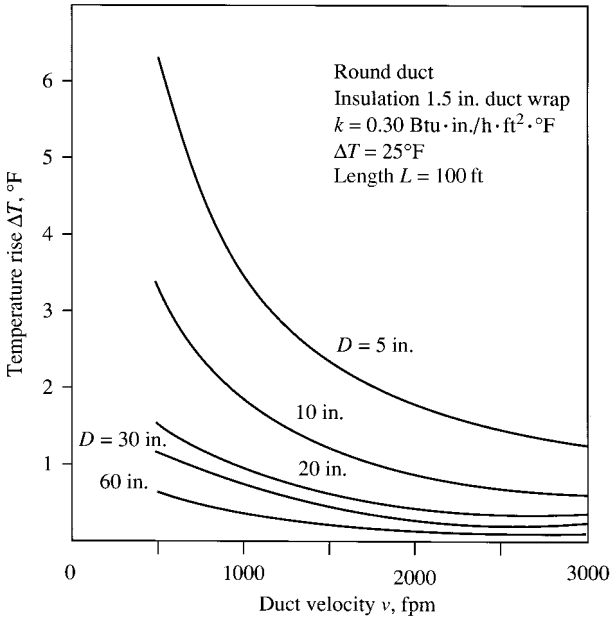


FIGURE 17.5 Temperature rise curves from duct heat gain.

If the temperature rise or drop has been determined, duct heat gain or heat loss can be either expressed in percentage of supply temperature differential or calculated by Eqs. (17.28) and (17.29).

17.5 FRICTIONAL LOSSES

In an air duct system, there are two types of resistance against the airflow: frictional losses and dynamic losses.

Darcey-Weisbach Equation

Frictional losses result mainly from the shearing stress between the fluid layers of the laminar sublayer, which is adjacent to the surface of the duct wall. Friction also exists when the fluid particles in the turbulent flow bump against the protuberances of the duct wall. These lead to the production of eddies and energy loss. Friction losses occur along the entire length of the air duct.

For a steady, incompressible fluid flow in a round duct or a circular pipe, friction head loss H_f, in ft (m) of air column, can be calculated by the Darcy-Weisbach equation in the following form:

$$H_f = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{17.36}$$

If the friction loss is presented in the form of pressure loss Δp_f, in $\text{lb}_\text{f}/\text{ft}^2$ (Pa), then

$$\Delta p_f = f\left(\frac{L}{D}\right)\left(\frac{\rho v^2}{2g_c}\right) \tag{17.37}$$

where f = friction factor
L = length of duct or pipe, ft (m)
D = diameter of duct or pipe, ft (m)
v = mean air velocity in duct, ft/s (m/s)

If air velocity v is expressed in fpm and Δp_f is expressed in in. WC, then a conversion factor $F_{\text{cv}} = (1/5.19)(1/60)^2 = 0.0000535$ should be used. That is,

$$\Delta p_f = 0.0000535f\left(\frac{L}{D}\right)\left(\frac{\rho v^2}{2g_c}\right) \tag{17.38}$$

In Eqs. (17.36), (17.37), and (17.38), strictly speaking, D should be replaced by hydraulic diameter D_h. But for round ducts and pipes, $D = D_h$.

Friction Factor

Friction loss is directly proportional to the friction factor f, which is dimensionless. The relationship between f and the parameters that influence the magnitude of the friction factor is shown in Fig. 17.6, which is called a Moody diagram. In Fig. 17.6, the term ε represents the absolute roughness of the surface protuberances, expressed in ft (m), as shown in Fig. 17.7. The ratio ε/D indicates the relative roughness of the duct or pipe.

For laminar flow in an air duct when $\text{Re}_D < 2000$, f is affected mainly by the viscous force of the fluid flow and, therefore, is a function of Re_D only. That is,

$$f = \frac{64}{\text{Re}_D} \tag{17.39}$$

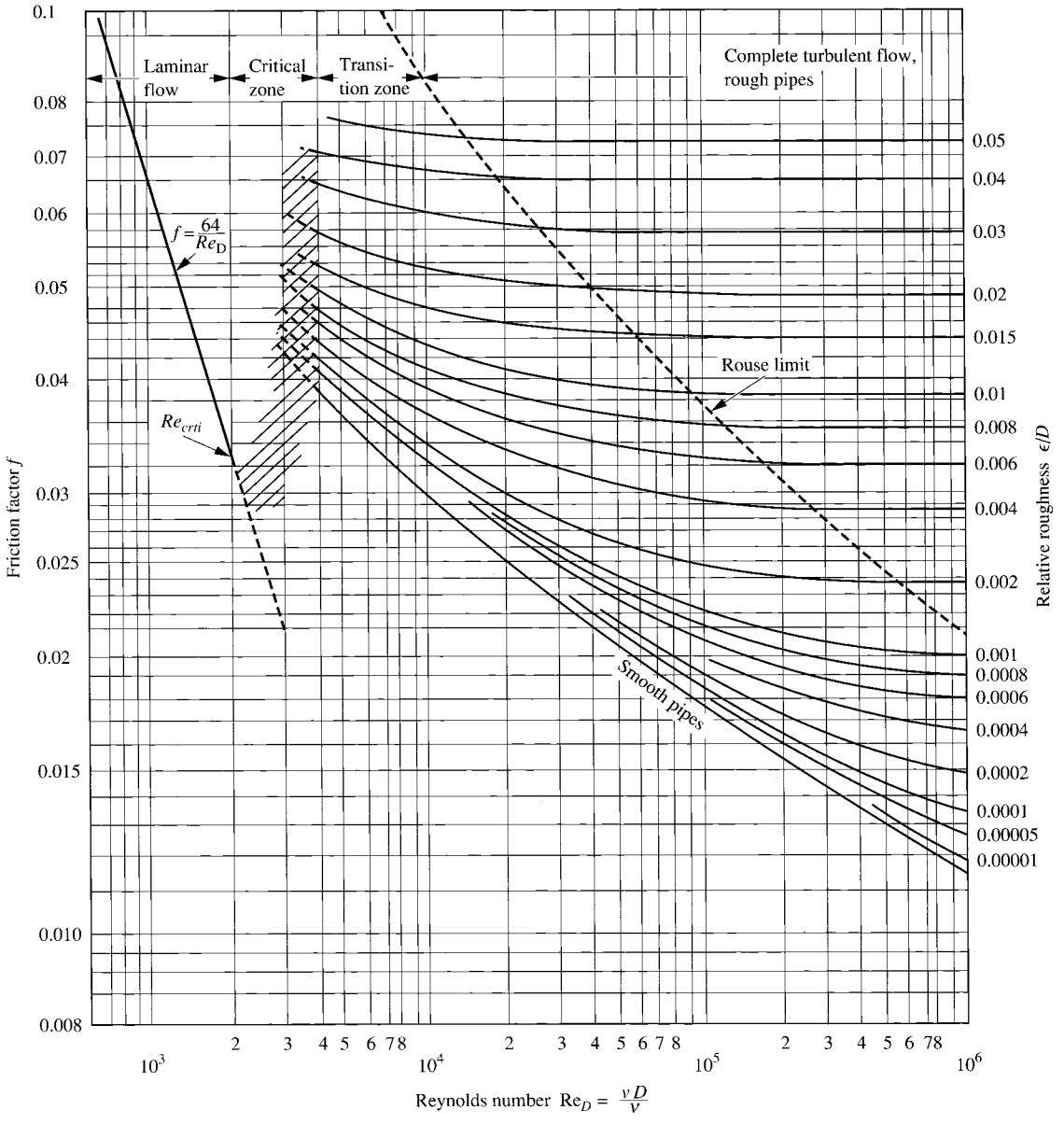


FIGURE 17.6 Moody diagram. (*Source: Moody, L.F., Transactions A.S.M.E., vol. 66, 1994. Reprinted with permission.*)

In an ideal smooth tube or duct, that is, $\varepsilon/D < 10^{-5}$, if $\mathrm{Re}_D > 4000$, the surface roughness is submerged in the laminar sublayer with a thickness of δ, and the fluid moves smoothly, passing over the protuberances. In this case, f decreases with an increase of Re_D. The relationship between f and Re_D may be expressed by the Blasius empirical formula

$$f = \frac{0.316}{\mathrm{Re}_D^{0.25}} \tag{17.40}$$

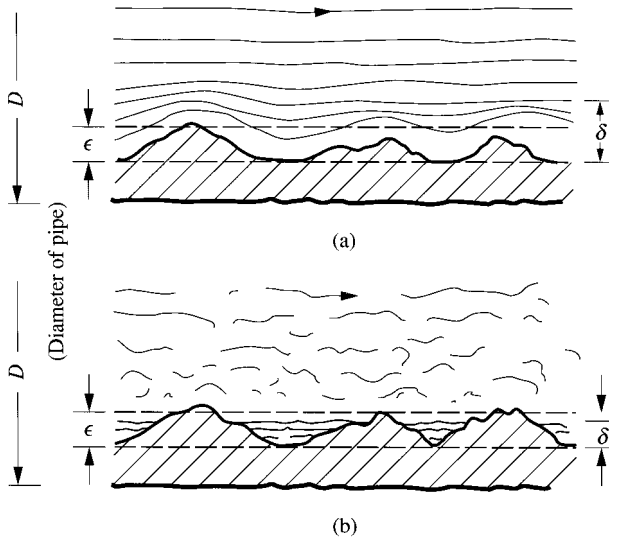


FIGURE 17.7 Modes of airflow when air passes over and around surface protuberances of the duct wall: (*a*) $\delta > \varepsilon$; (*b*) $\delta < \varepsilon$.

With a further increase of Re_D, the laminar sublayer becomes thinner, even thinner than the height of the irregularities ε, that is, $\delta < \varepsilon$. The protuberances form the separation of fluid flow, enhance the formation of vortices, and, therefore, increase the pressure loss as well as the value of f at greater Re_D.

If Re_D is beyond a limit called the Rouse limit, f depends mainly on the relative roughness of the duct wall. The Rouse limit line can be determined by $\mathrm{Re}_D = 200/\sqrt{f(\varepsilon/D)}$, as shown in Fig. 17.6.

Duct Friction Chart

In most air ducts, Re_D ranges from 1×10^4 to 2×10^6, and ε/D may vary from 0.005 to 0.00015. This covers a transition zone between hydraulic smooth pipes and the Rouse limit line. Within this region, f is a function of both Re_D and ε/D. Colebrook (1939) recommended the following empirical formula to relate Re_D, ε/D, and f for air ducts:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{\mathrm{Re}_D\sqrt{f}}\right) \tag{17.41}$$

In Eq. (17.41), ε and D must be expressed in the same units. Swamee and Jain suggested the use of an explicit expression that can give approximately the same value of f as Colebrook's formula:

$$f = \frac{0.25}{\{\log[\varepsilon/(3.7D) + 5.74/(0.9\,\mathrm{Re}_D)]\}^2} \tag{17.42}$$

In air duct calculations, a rough estimate of f can be determined from the Moody diagram based on knowing the relative roughness ε/D and Re_D.

For practical calculations, a friction chart for round ducts, developed by Wright, in the form shown in Fig. 17.8, is widely used. In this chart, air volume flow rate $\dot{V}$, in cfm, and the frictional

loss per unit length Δp_f, in in. WC per 100 ft, are used as coordinates. The mean air velocity v, in fpm, and the duct diameter are shown in inclined lines in this chart.

The duct friction chart can be applied to the following conditions without corrections:

- A temperature from 41 to 95°F (5 to 35°C)
- Up to an elevation of 1600 ft (488 m)
- Duct pressure difference of ± 20 in. WC (± 5000 Pa) with respect to ambient air pressure
- Duct material of medium-smooth roughness

Roughness and Temperature Corrections

The absolute roughness ε, in ft, given in *ASHRAE Handbook 1989, Fundamentals*, is listed in Table 17.7. When duct material differs from medium-smooth roughness, or when the air duct is installed at an elevation above 1600 ft (488 m), or when warm air is supplied at a temperature higher than 95°F (35°C), then corrections should be made to Δp_f as follows:

$$\Delta p_f = K_{sr} K_T K_{e1} \Delta p_{f,c} \tag{17.43}$$

where $\Delta p_{f,c}$ = friction loss found from the duct friction chart, in. WC (Pa). In Eq. (17.43), K_{sr} indicates the correction factor for surface roughness, which is dimensionless, and can be calculated as

$$K_{sr} = \frac{f_a}{f_c} \tag{17.44}$$

Here f_c denotes the friction factor of duct material of surface roughness that is specified by the duct friction chart, that is, $\varepsilon = 0.0005$ ft (0.09 mm). The symbol f_a represents the actual friction factor of duct material with surface roughness differing from f_c. Both f_a and f_c can be calculated from Eq. (17.41) or (17.42).

TABLE 17.7 Duct Roughness

Duct material (roughness, ft)	Roughness category	Absolute roughness ε, ft
Uncoated carbon steel, clean (0.00015 ft) PVC plastic pipe (0.00003 – 0.00015 ft) Aluminum (0.00015 – 0.0002 ft)	Smooth	0.0001
Galvanized steel, longitudinal seams, 4-ft joints (0.00016 – 0.00032 ft) Galvanized steel, spiral seams, with 1, 2, and 3 ribs, 12-ft joints (0.00018 – 0.00038 ft)	Medium smooth	0.0003
Galvanized steel, longitudinal seams, 2.5-ft joints (0.0005 ft)	Average	0.0005
Fibrous glass duct, rigid Fibrous glass duct liner, air side with facing material (0.0005 ft)	Medium rough	0.003
Fibrous glass duct liner, air side spray-coated (0.015 ft) Flexible duct, metallic (0.004 – 0.007 ft when fully extended) Flexible duct, all types of fabric and wire (0.0035 – 0.015 ft when fully extended) Concrete (0.001 – 0.01 ft)	Rough	0.01

Source: *ASHRAE Handbook 1989, Fundamentals*. Reprinted with permission.

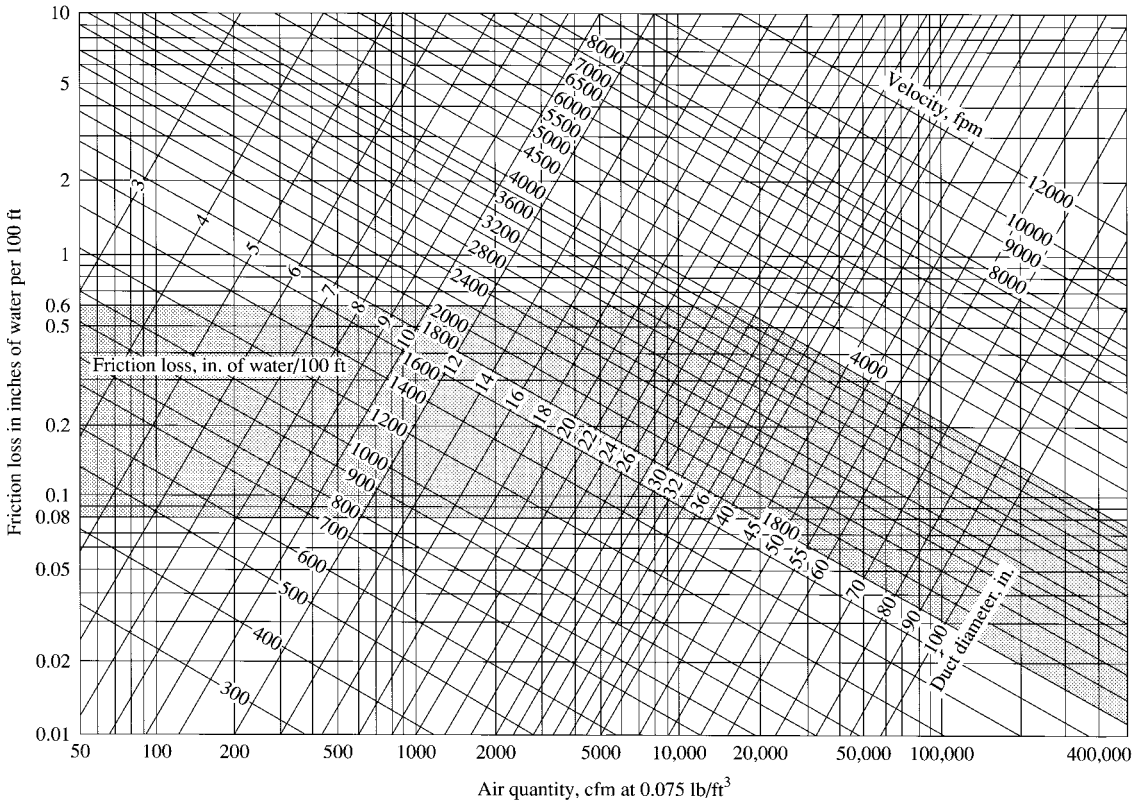


FIGURE 17.8 Friction chart for round ducts. (*Source: ASHRAE Handbook 1989. Reprinted with permission.*)

The term K_T indicates the correction factor for air temperature inside the duct, which affects the density of the air; K_T is dimensionless and can be calculated as

$$K_T = \left(\frac{530}{T_a + 460}\right)^{0.825} \tag{17.45}$$

where T_a = actual air temperature inside duct, °F. The term K_{el} indicates the correction factor for elevation, which also affects the density of the air; it is dimensionless. When the elevation is greater than 1600 ft (488 m), K_{el} can be evaluated as

$$K_{el} = \frac{p_{at}}{29.92} = \frac{p_{at,mm}}{760} \tag{17.46}$$

where p_{at} = actual atmospheric or barometric pressure, in. Hg
$p_{at,mm}$ = actual atmospheric or barometric pressure, mm Hg

Example 17.1. A fabric and wire flexible duct of 8-in. (200 mm) diameter installed in a commercial building at sea level has a surface roughness $\varepsilon = 0.12$ in. (3 mm). The mean air velocity inside the air duct is 800 fpm (4 m/s). Find the correction factor of surface roughness. The viscosity of air at 60°F is 1.21×10^{-5} lb/ft·s (2.0×10^{-5} Pa·s).

Solution. For the galvanized sheet duct specified in the duct friction chart, $\varepsilon = 0.0005$ ft, or 0.006 in. (0.15 mm). Then

$$\mathrm{Re}_D = \frac{\rho v D}{\mu} = \frac{0.075 \times 1000 \times 8}{60 \times 12 \times 1.21 \times 10^{-5}} = 5.5 \times 10^4$$

From Eq. (17.42),

$$f_c = \frac{0.25}{\{\log [\varepsilon/(3.7D) + 5.74/(0.9\,\mathrm{Re}_D)]\}^2}$$

$$= \frac{0.25}{\{\log[0.006/(3.7 \times 8) + 5.74/(0.9 \times 5.5 \times 10^4)]\}^2} = 0.0204$$

For the fabric and wire flexible duct, Re_D is the same as for the galvanized sheet duct. From the data given, $\varepsilon = 0.12$ in., so the actual friction factor is

$$f_a = \frac{0.25}{\{\log[0.12/(3.7 \times 8) + 5.74/(0.9 \times 5.5 \times 10^4)]\}^2} = 0.044$$

From Eq. (17.44), the correction factor of surface roughness is

$$K_{\mathrm{sr}} = \frac{f_a}{f_c} = \frac{0.044}{0.0204} = 2.16$$

Circular Equivalents

In Eq. (17.37), if D is replaced by D_h, then

$$\Delta p_f = f\left(\frac{L}{D_h}\right)\left(\frac{\rho v^2}{2g_c}\right)$$

Apparently, for circular or noncircular air ducts with different cross-sectional shapes and the same hydraulic diameter, the pressure loss is the same for an equal length of air duct at equal mean air velocities. Circular equivalents are used to convert the dimension of a noncircular duct into an equivalent diameter D_e, in in. (mm), of a round duct when their volume flow rates $\dot{V}$ and frictional losses per unit length $\Delta p_{f,u}$ are equal. A noncircular duct must be converted to a circular equivalent first before determining its $\Delta p_{f,u}$ from the duct friction chart. The hydraulic diameter D_h, in in. (mm), is defined as

$$D_h = \frac{4A}{P} \tag{17.47}$$

where A = area, in.^{2} (mm^2) and P = perimeter, in. (mm). Based on experimental results, Heubscher (1948) recommended the following formula to calculate D_e for rectangular duct at equal $\dot{V}$ and $\Delta p_{f,u}$:

$$D_e = \frac{1.30(ab)^{0.625}}{(a + b)^{0.25}} \tag{17.48}$$

where a, b = dimensions of two sides of rectangular duct, in. (mm). The circular equivalents of rectangular ducts at various dimensions calculated by Eq. (17.48) are listed in Table 17.8.

TABLE 17.8 Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity

One side, rectangular duct, in.	Adjacent side, in.																
	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0
3.0	3.8	4.0	4.2	4.4	4.6	4.7	4.9	5.1	5.2	5.5	5.7	6.0	6.2	6.4	6.6	6.8	7.0
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.5	7.7
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.4	6.7	7.0	7.3	7.6	7.8	8.0	8.3
4.5	4.6	4.9	5.2	5.4	5.7	5.9	6.1	6.3	6.5	6.9	7.2	7.5	7.8	8.1	8.4	8.6	8.8
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.3	7.6	8.0	8.3	8.6	8.9	9.1	9.4
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.6	8.0	8.4	8.7	9.0	9.3	9.6	9.9

One side, rectangular duct, in.	Adjacent side, in.																				Side rectangular duct
	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24	26	28	30	
6	6.6																				6
7	7.1	7.7																			7
8	7.6	8.2	8.7																		8
9	8.0	8.7	9.3	9.8																	9
10	8.4	9.1	9.8	10.4	10.9																10
11	8.8	9.5	10.2	10.9	11.5	12.0															11
12	9.1	9.9	10.7	11.3	12.0	12.6	13.1														12
13	9.5	10.3	11.1	11.8	12.4	13.1	13.7	14.2													13
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3												14
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4											15
16	10.4	11.3	12.2	13.0	13.7	14.4	15.1	15.7	16.4	16.9	17.5										16
17	10.7	11.6	12.5	13.4	14.1	14.9	15.6	16.2	16.8	17.4	18.0	18.6									17
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.7	17.3	17.9	18.5	19.1	19.7								18
19	11.2	12.2	13.2	14.1	14.9	15.7	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8							19
20	11.5	12.5	13.5	14.4	15.2	16.0	16.8	17.5	18.2	18.9	19.5	20.1	20.7	21.3	21.9						20
22	12.0	13.0	14.1	15.0	15.9	16.8	17.6	18.3	19.1	19.8	20.4	21.1	21.7	22.3	22.9	24.0					22
24	12.4	13.5	14.6	15.6	16.5	17.4	18.3	19.1	19.9	20.6	21.3	22.0	22.7	23.3	23.9	25.1	26.2				24
26	12.8	14.0	15.1	16.2	17.1	18.1	19.0	19.8	20.6	21.4	22.1	22.9	23.5	24.2	24.9	26.1	27.3	28.4			26
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.7	24.4	25.1	25.8	27.1	28.3	29.5	30.6		28
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.6	28.0	29.3	30.5	31.7	32.8	30
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.5	24.4	25.2	26.0	26.7	27.5	28.9	30.2	31.5	32.7	33.9	32
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5	28.3	29.7	31.0	32.4	33.7	34.9	34
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	22.9	23.9	24.8	25.7	26.6	27.4	28.2	29.0	30.5	32.0	33.3	34.6	35.9	36
38	15.0	16.5	17.8	19.0	20.2	21.4	22.4	23.5	24.5	25.4	26.4	27.2	28.1	28.9	29.8	31.3	32.8	34.2	35.6	36.8	38
40	15.3	16.8	18.2	19.5	20.7	21.8	22.9	24.0	25.0	26.0	27.0	27.9	28.8	29.6	30.5	32.1	33.6	35.1	36.4	37.8	40

42	15.6	17.1	18.5	19.9	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.3	31.2	32.8	34.4	35.9	37.3	38.7	42
44	15.9	17.5	18.9	20.3	21.5	22.7	23.9	25.0	26.1	27.1	28.1	29.1	30.0	30.9	31.8	33.5	35.1	36.7	38.1	39.5	44
46	16.2	17.8	19.3	20.6	21.9	23.2	24.4	25.5	26.6	27.7	28.7	29.7	30.6	31.6	32.5	34.2	35.9	37.4	38.9	40.4	46
48	16.5	18.1	19.6	21.0	22.3	23.6	24.8	26.0	27.1	28.2	29.2	30.2	31.2	32.2	33.1	34.9	36.6	38.2	39.7	41.2	48
50	16.8	18.4	19.9	21.4	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8	33.7	35.5	37.2	38.9	40.5	42.0	50
52	17.1	18.7	20.2	21.7	23.1	24.4	25.7	26.9	28.0	29.2	30.3	31.3	32.3	33.3	34.3	36.2	37.9	39.6	41.2	42.8	52
54	17.3	19.0	20.6	22.0	23.5	24.8	26.1	27.3	28.5	29.7	30.8	31.8	32.9	33.9	34.9	36.8	38.6	40.3	41.9	43.5	54
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.3	33.4	34.4	35.4	37.4	39.2	41.0	42.7	44.3	56
58	17.8	19.5	21.2	22.7	24.2	25.5	26.9	28.2	29.4	30.6	31.7	32.8	33.9	35.0	36.0	38.0	39.8	41.6	43.3	45.0	58
60	18.1	19.8	21.5	23.0	24.5	25.9	27.3	28.6	29.8	31.0	32.2	33.3	34.4	35.5	36.5	38.5	40.4	42.3	44.0	45.7	60
62		20.1	21.7	23.3	24.8	26.3	27.6	28.9	30.2	31.5	32.6	33.8	34.9	36.0	37.1	39.1	41.0	42.9	44.7	46.4	62
64		20.3	22.0	23.6	25.1	26.6	28.0	29.3	30.6	31.9	33.1	34.3	35.4	36.5	37.6	39.6	41.6	43.5	45.3	47.1	64
66		20.6	22.3	23.9	25.5	26.9	28.4	29.7	31.0	32.3	33.5	34.7	35.9	37.0	38.1	40.2	42.2	44.1	46.0	47.7	66
68		20.8	22.6	24.2	25.8	27.3	28.7	30.1	31.4	32.7	33.9	35.2	36.3	37.5	38.6	40.7	42.8	44.7	46.6	48.4	68
70		21.0	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.4	35.6	36.8	37.9	39.1	41.2	43.3	45.3	47.2	49.0	70
72			23.1	24.8	26.4	27.9	29.4	30.8	32.2	33.5	34.8	36.0	37.2	38.4	39.5	41.7	43.8	45.8	47.8	49.6	72
74			23.3	25.1	26.7	28.2	29.7	31.2	32.5	33.9	35.2	36.4	37.7	38.8	40.0	42.2	44.4	46.4	48.4	50.3	74
76			23.6	25.3	27.0	28.5	30.0	31.5	32.9	34.3	35.6	36.8	38.1	39.3	40.5	42.7	44.9	47.0	48.9	50.9	76
78			23.8	25.6	27.3	28.8	30.4	31.8	33.3	34.6	36.0	37.2	38.5	39.7	40.9	43.2	45.4	47.5	49.5	51.4	78
80			24.1	25.8	27.5	29.1	30.7	32.2	33.6	35.0	36.3	37.6	38.9	40.2	41.4	43.7	45.9	48.0	50.1	52.0	80
82				26.1	27.8	29.4	31.0	32.5	34.0	35.4	36.1	38.0	39.3	40.6	41.8	44.1	46.4	48.5	50.6	52.6	82
84				26.4	28.1	29.7	31.3	32.8	34.3	35.7	37.1	38.4	39.7	41.0	42.2	44.6	46.9	49.0	51.1	53.2	84
86				26.6	28.3	30.0	31.6	33.1	34.6	36.1	37.4	38.8	40.1	41.4	42.6	45.0	47.3	49.6	51.6	53.7	86
88				26.9	28.6	30.3	31.9	33.4	34.9	36.4	37.8	39.2	40.5	41.8	43.1	45.5	47.8	50.0	52.2	54.3	88
90				27.1	28.9	30.6	32.2	33.8	35.3	36.7	38.2	39.5	40.9	42.2	43.5	45.9	48.3	50.5	52.7	54.8	90
92					29.1	30.8	32.5	34.1	35.6	37.1	38.5	39.9	41.3	42.6	43.9	46.4	48.7	51.0	53.2	55.3	92
96					29.6	31.4	33.0	34.7	36.2	37.7	39.2	40.6	42.0	43.3	44.7	47.2	49.6	52.0	54.2	56.4	96

One side, rectangular duct, in.	Adjacent side, in.																				Side rectangular duct
	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	
32	35.0																				32
34	36.1	37.2																			34
36	37.1	38.2	39.4																		36
38	38.1	39.3	40.4	41.5																	38
40	39.0	40.3	41.5	42.6	43.7																40
42	40.0	41.2	42.5	43.7	44.8	45.9															42
44	40.9	42.2	43.5	44.7	45.8	47.0	48.1														44
46	41.8	43.1	44.4	45.7	46.9	48.0	49.2	50.3													46
48	42.6	44.0	45.3	46.6	47.9	46.1	50.2	51.4	52.5												48
50	43.6	44.9	46.2	47.5	48.8	50.0	51.2	52.4	53.6	54.7											50

TABLE 17.8 (*Continued*)

One side, rectangular duct, in.	Adjacent side, in.																				Side rectangular duct
	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	
52	44.3	45.7	47.1	48.4	49.7	51.0	52.2	53.4	54.6	55.7	56.8										52
54	45.1	46.5	48.0	49.3	50.7	52.0	53.2	54.4	55.6	56.8	57.9										54
56	45.8	47.3	48.8	50.2	51.6	52.9	54.2	55.4	56.6	57.8	59.0	61.2									56
58	46.6	48.1	49.6	51.0	52.4	53.8	55.1	56.4	57.6	58.8	60.0	62.3									58
60	47.3	48.9	50.4	51.9	53.3	54.7	60.0	57.3	58.6	59.8	61.0	63.4	65.6								60
62	48.0	49.6	51.2	52.7	54.1	55.5	56.9	58.2	59.5	60.8	62.0	64.4	66.7								62
64	48.7	50.4	51.9	53.5	54.9	56.4	57.8	59.1	60.4	61.7	63.0	65.4	67.7	70.0							64
66	49.4	51.1	52.7	54.2	55.7	57.2	58.6	60.0	61.3	62.6	63.9	66.4	68.8	71.0							66
68	50.1	51.8	53.4	55.0	56.6	58.0	59.4	60.8	62.2	63.6	64.9	67.4	69.8	72.1	74.3						68
70	50.8	52.5	54.1	55.7	57.3	58.8	60.3	61.7	63.1	64.4	65.8	68.3	70.8	73.2	75.4						70
72	51.4	53.2	54.8	56.6	58.0	59.6	61.1	62.5	63.9	65.3	66.7	69.3	71.8	74.2	76.5	78.7					72
74	52.1	53.8	55.5	57.2	58.8	60.3	61.9	63.3	64.8	66.2	67.5	70.2	72.7	75.2	77.5	79.8					74
76	52.7	54.5	56.2	57.9	59.5	61.1	62.6	64.1	65.6	67.0	68.4	71.1	73.7	76.2	78.6	80.9	83.1				76
78	53.3	55.1	56.9	58.6	60.2	61.8	63.4	64.9	66.4	67.9	69.3	72.0	74.6	77.1	79.6	81.9	84.2				78
80	53.9	55.8	57.5	59.3	60.9	62.6	64.1	65.7	67.2	68.7	70.1	72.9	75.4	78.1	80.6	82.9	85.2	87.5			80
82	54.5	56.4	58.2	59.9	61.6	63.3	64.9	66.5	68.0	69.5	70.9	73.7	76.4	79.0	81.5	84.0	86.3	88.5			82
84	55.1	57.0	58.8	60.6	62.3	64.0	65.6	67.2	68.7	70.3	71.7	74.6	77.3	80.0	82.5	85.0	87.3	89.6	91.8		84
86	55.7	57.6	59.4	61.2	63.0	64.7	66.3	67.9	69.5	71.0	72.5	75.4	78.2	80.9	83.5	85.9	88.3	90.7	92.9		86
88	56.3	58.2	60.1	61.9	63.6	65.4	67.0	68.7	70.2	71.8	73.3	76.3	79.1	81.8	84.4	86.9	89.3	91.7	94.0	96.2	88
90	56.8	58.8	60.7	62.5	64.3	66.0	67.7	69.4	71.0	72.6	74.1	77.1	79.9	82.7	85.3	87.9	90.3	92.7	95.0	97.3	90
92	57.4	59.3	61.3	63.1	64.9	66.7	68.4	70.1	71.7	73.3	74.9	77.9	80.8	83.5	86.2	88.8	91.3	93.7	96.1	98.4	92
94	57.9	59.9	61.9	63.7	65.6	67.4	69.1	70.8	72.4	74.0	75.6	78.7	81.6	84.4	87.1	89.7	92.3	94.7	97.1	99.4	94
96	58.4	60.5	62.4	64.3	66.2	68.0	69.7	71.5	73.1	74.8	76.3	79.4	82.4	85.3	88.0	90.7	93.2	95.7	98.1	100.5	96

Source: *ASHRAE Handbook 1989, Fundamentals*. Reprinted with permission.

For galvanized steel flat oval ducts with spiral seams, Heyt and Diaz (1975) proposed the following formula to calculate the circular equivalent for use of the duct friction chart:

$$D_e = \frac{1.55A^{0.625}}{P^{0.25}} \tag{17.49}$$

Here A is the cross-sectional area of the flat oval duct, in.^{2} (mm^2), and is given as

$$A = \frac{\pi b^2}{4} + b(a - b) \tag{17.50}$$

and the perimeter P, in in. (mm), is calculated as

$$P = \pi b + 2(a + b) \tag{17.51}$$

The dimensions a and b of the flat oval duct are shown in Fig. 17.4*c*.

17.6 DYNAMIC LOSSES

When air flows through duct fittings, such as, elbows, tees, diffusers, contractions, entrances and exits, or certain equipment, a change in velocity or direction of flow may occur. Such a change leads to flow separation and the formation of eddies and disturbances in that area. The energy loss resulting from these eddies and disturbances is called *dynamic loss* Δp_{dy}, in in. WC (Pa). Although a duct fitting is fairly short, the disturbances it produces may persist over a considerable distance downstream.

In addition to the presence of dynamic loss Δp_{dy}, frictional loss Δp_f occurs when an airstream flows through a duct fitting. For convenience in calculation, the length of the duct fitting is usually added to the adjacent duct sections connected with this duct fitting of the same mean air velocity.

When airstreams of the same Reynolds number flow through geometrically similar duct fittings, that is, in dynamic similarity, the dynamic loss is proportional to their velocity pressure p_v. Dynamic loss may be calculated as

$$\Delta p_{\text{dy}} = C_o p_v = \frac{C_o \rho v_o^2}{2g_c \text{cf}} = C_o \left(\frac{v_o}{4005} \right)^2 \tag{17.52}$$

where C_o = local loss coefficient or dynamic loss coefficient
ρ = air density, lb/ft^3 (kg/m^3)
v_o = mean air velocity of airstream at reference cross section o, fpm (m/s)
g_c = dimensional constant, 32.2 lb$_m$ · ft/lb$_f$ · s^2, for SI units, $g_c = 1$
cf = conversion factor, for SI units, cf = 1

Because the mean velocity of the airstream may vary at different ends of a duct fitting, C_o is always specified with respect to a velocity of a reference cross section o in the duct fitting.

Elbows

An elbow is a duct fitting in which the airflow changes direction. Elbows are shown in Figs. 17.9 and 17.10. Consider an elbow that makes a 90° turn in a round duct, as shown in Fig. 17.9*a*. Because of the change of airstream direction, centrifugal force is created and acts toward the outer wall of the duct. When the airstream flows from the straight part of the duct to the curved part, it is accompanied by an increase in pressure and a decrease in air velocity at the outer wall. At the same time, a decrease in pressure and an increase in air velocity take place at the inner wall. Therefore, a diffuser effect occurs near the outer wall, and a bell mouth forms near the inner wall. After turning, the opposite effect takes place as the airstream flows from the curved part to the straight part of the

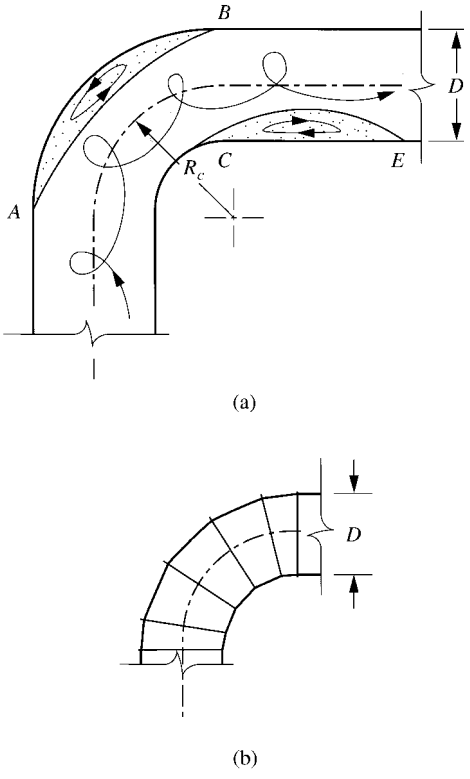


FIGURE 17.9 Round elbows: (*a*) region of eddies and turbulences in a round elbow; (*b*) five-piece 90° round elbow.

duct. The diffuser effects lead to flow separations from both walls, and eddies and large-scale turbulence form in regions *AB* and *CE*, as shown in Fig. 17.9*a*.

In a rectangular elbow, a radial pressure gradient is also formed by the centrifugal force along its centerline *NR*, as shown in Fig. 17.10*c*. A secondary circulation is formed along with the main forward airstream.

The magnitude of the local loss coefficient of an elbow is influenced by the following factors:

- Turning angle of the elbow
- Ratio of centerline radius (CLR) to diameter $= R_c/D$ or R_c/W, where R_c represents the throat radius and W the width of the duct, both in in. (mm), as shown in Fig. 17.9*a*
- A three-gore (number of pieces), five-gore, or pleated seven-gore 90° elbow (Fig. 17.9*b*)
- Installation of splitter vanes, which reduce the eddies and turbulence in an elbow
- Shape of cross-sectional area of the duct

As the CLR becomes greater, the flow resistance of the airstream becomes smaller. However, a greater CLR requires more duct material, a higher labor cost, and a larger allocated space. A value of R_c/D (CLR) $= 1$ or 1.5 is often used if the space is available.

The installation of splitter vanes, or turning vanes, in rectangular ducts can effectively reduce the pressure loss at the elbow. For a rectangular 90° elbow with $R_c/W = 0.75$ and two splitter

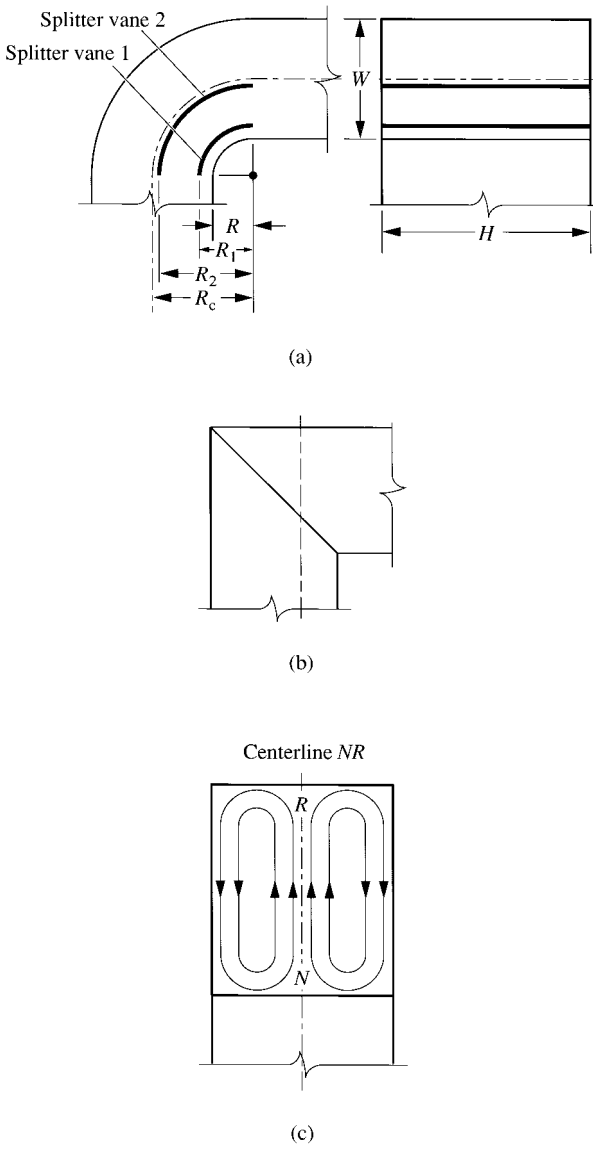


FIGURE 17.10 Rectangular elbows: (*a*) rectangular elbow, smooth radius, two splitter vanes; (*b*) mitered elbow; (*c*) secondary flow in a mitered elbow.

vanes, as shown in Fig. 17.10*a*, the local loss coefficient C_o is only 0.04 to 0.05. The values of R_o, R_1, and R_2 can be found from Table 17.11. For details, refer to *ASHRAE Handbook 1997*, *Fundamentals*.

For a mitered rectangular 90° elbow without turning vanes, C_o is about 1.1. If turning vanes are installed, C_o drops to only 0.12 to 0.18. Installation of splitter vanes in mitered elbows is also

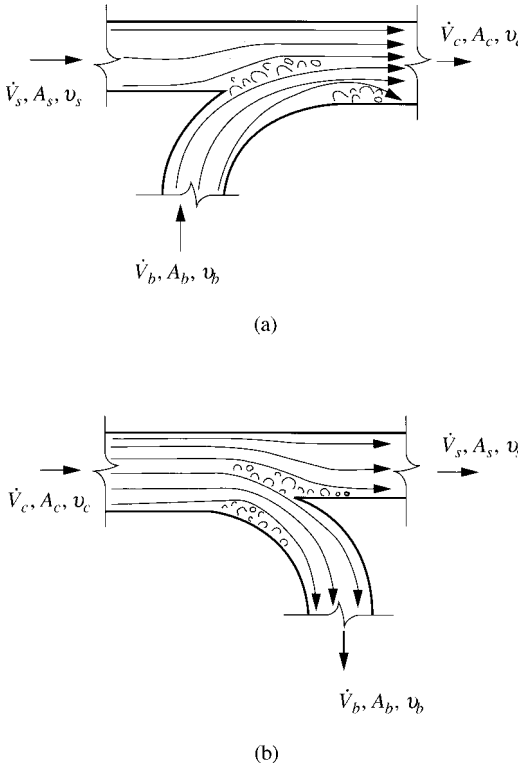


FIGURE 17.11 Converging and diverging wyes: (*a*) converging wye, rectangular, $\theta = 90°$; (*b*) diverging wye, rectangular, $\theta = 90°$.

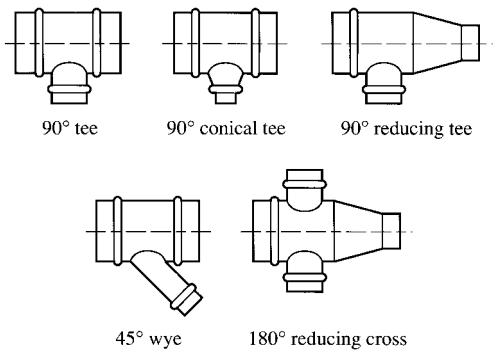
advantageous for noise control because of the smaller fan total pressure and, therefore, lower fan noise as well as lower airflow noise at the elbow.

Converging and Diverging Tees and Wyes

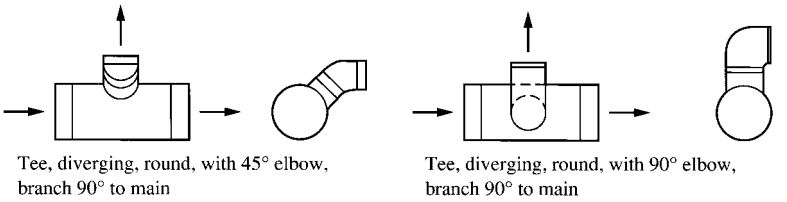
A branch duct that combines with or diverges from the main duct at an angle of 90° is called a tee. However, if the angle lies between 15° and 75°, it is called a wye (Figs. 17.11 and 17.12). Tees and wyes can be round, flat oval, or rectangular. Various types of converging and diverging tees and wyes for round and flat oval ducts appear in Fig. 17.12.

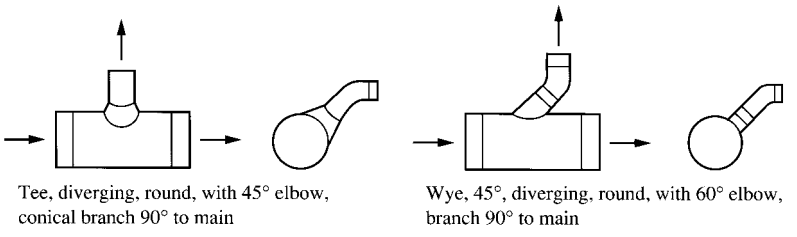
The function of a converging tee or wye is to combine the airstream from the branch duct with the airstream from the main duct. The function of a diverging tee or wye is to diverge part of the airflow from the main duct into the branch takeoff. For airstreams flowing through a converging or diverging tee or wye, the dynamic losses for the main stream can be calculated as

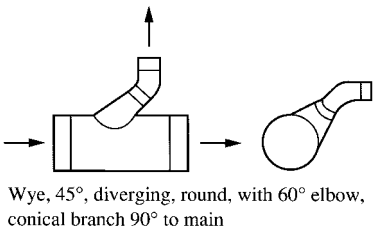
$$\Delta p_{c,s} = \frac{C_{c,s}\rho v_c^2}{2g_c\text{cf}} = C_{c,s}\left(\frac{v_c}{4005}\right)^2$$
$$\Delta p_{s,c} = \frac{C_{s,c}\rho v_c^2}{2g_c\text{cf}} = C_{s,c}\left(\frac{v_c}{4005}\right)^2 \tag{17.53}$$



(a)







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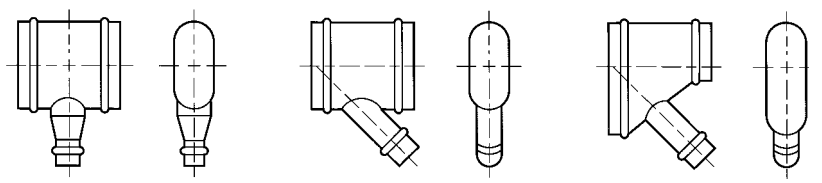


(c)

FIGURE 17.12 Round and flat oval tees and wyes: (*a*) round tees, wyes, and cross; (*b*) diverging tees and wyes with elbows; (*c*) flat oval tees and wyes.

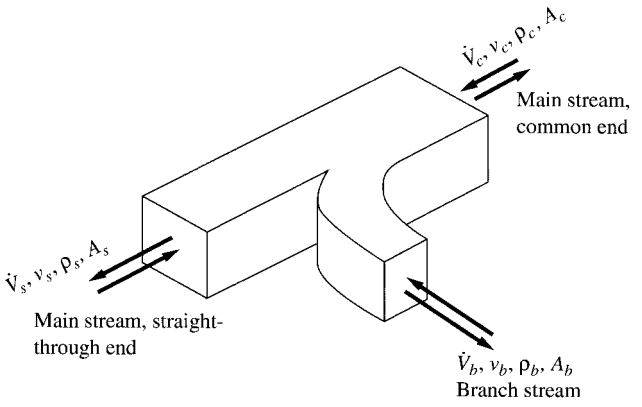


FIGURE 17.13 Airflows through a rectangular converging or diverging wye.

For the branch stream,

$$\Delta p_{c,b} = \frac{C_{c,b}\rho v_c^2}{2g_c\text{cf}} = C_{c,b}\left(\frac{v_c}{4005}\right)^2$$
$$\Delta p_{b,c} = \frac{C_{b,c}\rho v_c^2}{2g_c\text{cf}} = C_{b,c}\left(\frac{v_c}{4005}\right)^2 \tag{17.54}$$

In Eqs. (17.53) and (17.54), subscript *c* represents the common end, *s* the straight-through end, and *b* the branch takeoff. The subscript *c,s* indicates the flow of main stream from the common end to the straight-through end, and *s,c* the flow from the straight-through end to the common end. Similarly, *c,b* denotes the flow of the branch stream from the common end to the branch takeoff, and *b,c* the airflow from the branch duct to the common end. The airstreams flowing through a rectangular converging or diverging tee are shown in Fig. 17.13.

As mentioned in Sec. 17.2, the total pressure of the main stream may increase when it flows through a converging or diverging wye or tee. This occurs because energy is received from the airstream of higher velocity or the diverging of the slowly moving boundary layer into branch takeoff from the main airstream. However, the sum of the energies of the main and branch streams leaving the duct fitting is always smaller than that entering the duct fitting because of the energy losses.

The magnitude of local loss coefficients $C_{c,s}$, $C_{s,c}$, $C_{b,c}$, and $C_{c,b}$ is affected by the shape and construction of the tee or wye, the velocity ratios v_s/v_c and v_b/v_c, the volume flow ratios $\dot{V}_s/\dot{V}_c$ and $\dot{V}_b/\dot{V}_c$, and area ratios A_s/A_c and A_b/A_c.

For rectangular ducts, the converging or diverging wye of the configuration shown in Fig. 17.13 gives the lower C_o and, therefore, less energy loss. For round ducts, the C_o values for various types of diverging tees and wyes with nearly the same outlet direction and the same velocity ratio $v_b/v_c = 0.6$, as shown in Fig. 17.12*b*, are quite different.

Tee, diverging, round, with 45° elbow, branch 90° to main	$C_o = 1.60$
Tee, diverging, round, with 90° elbow, branch 90° to main	$C_o = 1.18$
Tee, diverging, round, with 45° elbow, conical branch 90° to main	$C_o = 0.84$

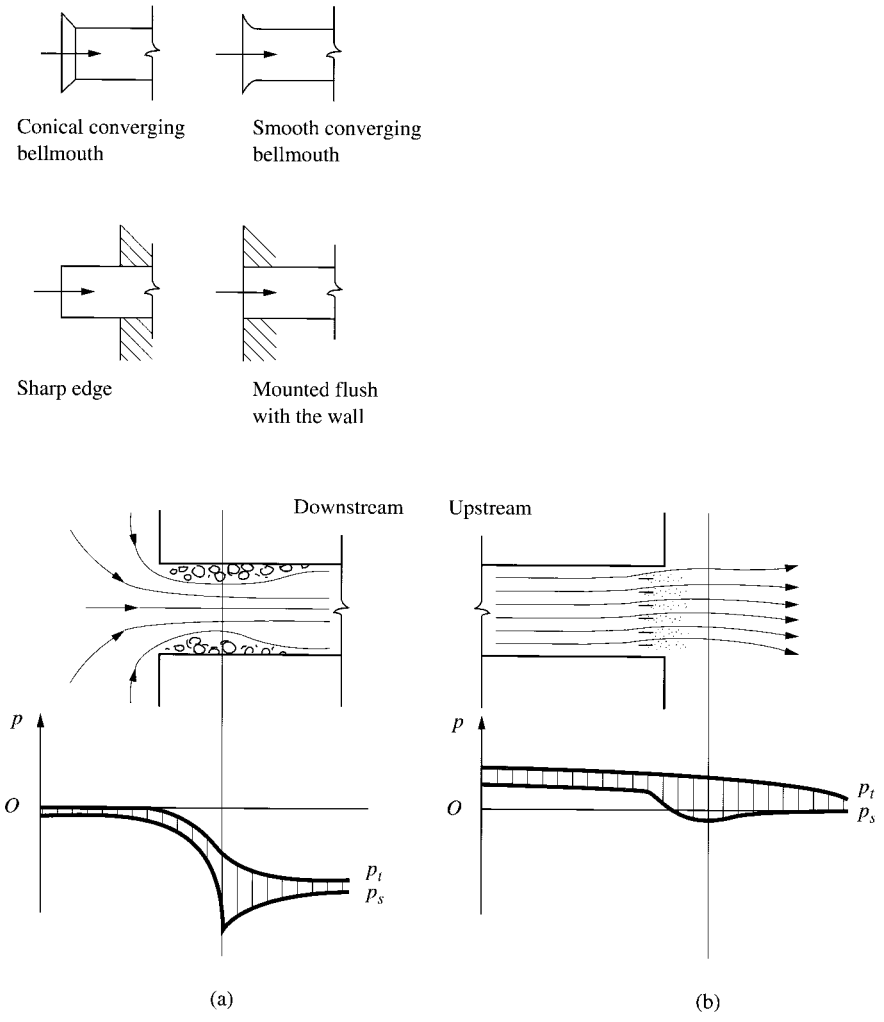


FIGURE 17.14 Openings mounted on a duct or a duct wall: (*a*) entrances; (*b*) exits.

Wye, 45°, diverging, round, with 60° elbow, branch 90° to main	$C_o = 0.68$
Wye, 45°, diverging, round, with 60° elbow, conical branch 90° to main	$C_o = 0.52$

The different C_o values help in choosing the optimum duct fitting and balancing the total pressure between different paths of airflow in an air duct system.

In Table 17.11 are listed the local loss coefficients of diverging tees and wyes. For details, refer to *ASHRAE Handbook Fundamentals* (1997) and Idelchik (1986).

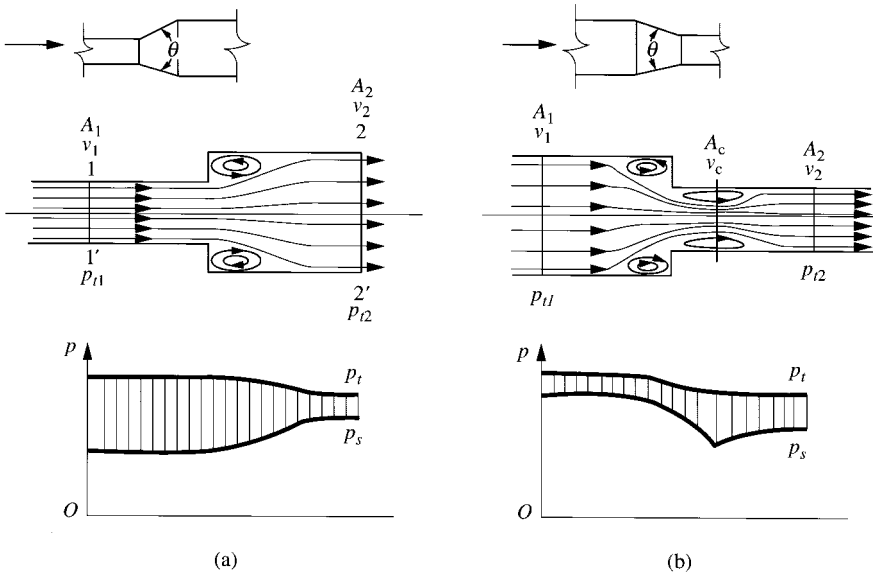


FIGURE 17.15 Enlargements and contractions: (*a*) abrupt enlargement; (*b*) sudden contraction.

Entrances, Exits, Enlargements, and Contractions

Entrances and exits are the end openings mounted on a duct or a duct wall, as shown in Fig. 17.14. Because of the change in direction of the streamlines at the entrance, eddies and large-scale turbulences develop along the duct wall when the airstream passes through the entrance. Generally, the total pressure drop Δp_t of the airstream before it enters the entrance is negligible. A sharp-edge entrance may have a $C_o = 0.9$, whereas for an entrance flush-mounted with the wall, C_o reduces to 0.5. An entrance with a conical converging bellmouth may further reduce C_o to 0.4. If an entrance with a smooth converging bell mouth is installed, C_o could be as low as 0.1.

When the air flows through a wall outlet or an exit, flow separation occurs along the surface of the vanes, and wakes are formed downstream, so there is a drop in total pressure. The velocity of the airstream reaches its maximum value at the vena contracta, where the cross section of airflow is minimum and the static pressure is negative. The total pressure loss at the outlet always includes the velocity pressure of the discharge airstream.

Various types of return inlets, such as grilles and louvers, and supply outlets, such as diffusers, are discussed in Chapter 18.

When air flows through an enlargement or a contraction, flow separation occurs and produces eddies and large-scale turbulences after the enlargement, or before and after the contraction. Both cause a total pressure loss Δp_t, as shown in Fig. 17.15. To reduce the energy loss, a gradual expansion, often called a diffuser, or converging transition is preferred. An expansion with an including angle of enlargement $\theta = 14°$, as shown in Fig. 17.15*a*, is ideal. In actual practice, θ may be from 14° to 45° because of limited space. For converging transitions, an including angle of 30° to 60° is usually used.

17.7 FLOW RESISTANCE

Flow resistance is a property of fluid flow that measures the characteristics of a flow passage which resist the fluid flow in that passage with a corresponding total pressure loss at a specific volume

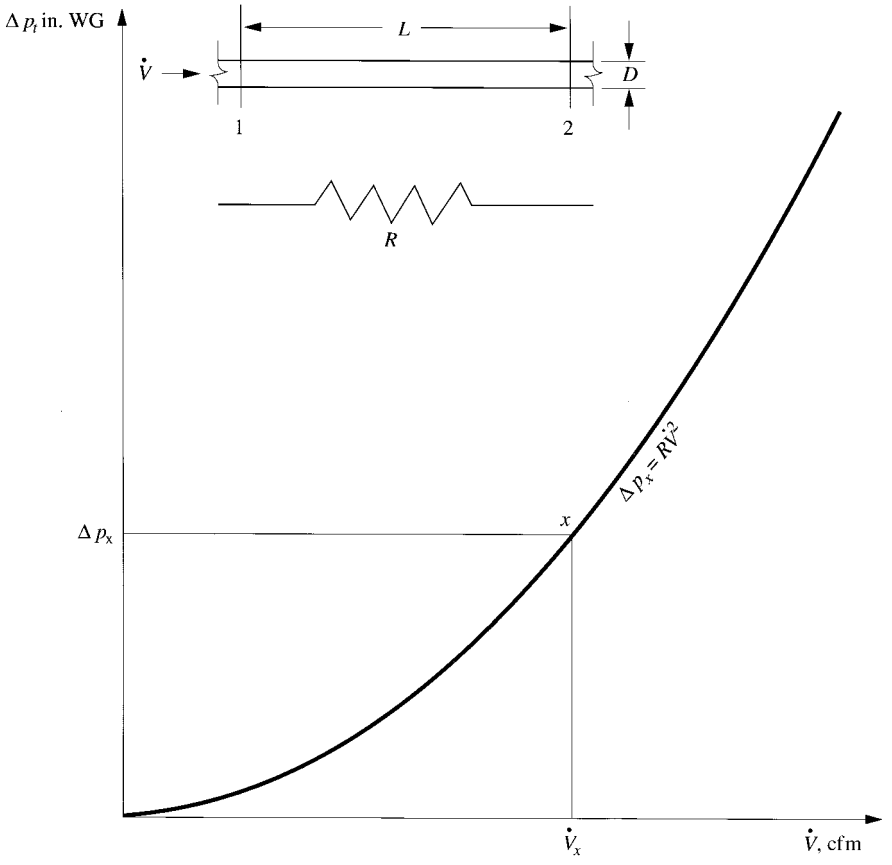


FIGURE 17.16 Total pressure loss Δp_t and flow resistance R of a round duct section.

flow rate. Consider a round duct section (Fig. 17.16), between two cross-sectional planes 1 and 2, with a diameter D, in ft (mm), and length L, in ft (m). The fluid flow has a velocity of v, in fpm (m/s), and a volume flow rate of $\dot{V}$, in cfm (m^3/s). From Eq. (17.37), the total pressure loss of the fluid flow Δp_t, in in. WC (Pa), between planes 1 and 2 can be calculated as

$$\Delta p_t = C_v f\left(\frac{L}{D}\right)\left(\frac{\rho v^2}{2g_c}\right) = \frac{8C_v f L\rho}{g_c \pi^2 D^5}\dot{V}^2 \tag{17.55}$$

where f = friction factor
ρ = density of fluid, lb/ft^3 (kg/m^3)
g_c = dimensional constant, 32.2 $lb_m \cdot ft/lb_f \cdot s^2$ (for SI unit, $g_c = 1$)
C_v = conversion constant

Let $R = 8C_v fL\rho/(g_c\,\pi^2 D^5)$, so that total pressure loss becomes

$$\Delta p_t = R\dot{V}^2$$

$$R = \frac{\Delta p_t}{\dot{V}^2} \tag{17.56}$$

In Eq. (17.56), R is defined as flow resistance, which indicates the resistance to fluid flow of this duct section, and is characterized by its specific total pressure loss and volume flow rate. Flow resistance in I-P units is expressed as in. WC/(cfm)^2 (in SI units, Pa·s^2/m^6).

For a given duct section, D and L are constants. In addition, the difference in the mean values of f and ρ at different volume flow rates is small, so R can be considered a constant. The relationship between Δp_t and $\dot{V}$ of this section can be represented by a parabola whose vertex coincides with the origin.

Differentiating Eq. (17.56) with respect to $\dot{V}$, then, gives

$$\frac{d\Delta p_t}{d\dot{V}} = 2R\dot{V} \tag{17.57}$$

This means that the slope of any segment of the curve $\Delta p_t = R\dot{V}^2$ is $2R\dot{V}$. Consequently, the greater the flow resistance R, the steeper the slope of the characteristic curve of this duct section.

Because the flow resistance R of a specific duct section is related to $\dot{V}$ by means of Eq. (17.56), the total pressure loss of the fluid flow in this duct section at any $\dot{V}$ can be calculated. Therefore, the total pressure loss Δp_t curve of this duct section can be plotted at different $\dot{V}$ values, and forms the Δp_t-$\dot{V}$ diagram. On the Δp_t-$\dot{V}$ diagram, each characteristic curve represents a unique value of flow resistance R. If a duct section contains additional dynamic losses $C_{\text{dy}}\rho v^2/(2g_c)$, as mentioned in Sec. 17.6, the relationship between Δp_t, R, and $\dot{V}$ defined by Eq. (17.56) still holds, but the flow resistance R is changed to

$$R = \frac{8\rho(C_v fL/D + C_{\text{dy}})}{g_c \pi^2 D^4} \tag{17.58}$$

where C_{dy} = local loss coefficient.

Flow Resistances Connected in Series

Consider a duct system that consists of several duct sections connected in series with air flowing inside the duct, as shown in Fig. 17.17*a*. When these duct sections are connected in series, the volume flow rate of air that flows through each section must be the same. The total pressure loss between planes 1 and 4 is the sum of the total pressure losses of each duct section, i.e.,

$$\begin{aligned} \Delta p_t &= \Delta p_{12} + \Delta p_{23} + \cdots + \Delta p_n \\ &= (R_{12} + R_{23} + \cdots + R_n)\dot{V}^2 = R_s\dot{V}^2 \end{aligned} \tag{17.59}$$

where $R_{12}, R_{23}, \ldots, R_n$ = flow resistances of duct sections 1, 2, . . . , n, in. WC/(cfm)^2 (Pa·s^2/m^6)
$\Delta p_{12}, \Delta p_{23}, \ldots, \Delta p_n$ = total pressure losses across flow resistances $R_{12}, R_{23}, \ldots, R_n$, in. WC (Pa)
$\dot{V}_{12}, \dot{V}_{23}, \ldots, \dot{V}_n$ = volume flow rates of air flowing through resistances $R_{12}, R_{23}, \ldots, R_n$, cfm (m^3/s)

The flow resistance of the duct system R_s is equal to the sum of individual flow resistances of duct sections connected in series, or

$$R_s = R_{12} + R_{23} + \cdots + R_n \tag{17.60}$$

If the characteristic curves of each individual duct section with flow resistances of R_{12}, $R_{23}, \ldots, R_n$ are plotted on a Δp_t-$\dot{V}$ diagram, then the characteristic curve of the entire duct can be plotted either by using the relationship $R_s = \Delta p_t/\dot{V}^2$ or by graphical method. Plotting by graphical method is done by drawing a constant $\dot{V}$ line. The points of Δp_t on the characteristic curve of the duct can be found by summing the individual total pressure losses $\Delta p_{12}, \Delta p_{23}, \ldots, \Delta p_n$ of individual duct sections 1, 2, . . . , n, as shown in Fig. 17.17*a*.

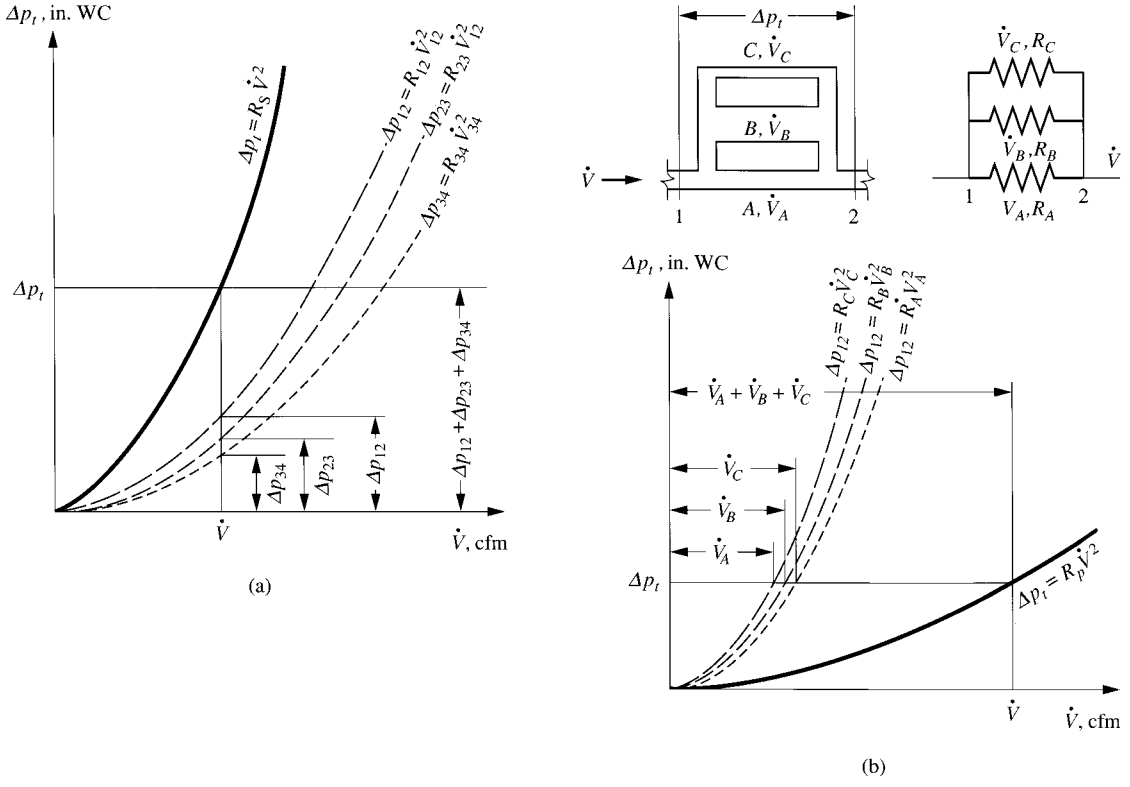


FIGURE 17.17 Combination of flow resistances: (*a*) connected in series; (*b*) connected in parallel.

Flow Resistances Connected in Parallel

When a duct system has several sections connected in parallel, as shown in Fig. 17.17*b*, the total volume flow rate of this duct $\dot{V}$ that flows through cross-sectional planes 1 and 2 is the sum of the individual volume flow rates $\dot{V}_A, \dot{V}_B, \ldots, \dot{V}_n$ of each duct section, i.e.,

$$\begin{aligned} \dot{V} &= \dot{V}_A + \dot{V}_B + \cdots + \dot{V}_n \\ &= \sqrt{\frac{\Delta p_{12}}{R_A}} + \sqrt{\frac{\Delta p_{12}}{R_B}} + \cdots + \sqrt{\frac{\Delta p_{12}}{R_n}} \end{aligned} \tag{17.61}$$

where Δp_{12} = total pressure loss between planes 1 and 2, in. WC (Pa)
$R_A, R_B, \ldots, R_n$ = flow resistance of duct sections $A, B, \ldots, n$, in. WC/(cfm)^2

As for the entire duct, $\dot{V} = \sqrt{\Delta p_{12}/R_p}$. Here R_p represents the flow resistance of the entire duct whose sections are in parallel connection. Then

$$\frac{1}{\sqrt{R_p}} = \frac{1}{\sqrt{R_A}} + \frac{1}{\sqrt{R_B}} + \cdots + \frac{1}{\sqrt{R_n}} \tag{17.62}$$

If duct sections are connected in parallel, the total pressure loss across the common junctions 1 and 2 must be the same, or

$$\Delta p_t = \Delta p_{12} \tag{17.63}$$

For two sections with flow resistances R_A and R_B connected in parallel, the flow resistance of the combination is then

$$R_p = \frac{R_A R_B}{R_A + R_B + 2\sqrt{R_A R_B}} \tag{17.64}$$

As in series connection, the characteristic curve of the duct with segments connected in parallel can be plotted on the Δp_t-$\dot{V}$ diagram either by the relationship $\Delta p_t = R_p \dot{V}^2$ or by graphical method. In the graphical method, draw a constant-Δp_t line; the total volume flow rate on the characteristic curve of the duct can be found by adding the volume flow rates of the individual sections, as shown in Fig. 17.17*b*.

Flow Resistance of a Y Connection

When a round duct consists of a main duct section and two branches, as shown in Fig. 17.18, a Y-connection flow circuit is formed. In a Y-connection flow circuit, the volume flow of the main duct section 01, denoted by $\dot{V}_{01}$, is the sum of the volume flow rates at branches 11′ and 12′, that is, $\dot{V}_{01} = \dot{V}_{11'} + \dot{V}_{12'}$. If the ambient air that surrounds the duct is at atmospheric pressure, then $\Delta p_{11'} = \Delta p_{12'}$ and $\Delta p_{01'} = \Delta p_{02'}$.

Because $\Delta p_{11'} = R_{11'} \dot{V}_{11'}^2$, $= \Delta p_{12'} = R_{12'} \dot{V}_{12'}^2$,

$$\begin{aligned} \dot{V}_{11'} &= \frac{1}{1 + \sqrt{R_{11'}/R_{12'}}} \dot{V}_{01} = K_{11'} \dot{V}_{01} \\ \dot{V}_{12'} &= \frac{1}{1 + \sqrt{R_{12'}/R_{11'}}} \dot{V}_{01} = K_{12'} \dot{V}_{01} \end{aligned} \tag{17.65}$$

Therefore, total pressure loss at ductwork 01′ and 02′ is calculated as

$$\begin{aligned} \Delta p_{01'} &= \Delta p_{01} + \Delta p_{11'} \\ &= R_{01} \dot{V}_{01}^2 + R_{11'} \dot{V}_{11'}^2 = (R_{01} + R_{11'} K_{11'}^2) \dot{V}_{01}^2 \\ &= R_{01'} \dot{V}_{01}^2 \end{aligned} \tag{17.66}$$

Similarly,

$$\Delta p_{02'} = R_{02'} \dot{V}_{01}^2 \tag{17.67}$$

For a given duct, C_{dy}, L_{01}, $L_{11'}$, $L_{12'}$, D_{01}, $D_{11'}$, and $D_{12'}$ are constants. Air density ρ can be considered a constant. Although the friction factor f is a function of air velocity v and the volume flow rate $\dot{V}$, at high Reynolds numbers f is least affected and can also be taken as a constant. Therefore, $R_{01'}$ and $R_{02'}$ can be considered constants.

Figure 17.18 also shows the characteristic curve of a Y connection plotted on a Δp_t-$\dot{V}$ diagram.

Flow Resistance of a Duct System

Supply and return duct systems are composed of Y connections. If the ambient air is at atmospheric pressure, the total pressure loss of a supply duct system and its flow characteristics, as shown in

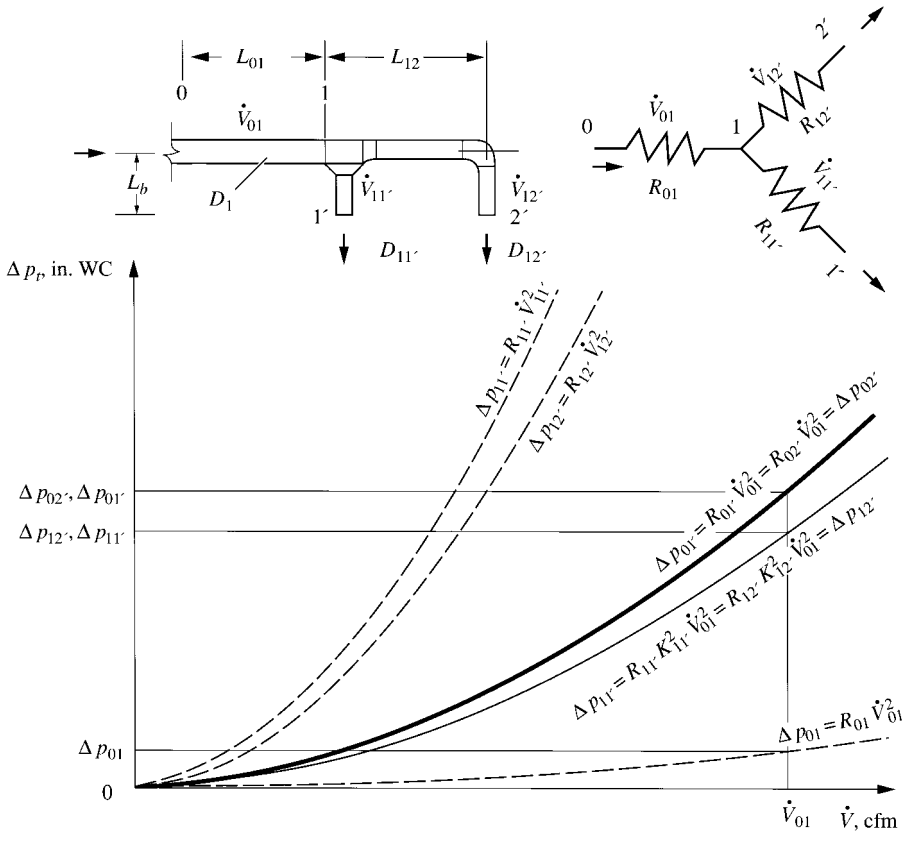


FIGURE 17.18 Flow resistance for a Y connection.

Fig. 17.19*a*, are given as

$$\begin{aligned}\Delta p_{0n'} &= \Delta p_{01} + \Delta p_{12} + \cdots + \Delta p_{(n-1)n'} \\ &= R_{01}\dot{V}_{01}^2 + R_{12}K_{12}^2\dot{V}_{01}^2 + \cdots + R_{(n-1)n'}K_{(n-1)n'}\dot{V}_{01}^2 \\ &= R_{0n'}\dot{V}_{01}^2 \end{aligned} \tag{17.68}$$

and

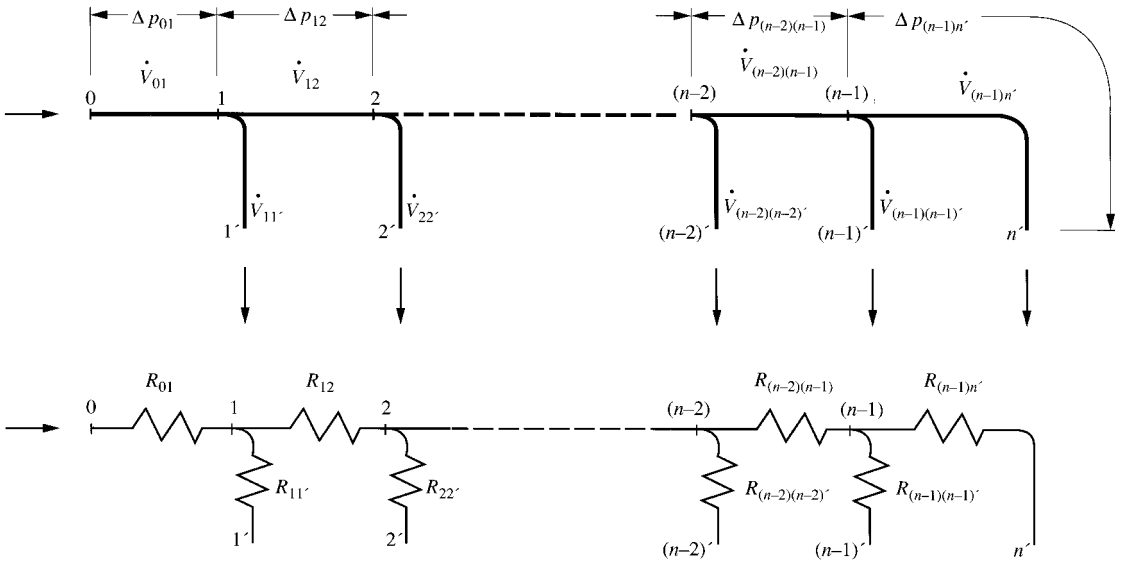
$$R_{0n'} = R_{01} + R_{12}K_{12}^2 + \cdots + R_{(n-1)n'}K_{(n-1)n'} \tag{17.69}$$

The characteristic curve of a duct system on a Δp_t-$\dot{V}$ diagram is called the system curve. It is shown in Fig. 17.19*b*.

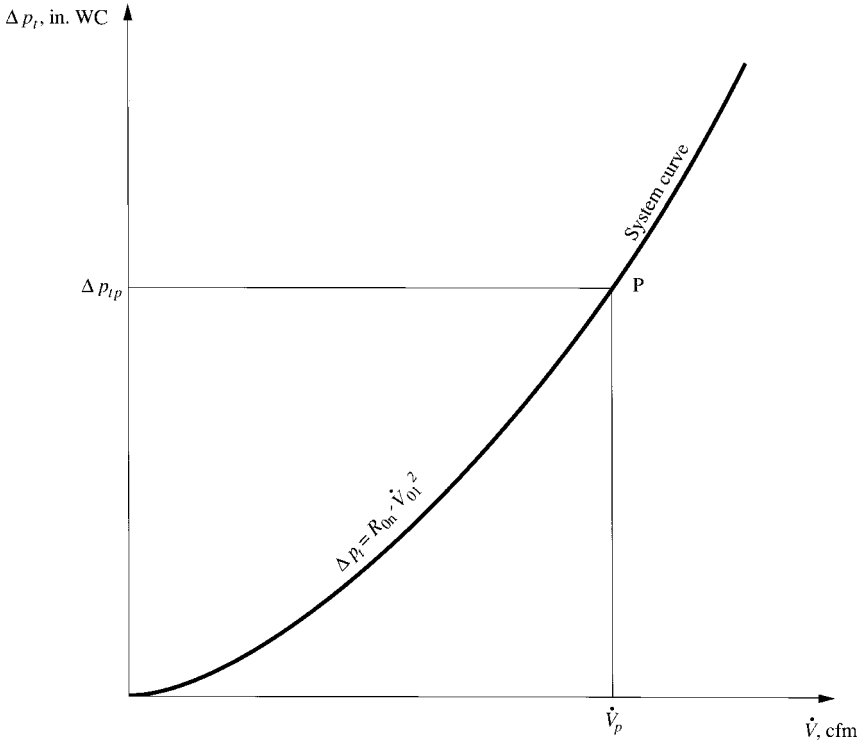
17.8 PRINCIPLES AND CONSIDERATIONS IN AIR DUCT DESIGN

Optimal Air Duct Design

An optimal air duct system transports the required amount of conditioned, recirculated, or exhaust air to the specific space and meets the following requirements:



(a)



(b)

FIGURE 17.19 Flow characteristics of a supply duct system: (*a*) schematic diagram; (*b*) system curve.

- An optimal duct system layout within the allocated space
- A satisfactory system balance, achieved through the pressure balance of various paths by changing duct sizes or using different configurations of duct fittings
- Space sound level lower than the allowable limits
- Optimum energy loss and initial cost
- Installation with only necessary balancing devices such as air dampers and nozzle plates
- National, ASHRAE, and local codes of fire protection, duct construction, and duct insulation met

These requirements result in the design of optimal duct layout, duct size, and total pressure loss of the duct system.

Air duct system design often requires comprehensive analysis and computer-aided calculating programs. Different air duct systems have different transport functions and thus have their own characteristics. It is difficult to combine the influences of cost, system balance, and noise together with duct characteristics into one or two representative indices.

Design Velocity

For any air duct system, the nearer the duct section is to the fan outlet or inlet, the greater its volume flow rate. The fan outlet or the main duct connected to the fan outlet is often the location where maximum air velocity occurs. The maximum design air velocity $v_{d,\max}$ is determined mainly according to the space available, noise control, energy use, and cost considerations. Return and branch ducts are nearer to the conditioned space and have comparatively lower volume flow rates than the supply ducts; therefore, supply main ducts often allow a higher $v_{d,\max}$ than return and branch ducts, except when there is a surplus pressure in a branch duct or a higher branch velocity is required to produce a negative local loss coefficient.

For supply air duct systems in high-rise commercial buildings, the maximum design air velocity in the supply duct $v_{d,\max}$ is often determined by the space available between the bottom of the beam and the suspended ceiling, as allocated by the architect, where the main duct traverses under the beam. Because of the impact in recent years of energy-efficient design of HVAC&R systems in commercial buildings (as influenced by ASHRAE Standards, local codes, and regulations) the following are true:

- In supply main ducts $v_{d,\max}$ usually does not exceeds 3000 fpm (15 m/s). Airflow noise must be checked at dampers, elbows, and branch takeoffs to satisfy the indoor *NC* range.
- In buildings with more demanding noise control criteria, such as hotels, apartments, and hospital wards, in supply main ducts usually $v_{d,\max} \leq$ 2000 to 2500 fpm (10 to 12.5 m/s), in return main ducts $v_{d,\max} \leq$ 1600 fpm (8 m/s), and in branch ducts $v_{d,\max} \leq$ 1200 fpm (6 m/s).

Higher air velocity results in a higher energy cost, and lower air velocity increases the material and labor costs of the installation. If a commercial building has sufficient headroom in its ceiling plenum, or if an industrial application has enough space at a higher level, an optimization procedure can then be used to reach a compromise between energy and installation costs. A check of airflow noise is still necessary.

For a particulate-transporting duct system, the air velocity must be higher than a specific value at any section of the duct system to float and transport the particulates. Design velocities for the components in an air duct system are listed in Table 17.9. The face velocity v_{fc}, in fpm (m/s), is defined as

$$v_{fc} = \frac{\dot{V}}{A_{g,fc}} \tag{17.70}$$

where $\dot{V}$ = air volume flow rate flowing through component, cfm (m^3/s)
$A_{g,fc}$ = gross face area of component perpendicular to airflow, width × height, ft^2 (m^2)

TABLE 17.9 Design Velocities for Air-Handling System Components

Duct element	Face velocity, fpm
Louvers	
Intake	
1. 7000 cfm and greater	400
2. 2000 to 7000 cfm	250 to 400
Exhaust	
1. 5000 cfm and greater	500
2. Less than 5000 cfm	300 to 400
Filters	
Panel filters	
1. Viscous impingement	200 to 800
2. Dry-type extended-surface	
Flat (low efficiency)	Duct velocity
Pleated media (medium efficiency)	up to 750
HEPA	250
Renewable media filters	
1. Moving-curtain viscous impingement	500
2. Moving-curtain dry-media	200
Electronic air cleaners, ionizing-type	150 to 350
Heating coils	
Steam and hot water	400 to 600 (200 min., 1500 max.)
Dehumidifying coils	500 to 600
Air washers	
Spray type	300 to 700
High-velocity spray type	1200 to 1800

Source: Adapted with permission from *ASHRAE Handbook 1989, Fundamentals*.

These face velocities are recommended based on the effectiveness in operation of the system component and its optimum pressure loss.

System Balancing

For an air duct system, system balancing means that the volume flow rate that comes from each outlet or flows into each inlet should be (1) equal or nearly equal to the design value for constant-volume systems and (2) equal to the predetermined values at maximum and minimum flow for variable-air-volume systems. System balancing is one of the primary requirements in air duct design. For supply duct systems installed in commercial buildings, using dampers only to provide design airflow often causes additional air leakage, as well as an increase in installation cost, and in some cases objectional noise. Therefore, system balancing using dampers only is not recommended.

A typical small supply duct system in which p_t of the conditioned space is equal to zero is shown in Fig. 17.20. For such a supply duct system, the pressure and volume flow characteristics are as follows:

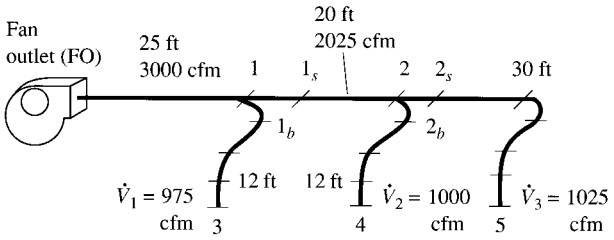


FIGURE 17.20 System balancing and critical path of a supply duct system.

- From node 1, the total pressure loss of the airstreams flowing through various paths 1-3, 1-2-4, and 1-2-5 to the conditioned space is always equal, i.e.,

$$\Delta p_{t,1\text{-}3} = \Delta p_{t,\,1\text{-}2\text{-}4} = \Delta p_{t,\,1\text{-}2\text{-}5} \tag{17.71}$$

- Volume flow rates $\dot{V}_1$, $\dot{V}_2$, and $\dot{V}_3$ supplied from the branch takeoffs, 1-3, 2-4, and 2-5 depend on the size and the configuration of the duct fittings and duct sections in the main duct and in the branch takeoffs as well as the characteritics of the supply outlet.

The relationship between the total pressure loss of any duct section Δp_t, the volume flow rate of the duct section $\dot{V}$, and the flow resistance R can be expressed as follows:

$$\Delta p_t = R\dot{V}^2 \tag{17.72}$$

For duct paths 1-3 and 1-2-45,

$$\begin{aligned}\Delta p_{t,1\text{-}3} &= \Delta p_{t,\,1\text{-}2\text{-}45} \\ &= R_{1\text{-}3}\dot{V}_1^2 = R_{1\text{-}2\text{-}45}(\dot{V}_2 + \dot{V}_3)^2\end{aligned}$$

Here, path 1-2-45 indicates the airflow having volume flow rate $\dot{V}_2 + \dot{V}_3$ and flowing through nodes 1, 2 and a combined parallel path 2-4 and 2-5.

At design conditions, the flow resistance $R_{1\text{-}2\text{-}45}$ is determined in such a manner that the total pressure loss $\Delta p_{t,1\text{-}2\text{-}45}$ along path 1-2-45 at a volume flow rate of $\dot{V}_2 + \dot{V}_3$ is balanced with the total pressure loss $\Delta p_{t,1\text{-}3.}$

- If the diverging wye or tee, terminal, duct fittings, and duct section used in path 1-3 are similar to those of paths 2-4 and 2-5, most probably $\dot{V}_1$ will be greater than the required design value as the result of lower $R_{1\text{-}3}$. To have required $\dot{V}_1$ flowing through path 1-3 at design conditions to provide a system balance, the following means are needed: (1) Decrease $R_{1\text{-}2\text{-}45}$, including an increase in the size of duct section 1-2; (2) increase $R_{1\text{-}3}$, mainly by using a smaller duct in section 1-3 and a diverging wye or tee of greater local loss coefficient $C_{c,b}$.
- In a duct system, if the flow resistance of each branch duct R_b is greater, the duct system is more easily adjusted to achieve an equal amount of supply volume flow from each of the branch supply outlets. For the branch takeoff or connecting duct having a length less than 1 to 2 ft (0.3 to 0.6 m) long, it is often difficult to increase R_b by reducing its size. Adding a volume damper directly on the supply outlet may alter space airflow patterns. A better remedy is to vary the sizes of the successive main duct sections with outlets of greater flow resistance to achieve a better system balance.
- A variable-air-volume duct system installed with a VAV box in each branch takeoff does provide system balancing automatically. However, only a part of the modulating capacity of a VAV box is allowed to be used to provide system balance of a specific branch takeoff (such as less than 20 percent), so that the quality of its modulation control at part-load operation is not impaired.

Critical Path

For any air duct system there exists a critical path, or design path of airflow, whose total flow resistance ΣR_{dgn} is maximum compared with other airflow paths when the volume flow rate of the critical path is equal to the design value at design conditions.

A critical path is usually a duct path with more duct fittings and comparatively higher volume flow rate; additionally, it may be the longest one. In Fig. 17.20, path FO-1-2-5 may be the critical path. For an energy-efficient air duct design, the total pressure loss, including the local loss coefficients of the duct fittings along the critical path, should always be minimized, especially at the fan inlet and outlet or their vicinity (see Chap. 21).

To reduce the dynamic losses of the critical path, the following are recommended:

- Maintain an optimum air velocity for airflow through the duct fittings.
- Emphasize the reduction of dynamic losses of duct fittings that are nearer to the fan outlet or inlet with a high air velocity, especially the fan system effects. Fan system effects are discussed in Chap. 21.
- Evans and Tsal (1996) recommend use of 90° elbows with an R_c/D or R_c/W of 1.5. If space is not available, use one or two splitter vanes in the elbows with throat radius or $R_c/W = 0.1$; the throat radius should not be smaller than 4 in. (100 mm). Turning vanes should not be used in a transitional elbow or an elbow other than 90°. Proper turning vane installation is critical to performance, which favors a factory-made unit.
- Set two duct fittings as far apart as allowable; if they are too close together, the eddies and large-scale turbulences of the first duct fitting often affect the velocity distribution in the second duct fitting and considerably increase the pressure loss in the second fitting.

For other duct paths, if the total pressure loss at the design flow rate is smaller than the pressure loss available, a smaller duct size and duct fittings of greater local loss coefficients may provide a better balance.

Air Leakage

Conditioned air leakage from the joints and seams of the air duct to the space, which is not air conditioned, is always a waste of refrigeration or heating energy as well as fan power. Based on their tested results, Swim and Griggs (1995) reported that the joints were the major leakage sites and accounted for 62 to 90 percent of the total air leakage from joints and seams. Air leakage depends mainly on the use of sealant on joints and seams, the quality of fabrication, and the shape of the ducts. Heat-sensitive tapes, mastic and glass fabric, and many other materials are used as the sealants for the joints and seams.

Duct leakage classifications, based on tests conducted by AISI, SMACNA, ASHRAE, and the Thermal Insulation Manufacturers Association (TIMA), are presented in Table 17.10. The air leakage rate $\dot{V}_L$, in cfm/100 ft^2 (L/s per m^2) of duct surface area can be calculated by the following formula:

$$\dot{V}_L = C_L \Delta p_{sd}^{0.65} \tag{17.73}$$

The leakage class can then be calculated as

$$C_L = \frac{\dot{V}_L}{\Delta p_{sd}^{0.65}} \tag{17.73a}$$

where C_L = leakage class, cfm per 100-ft^2 duct surface area at a static pressure difference of 1 in. WC [based on Eq. (17.73*a*)]
C = constant affected by area characteristics of leakage path
Δp_{sd} = static pressure differential between air inside and outside duct, in. WC (Pa)

TABLE 17.10 Duct Leakage Classification

Type of duct	Predicted leakage class C_L Sealed	Unsealed
Metal (flexible excluded)		
Round and oval	3	30 (6 to 70)
Rectangular		
≤2 in. WC (both positive and negative pressures)	12	48 (12 to 110)
>2 and ≤10 in. WC (both positive and negative pressures)	6	48 (12 to 110)
Flexible		
Metal, aluminum	8	30 (12 to 54)
Nonmetal	12	30 (4 to 54)
Fibrous glass		
Rectangular	6	NA
Round	3	NA

The leakage classes listed in this table are averages based on tests conducted by AISA/SMACNA 1972, ASHRAE/SMACNA/TIMA 1985, and ASHRAE 1988. Leakage classes listed are not necessarily recommendations on allowable leakage. The designer should determine allowable leakage and specify acceptable duct leakage classifications.

Source: *ASHRAE Handbook 1989, Fundamentals*. Reprinted with permission.

ASHRAE/IESNA Standard 90.1-1999 mandates that ductwork and plenum shall be sealed in accordance with the following requirements:

Location	Supply duct ≤2 in. WC	Supply duct >2 in. WC	Exhaust duct	Return duct
Outdoors	A	A	C	A
Unconditioned spaces	B	A	C	B
Conditioned spaces	C	B	B	C

The sealing requirement of sealing levels A, B, and C are as follows:

A = All transverse joints, longitudinal seams, and duct wall penetrations. Pressure-sensitive tape shall not be used as the primary sealant.

B = All transverse joints, longitudinal seams. Pressure-sensitive tape shall not be used as the primary sealant.

C = Transverse joints only.

Longitudinal seams are joints oriented in the direction of airflow. Transverse joints are connections of two duct sections oriented perpendicular to air flow. Duct wall penetrations are openings made by any screw fastener, pipe, rode, or wire. Spiral lock seams in round and flat oval duct need not be sealed. All other connections are considered transverse joints.

ASHRAE/IESNA Standard 90.1-1999 also mandates that:

- Metal round and flat oval ducts should have a leaking class C_L3, and metal rectangular ducts, rectangular fibrous ducts, and round flexible ducts have a C_L6.
- Ducts that are designed to operate at a static pressure exceeding 3 in. WC (750 Pa) shall be leak tested according to industry accepted test procedures. Representative sections totaling no less than 25 percent of total installed duct area for the designed pressure class shall be tested.
- The maximum permitted duct leakage $\dot{V}_L$, in cfm/100 ft^2 (L/s·m^2), shall be calculated from Eq. (17.73).

Shapes and Material of Air Ducts

When a designer chooses the shape (round, rectangular, or flat oval duct) or material (galvanized sheet, aluminum, fiberglass, or other materials) of an air duct, the choices depend mainly on the space available, noise, cost, local customs and union agreements, experience, quality, and the requirements of the project. In many high-rise commercial buildings, factory-fabricated round ducts and sometimes flat oval ducts with spiral seams are used because they have fewer sound problems, lower air leakage, and many configurations of wyes and tees available for easier pressure balance. Round ducts also have the advantage of high breakout transmission loss at low frequencies (see Chap. 19).

For ducts running inside the air conditioned space in industrial applications, metal rectangular ducts are often chosen for their large cross-sectional areas and convenient fabrication. Round ducts are often used for more demanding projects. In projects designed for lower cost, adequate duct insulation, and sound attenuation, fiberglass ducts may sometimes be the optimum selection.

Ductwork Installation

Ductwork installation, workmanship, materials, and methods must be monitored at all stages of the design and construction process to ensure that they meet the design intent.

Fire Protection

The design of air duct systems must meet the requirements of National Fire Codes NFPA 90A, *Standards for the Installation of Air Conditioning and Ventilating Systems, Warm Air Heating and Air Conditioning Systems, and Blower and Exhaust Systems* as well as local codes. Refer to these standards for details. The following are some of the requirements:

- The duct material discussed in Sec. 17.3 must be made of class 0 or class 1 material. Also the duct coverings and linings—including adhesive, insulation, banding, coating, and film covering the outside surface and material lining the inside surface of the duct—must have a flame spread rating not over 25 and a smoke development rating not over 50 except for ducts outside buildings. Supply ducts that are completely encased in a concrete floor slab not less than 2 in. (50 mm) thick need not meet the class 0 or class 1 requirement.
- Vertical ducts more than two stories high must be constructed of masonry, concrete, or clay tile.
- When ducts pass through the floors of buildings, the vertical openings must be enclosed with partitions and walls with a fire protection rating of not less than 1 h in buildings less than four stories high and greater than 2 h in buildings four stories and higher.
- Clearances between the ducts and combustible construction and material must be made as specified in NFPA 90A.
- The opening through a fire wall by the duct system must be protected by (1) a fire damper closing automatically within the fire wall and having a fire protection rating of not less than 3 h or (2) fire doors on the two sides of the fire wall. A service opening must be provided in ducts adjacent to

each fire damper. Many regulatory agencies have very rigid requirements for fire dampers, smoke dampers, fire/smoke dampers in combination, and smoke venting.

- When a duct penetrates through walls, floors, and partitions, the gap between the ducts and the walls, floors, and partitions must be filled with noncombustible material to prevent the spread of flames and smoke.
- Duct systems for transporting products, vapor, or dust in industrial applications must be constructed entirely of metal or noncombustible material. Longitudinal seams must be continuously welded, lapped and riveted, or spot-welded on maximum centers of 3 in. (75 mm). Transitions must be 5 in. (125 mm) long for every 1-in. (25-mm) change in diameter. Rectangular ducts may be used only when the space is not available and must be made as square as possible.

17.9 AIR DUCT DESIGN PROCEDURE AND DUCT LAYOUT

Design Procedure

Before an air duct system is designed, the supply volume flow rate for each conditioned space, room, or zone should be calculated, and the locations of the supply outlets and return inlets should also be settled according to the requirements of space air diffusion (see Chap. 18). For an air duct system, the supply volume flow rate of cold supply air in summer is usually greater than the warm volume flow rate needed in winter. If an air duct system conditions the space with cold air supply in summer, it often also conditions the space with warm air supply in winter.

Computer-aided duct design and sizing programs are widely used for more precise calculation and optimum sizing of large and more complicated duct systems. Manual air duct design and sizing are often limited to small and simple duct systems. Computer-aided duct design and sizing programs are discussed in a later section in this chapter.

The design procedure for an air duct system is as follows:

1. Designer should verify local customs, local codes, local union agreements, and material availability constraints before proceeding with duct design.

2. The designer proposes a preliminary duct layout to connect the supply outlets and return inlets with the fan(s) and other system components through the main ducts and branch takeoffs. The shape of the air duct is selected. Space available under the beam often determines the shape of the duct and affects the layout in high-rise buildings.

3. The duct layout is divided into consecutive duct sections, which converge and diverge at nodes or junctions. In a duct layout, a node or junction is represented by a cross-sectional plane perpendicular to airflow. The volume flow rate of any of the cross sections perpendicular to airflow in a duct section remains constant. A duct section may contain one or more duct segments (including duct fittings). A duct system should be divided at a node or junction where the airflow rate changes.

4. The local loss coefficients of the duct fittings along the tentative critical path should be minimized, especially adjacent to fan inlets and outlets.

5. Duct sizing methods should be selected according to the characteristics of the air duct system. The maximum design air velocity is determined based on the space available, noise, energy use, and initial cost of the duct system. Various duct sections along the tentative critical path are sized.

6. The total pressure loss of the tentative critical path as well as the air duct system is calculated.

7. The designer sizes the branch ducts and balances the total pressure at each junction of the duct system by varying the duct and component sizes, and the configuration of the duct fittings.

8. The supply volume flow rates are adjusted according to the duct heat gain at each supply outlet.

9. The designer resizes the duct sections, recalculates the total pressure loss, and balances the parallel paths from each junction.

10. The airborne and breakout sound level from various paths should be checked and the necessary attenuation added to meet requirements.

Duct System Characteristics

Air duct systems can be classified into the following three categories according to their system characteristics:

1. Supply duct, return duct, or exhaust duct systems with a certain pressure loss in branch takeoffs
2. Supply duct, return duct, or exhaust duct systems in which supply outlets or return grilles either are mounted directly on the duct or have only very short connecting duct between the outlet or inlet and the main duct
3. Industrial exhaust duct systems to transport dust particulates or other particulate products

Duct Layout

When a designer starts to sketch a preliminary duct layout using computer-aided design and drafting (CADD) or manually, the size of the air duct system (the conditioned space served by the air duct system) must be decided first. The size of an air duct system must be consistent with the size of the air system or even the air conditioning system. From the point of view of the air duct system itself, a smaller and shorter system requires less power consumption by the fan and shows a smaller duct heat gain or loss. The air duct system is also comparatively easier to balance and to operate.

If the designer uses a more symmetric layout, as shown in Fig. 17.21, a more direct and simpler form for the critical path can generally be derived. A symmetric layout usually has a smaller main duct and a shorter design path; it is easier to provide system balance for a symmetric than a nonsymmetric layout. A more direct and simpler form of critical path usually means a lower total pressure loss of the duct system.

For variable-air-volume air duct systems, the ends of the main ducts are connected to each other to form one or more duct loops, as shown by the dotted lines in Fig. 17.21. Duct looping(s) allow some of the duct sections to be fed from the opposition direction. Balance points exist where the total pressure of the opposite flowing airstreams is zero. The positions of the balance points often follow the sun's position and the induced cooling load at various zones during the operating period. Duct looping optimizes transporting capacity and results in a smaller main duct than without looping.

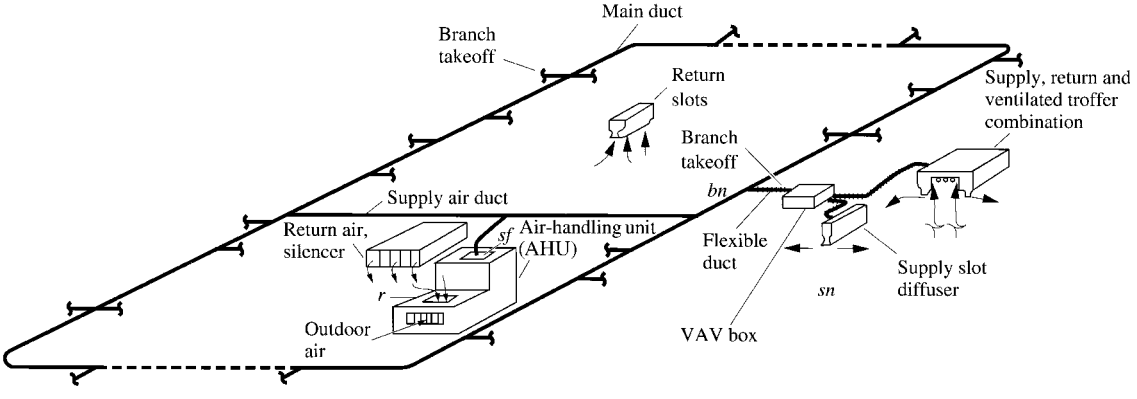


FIGURE 17.21 A typical supply duct system with symmetric duct layout (bold line) and duct loopings (connected by dotted lines) for a typical floor in a high-rise building.

The designer then compares various alternative layouts and reduces the number of duct fittings, especially the fittings with higher velocity and high local loss coefficients along the critical path, in order to achieve a duct system with lower pressure loss.

When duct systems are installed in commercial and public buildings without suspended ceiling, duct runs should be closely matched with the building structures and give a neat and harmonious appearance.

17.10 DUCT SIZING METHODS

Duct sizing determines the dimensions of each duct section in the air duct system. After the duct sections have been sized, the total pressure loss of the air duct system can then be calculated, and the supply, return or relief fan total pressure can be calculated from the total pressure losses of the supply and return duct systems and the pressure loss in the air-handling unit or packaged unit.

Four duct-sizing methods are currently used:

1. Equal-friction method with maximum velocity
2. Constant-velocity method
3. Static regain method
4. T method

Equal-Friction Method

This method sizes the air duct so that the duct friction loss per unit length $\Delta p_{f,u}$ at various duct sections always remains constant. The final dimensions of sized ducts should be rounded to standard size. The total pressure loss of the duct system Δp_t, in in. WC (Pa), equals the sum of the frictional losses and dynamic losses at various duct sections along the critical path:

$$\Delta p_t = \Delta p_{f,u}[(L_1 + L_2 + \cdots + L_n) + (L_{e1} + L_{e2} + \cdots + L_{en})] \tag{17.74}$$

where $L_1, L_2, \ldots, L_n$ = length of duct sections 1, 2, . . . , n, ft (m)
$L_{e1}, L_{e2}, \ldots, L_{en}$ = equivalent length of duct fittings in duct sections 1, 2, . . . , n, ft (m)

If the dynamic loss of a duct fitting is equal to the friction loss of a duct section of length L_e, in ft (m), then

$$\frac{C_o \rho v^2}{2g_c} = \frac{f(L_e/D)\rho v^2}{2g_c}$$

and the equivalent length is

$$L_e = \frac{C_o D}{f} \tag{17.75}$$

The selection of $\Delta p_{f,u}$ is usually based on experience, such as 0.1 in. WC per 100 ft (0.82 Pa/m) for low-pressure systems. A maximum velocity is often used as the upper limit.

The equal-friction method does not aim at an optimal cost. Dampers are sometimes necessary for a system balance. Because of its simple calculations, the equal-friction method is still used in many low-pressure systems in which airborne noise due to higher air velocity is not a problem or for small duct systems.

Constant-Velocity Method

The constant-velocity method is often used for exhaust systems that convey dust particles in industrial applications. This method first determines the minimum air velocity at various duct sections

according to the requirement to float the particles, either by calculation or by experience. On the basis of the determined air velocity, the cross-sectional area and, therefore, the dimension of the duct can be estimated and then rounded to a standard size. The total pressure loss of the duct system Δp_t, in in. WC (Pa), along the critical path can be calculated as

$$\Delta p_t = \frac{K\rho}{2g_c}\left[\left(\frac{f_1 L_1}{D_1} + C_1\right)v_1^2 + \left(\frac{f_2 L_2}{D_2} + C_2\right)v_2^2 + \cdots + \left(\frac{f_n L_n}{D_n} + C_n\right)v_n^2\right] \tag{17.76}$$

where $v_1, v_2, \ldots, v_n$ = mean air velocity at duct sections 1, 2, . . . , n, respectively, fpm (m/s)
$C_1, C_2, \ldots, C_n$ = local loss coefficients at duct sections 1, 2, . . . , n, respectively
$K = 5.35 \times 10^{-5}$ for I-P unit (1 for SI unit)

Static Regain Method

This method sizes the air duct so that the increase of static pressure (static regain) due to the reduction of air velocity in the supply main duct after each branch takeoff nearly offsets the pressure loss of the succeeding duct section along the main duct. As a consequence, the static pressure at the common end of the diverging tee or wye of the sized duct section remains approximately the same as that of the preceding section.

A rectangular duct section 1-2 between the cross-sectional planes 1 and 2 is illustrated in Fig. 17.22. The size of this duct section is to be determined. Let v_1 and v_2 be the mean velocities at

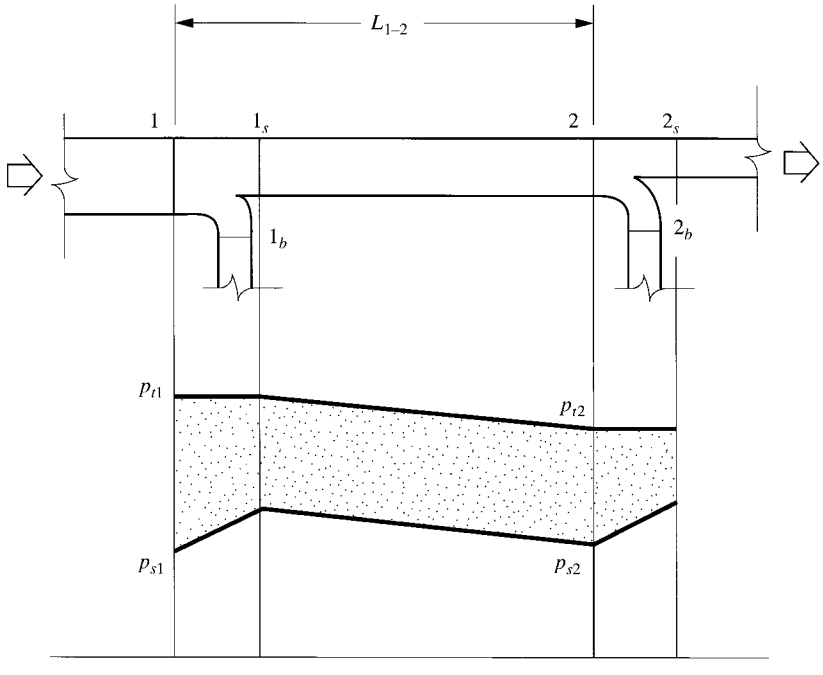


FIGURE 17.22 Pressure characteristics of a main duct section.

planes 1 and 2, $\dot{V}_1$ and $\dot{V}_2$ the volume flow rates, and A_1 and A_2 the cross-sectional areas. The total pressure loss in duct section 1-2 consists of the duct friction loss $\Delta p_{f1\text{-}2}$ and the dynamic loss of the main airstream flowing through the diverging tee $\Delta p_{1c,s}$. The relationship between the total pressure at planes 1 and 2 can be expressed as

$$p_{t1} = p_{t2} + \Delta p_{f1\text{-}2} + \Delta p_{1\,c,s} \tag{17.77}$$

because $p_t = p_s + p_v$. Ignore the difference between air densities ρ_1 and ρ_2. Let $\Delta p_{f1\text{-}2} = \Delta p_{f,u} L_{1\text{-}2}$. Here $L_{1\text{-}2}$ represents the length of the duct section 1-2, in ft (m). If the static pressures at planes 1 and 2 are equal, that is, $p_{s1} = p_{s2}$, then

$$\frac{\rho_1(v_1^2 - v_2^2)}{2g_c} = \Delta p_{f,u} L_{1\text{-}2} + \frac{C_{1\,c,s}\rho_1 v_1^2}{2g_c} \tag{17.78}$$

If v is expressed in fpm and $\Delta p_{f,u}$ in in. WC per 100 ft, $\rho = 0.075$ lb/ft^3, and $g_c = 32.2$ lb$_m \cdot$ ft/lb$_f \cdot$ s^2, the mean air velocity of the sized duct section is

$$v_2 = [(1 - C_{1\,c,s})\, v_1^2 - 1.6 \times 10^5 \Delta p_{f,u} L_{1\text{-}2}]^{0.5} \tag{17.79}$$

In SI units,

$$v_2 = \left[\frac{(1 - C_{1\,c,s})v_1^2 - 2\Delta p_{f,u} L_{1\text{-}2}}{\rho}\right]^{0.5} \tag{17.79a}$$

with v_1 and v_2 both in m/s, $\Delta p_{f,u}$ in Pa/m, $L_{1\text{-}2}$ in m, and ρ in kg/m^3.

For any duct section between cross-sectional planes $n - 1$ and n, if the total local loss coefficient of the duct fittings is C_n, excluding the local loss coefficient $C_{(n-1)c,s}$, the mean air velocity of the sized duct section is

$$v_n = \left\{\frac{[(1 - C_{(n-1)\,c,s})v_{n-1}^2 - 1.6 \times 10^5\, \Delta p_{f,u} L_n]}{1 + C_n}\right\}^{0.5} \tag{17.80}$$

Because v_{n-1}, ρ_n, L_n, *and* C_n are known values, by using iteration methods, v_n can be determined. The dimension of the duct section and its rounded standard size can also be determined.

The static regain method can be applied only to supply duct systems. It tends to produce a more even static pressure at the common end of each diverging tee or wye leading to the corresponding branch takeoff, which is helpful to the system balance. It does not consider cost optimization. The main duct sections remote from the fan discharge often have larger dimensions than those in the equal-friction method. Sound level and space required should be checked against determined air velocity and dimension.

When one is using the static regain method to size air ducts, it is not recommended to allow only part of static regain to be used in the calculation.

T Method

The T method, first introduced by Tsal et al. (1988), is an optimizing procedure to size air ducts by minimizing their life-cycle cost. It is based on the tree-staging idea and is therefore called the T method. The goal of this method is to optimize the ratio between the velocities in all sections of the duct system. The T method consists of the following procedures:

1. *System condensing*—condensing various duct sections of a duct system into a single imaginary duct section having the same hydraulic characteristics and installation costs as the duct system

2. *Fan selection*—selecting a fan that provides the optimum system pressure loss
3. *System expansion*—expanding the imaginary duct section into the original duct system before condensing with the optimum distribution of total pressure loss between various duct sections

During optimization, local loss coefficients are considered constant at various stages of iteration. For details, refer to Tsal et al. (1988).

The T method can be used for sizing duct systems with certain total pressure losses in the branch ducts. However, the local loss coefficients are actually varied at various stages of the iteration and should be taken into consideration during optimization.

17.11 DUCT SYSTEMS WITH CERTAIN PRESSURE LOSSES IN BRANCH TAKEOFFS

Design Characteristics

Supply, return, or exhaust duct systems with certain pressure losses in branch takeoffs have the following characteristics:

- Duct is sized based on the optimization of the life-cycle cost of various duct sections of the duct system as well as the space available in the building.
- System balancing is achieved mainly through pressure balancing of various duct paths by changing of duct sizes and the use of various configurations of duct fittings and terminals instead of dampers or other devices.
- Sound level will be checked and analyzed. Excess pressure at each inlet of VAV box at design conditions is to be avoided. Sound attenuation arrangements are added if necessary.
- Local loss coefficients of the duct fittings and equipment along the critical path are minimized. It may be beneficial to use the surplus pressure available in the branch takeoff to produce a higher branch duct velocity and a smaller straight-through local loss coefficient $C_{s,c}$.
- Supply volume flow rates are adjusted according to the duct heat gain. For VAV systems, diversity factors are used to determine the volume flow rate of various duct sections along the critical path so that the volume flow rate nearly matches the block load at the fan discharge.

Cost Optimization

For any duct section in an air duct system, the total life-cycle cost C_{to}, as shown in Fig. 17.23, in dollars, can be calculated as

$$C_{\text{to}} = C_e \frac{1}{\text{CRF}} + C_{di} \tag{17.81}$$

In Eq. (17.81), CRF indicates the capital recovery factor and can be calculated as follows:

$$\text{CRF} = \frac{i(1 + i)^n}{(1 + i)^n - 1} \tag{17.82}$$

where i = interest rate and n = number of years under consideration.

The first-year energy cost C_e, in dollars, can be calculated as the product of electric energy consumed at the fan and the unit energy cost E_r, in \$/kWh, times the annual operating hours t_{an}, in

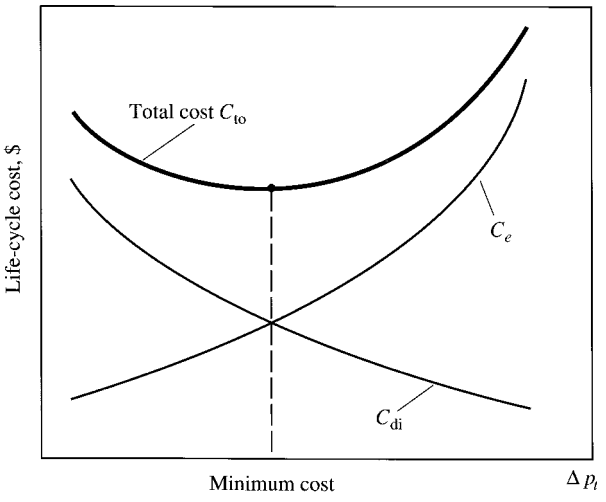


FIGURE 17.23 Cost analysis for a duct system.

h, or in I-P units

$$C_e = \Delta p_t \frac{0.746\ \dot{V} E_r t_{an}}{6350 \eta_f \eta_m}$$

$$= \Delta p_t \left(\frac{1.175 \times 10^{-4} \dot{V} E_r t_{an}}{\eta_f \eta_m} \right) = K_e \Delta p_t \tag{17.83}$$

For SI units,

$$K_e = \frac{\dot{V} E_r t_{an}}{1000 \eta_f \eta_m} \tag{17.84}$$

where Δp_t = total pressure loss of duct section, in. WC (Pa)
$\dot{V}$ = volume flow rate of duct section, cfm (m^3/s)
η_f, η_m = total efficiency of fan and efficiency of motor

Let $z_1 = K_e(1/\text{CRF})$. Then we can write

$$\frac{1}{\text{CRF}} C_e = z_1 \Delta p_t \tag{17.85}$$

Installation cost for round duct C_{di}, in dollars, can be calculated as

$$C_{di} = \pi D L C_{iu} \tag{17.86}$$

where D = diameter of round duct, ft (m)
L = length of duct section, ft (m)
C_{iu} = unit cost of duct installation, $/ft^2 ($/m^2)

The surface area of rectangular duct $A_{\text{rec}} = 2(W + H)L$. Here W is the width of the duct, and H is the height of the duct, both in ft (m). Then for rectangular duct

$$C_{di} = 2(W + H)LC_{iu} = \frac{2(W + H)}{\pi D}\pi DLC_{iu} = R_{\text{rec}}\pi DLC_{iu} \tag{17.87}$$

where R_{rec} = ratio of surface area of rectangular duct to surface area of round duct. For any duct section, the total pressure loss Δp_t, in in. WC (Pa), is

$$\Delta p_t = \frac{5.35 \times 10^{-5}(fL/D + \Sigma C)\rho v^2}{2g_c} \tag{17.88}$$

where ΣC = sum of local loss coefficients in duct section, based on air velocity of sized duct section
v = mean air velocity in sized duct section, fpm (m/s)

For frictional loss $D = f(\Delta p_t^{-0.2})$, and for dynamic loss $D = f(\Delta p_t^{-0.25})$. However, for simplicity, let us take $D = f(\Delta p_t^{-0.22})$ and consider $(fL/\sqrt{D}) + \Sigma C\sqrt{D}$ a constant. Then

$$\begin{aligned} D &= 0.1097\left[\frac{fL/\sqrt{D} + \Sigma C\sqrt{D})\rho}{g_c}\right]^{0.22}\dot{V}^{0.44}\Delta p_t^{-0.22} \\ &= 0.1097\left(\frac{\rho}{g_c}\right)^{0.22}K\frac{\Delta p_t^{-0.22}}{L} \end{aligned} \tag{17.89}$$

Because $g_c = 32.2\ \text{lb}_\text{m}\cdot\text{ft}/\text{lb}_\text{f}\cdot\text{s}^2$,

$$D = 0.0511\left[\left(\frac{fL}{\sqrt{D}} + \Sigma C\sqrt{D}\right)\rho\right]^{0.22}\dot{V}^{0.44}\Delta p_t^{-0.22} \tag{17.90}$$

where $K = (fL/\sqrt{D} + \Sigma C\sqrt{D})^{0.22}\dot{V}^{0.44}L$.

Let $z_2 = 0.1097\pi(\rho/g_c)^{0.22}C_{iu}$. By substituting into Eq. (17.86), for a round duct, we have

$$C_{di} = \pi DLC_{iu} = z_2K\,\Delta p_t^{-0.22} \tag{17.91}$$

For a rectangular duct

$$C_{di} = R_{\text{rec}}z_2K\Delta p_t^{-0.22} \tag{17.92}$$

Seeking a minimum cost by taking derivatives of Eq. (17.81) with respect to Δp_t, making it equal to zero, we have

$$\frac{dC_{to}}{d\Delta p_t} = z_1 - 0.22z_2K\Delta p_t^{-1.22} = 0$$

Because $g_c = 32.2\ \text{lb}_\text{m}\cdot\text{ft}/\text{lb}_\text{f}\cdot\text{s}^2$, then for a round duct

$$\Delta p_t = 108\left[\frac{LC_{iu}\eta_f\eta_m}{(1/\text{CRF})E_rt_{\text{an}}}\right]^{0.82}\left[\left(\frac{fL}{\sqrt{D}} + \Sigma C\sqrt{D}\right)\rho\right]^{0.18}\left(\frac{1}{\dot{V}^{0.46}}\right) \tag{17.93}$$

For a rectangular duct,

$$\Delta p_t = 108R_{\text{rec}}\left[\frac{LC_{iu}\eta_f\eta_m}{(1/\text{CRF})E_rt_{\text{an}}}\right]^{0.82}\left[\left(\frac{fL}{\sqrt{D}} + \Sigma C\sqrt{D}\right)\rho\right]^{0.18}\left(\frac{1}{\dot{V}^{0.46}}\right) \tag{17.94}$$

When the total pressure loss of the duct section that has the minimum cost is calculated from Eq. (17.93) or (17.94), the diameter or the circular equivalent of the duct section of minimum cost can then be calculated by Eq. (17.90).

Condensing Two Duct Sections

According to the T method, if two duct sections 1 and 2 connected in series are condensed into an imaginary section 1-2, as shown in Fig. 17.24*a*, then the volume flow rate at each duct section must be equal, i.e.,

$$\dot{V}_1 = \dot{V}_2 = \dot{V}_{1\text{-}2} \tag{17.95}$$

Here subscripts 1 and 2 and 1-2 represent the duct sections 1 and 2 and the imaginary duct section 1-2, respectively.

The total pressure loss of the condensed duct section must equal the sum of the total pressure losses of duct sections 1 and 2, or

$$\Delta p_{t1} + \Delta p_{t2} = \Delta p_{t1\text{-}2} \tag{17.96}$$

and the installation cost of the condensed section is

$$\begin{aligned} C_{di1\text{-}2} &= C_{di1} + C_{di2} \\ &= z_2(K_1 \Delta p_{t1}^{-0.22} + K_2 \Delta p_{t2}^{-0.22}) = z_2 K_{1\text{-}2} \Delta p_{t1\text{-}2}^{-0.22} \end{aligned} \tag{17.97}$$

When two duct sections 1 and 2 connected in parallel are condensed into an imaginary duct section 1-2, as shown in Fig. 17.24*b*, the following relationships hold:

$$\begin{aligned} \dot{V}_{1\text{-}2} &= \dot{V}_1 + \dot{V}_2 \\ \Delta p_{t1\text{-}2} &= \Delta p_{t1} = \Delta p_{t2} \\ C_{di1\text{-}2} &= C_{di1} + C_{di2} \end{aligned} \tag{17.98}$$

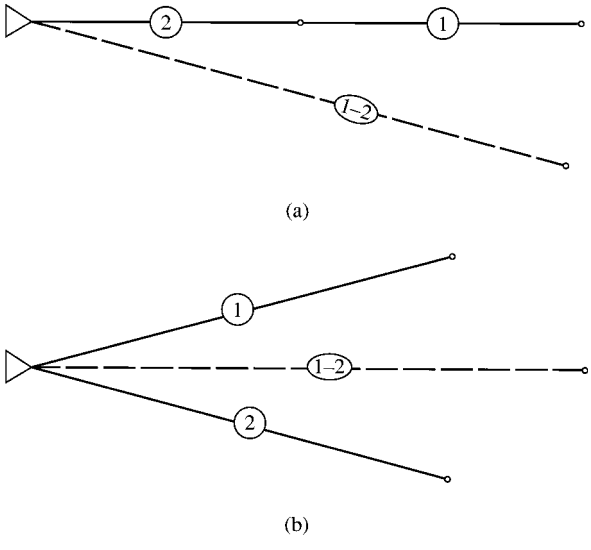


FIGURE 17.24 Two duct sections 1 and 2 condensed into an imaginary duct section 1-2: (*a*) two duct sections 1 and 2 connected in series; (*b*) two duct sections 1 and 2 connected in parallel.

If the whole duct system is condensed into one imaginary duct section, its minimum cost can be found by taking derivatives with respect to Δp_t of the imaginary duct section. In using such a procedure, the minimum cost includes both the main and branch ducts. However, if the local loss coefficients of the terminals and converging and diverging wyes are assumed constant, the benefits of using smaller terminals and different configuration of wyes to balance the branch duct paths are lost, as demonstrated by Dean et al. (1985). A more flexible and often economical alternative is that only the sizes of duct sections along the critical path of a duct system are determined according to the cost optimization procedure—that is, Eqs. (17.93), (17.94), and (17.90)—and rounded to standard size. The sizes of the duct sections of other duct paths, such as branch takeoffs, should be determined according to the difference between the total pressure at the junction and the space pressure for system or pressure balance using optimum terminals, wyes, and fittings.

Local Loss Coefficients for Diverging Tees and Wyes

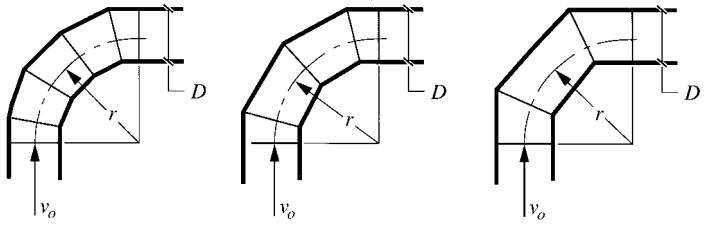
For a supply duct system, there are often more diverging tees or wyes than elbows along the design path. Selecting diverging tees or wyes with smaller local loss coefficients $C_{c,s}$ has a definite influence on the design of a duct system with minimizing total pressure loss of critical path.

Table 17.11 shows $C_{c,s}$ values for diverging wyes and tees for round ducts and diverging wyes for rectangular ducts. The following are recommendations for proper selection of wyes and tees:

- Select a diverging wye instead of a diverging tee except when there is surplus total pressure at the branch takeoffs. A 30° angle between the branch and the main duct has a smaller $C_{c,s}$ value than a 45° or a 60° angle.

TABLE 17.11 Local Loss Coefficients for Elbows, Diverging Tees, and Diverging Wyes

(1) Elbow; 3, 4, and 5 pieces, round



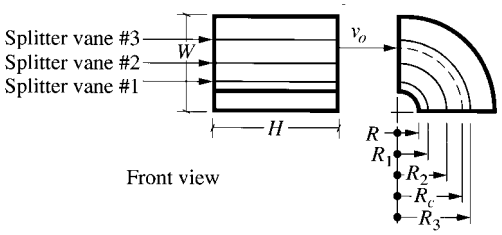
5-piece 4-piece 3-piece

Coefficients for 90° elbows C_o				
No. of pieces	R_c/D 0.75	1.0	1.5	2.0
5	0.46	0.33	0.24	0.19
4	0.50	0.37	0.27	0.24
3	0.54	0.42	0.34	0.33

Angle correction factors K_θ (Idelchik, 1986; diagram 6-1)											
θ	0	20	30	45	60	75	90	110	130	150	180
K_θ	0	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

TABLE 17.11 Local Loss Coefficients for Elbows, Diverging Tees, and Diverging Wyes (*Continued*)

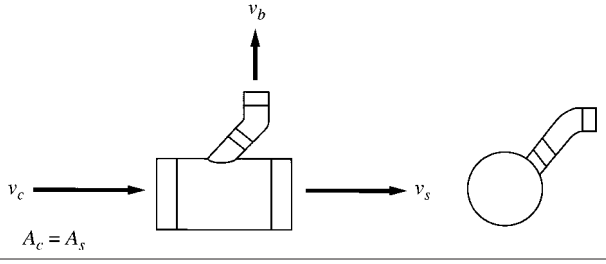
(2) Elbow, smooth radius with splitter vanes, rectangular



Coefficients for elbows with two splitter vanes C_o

			H/W										
R/W	R_c/W	CR	0.25	0.5	1.0	1.5	2.0	3.0	4.0	5.0	6.0	7.0	8.0
0.05	0.55	0.362	0.26	0.20	0.22	0.25	0.28	0.33	0.37	0.41	0.45	0.48	0.51
0.10	0.60	0.450	0.17	0.13	0.11	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.21
0.15	0.65	0.507	0.12	0.09	0.08	0.08	0.08	0.09	0.10	0.10	0.11	0.11	0.11
0.20	0.70	0.550	0.09	0.07	0.06	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07
0.25	0.75	0.585	0.08	0.05	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05
0.30	0.80	0.613	0.06	0.04	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04

(3) Wye, 45°, round, with 60° elbow, branch 90° to main



Branch											
v_b/v_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.88	0.77	0.68	0.65	0.69	0.73	0.88	1.14	1.54	2.2

Main									
v_s/v_c	0	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
$C_{c,s}$	0.40	0.32	0.26	0.20	0.14	0.10	0.06	0.02	0

(4) Tee, diverging, round, with 45° elbow, branch 90° to main

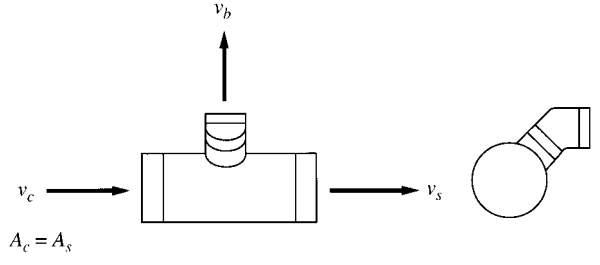
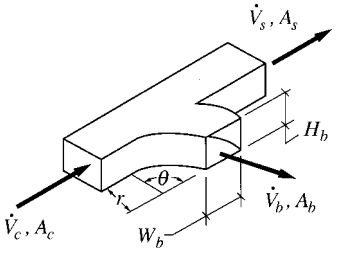


TABLE 17.11 Local Loss Coefficients for Elbows, Diverging Tees, and Diverging Wyes (*Continued*)

Branch										
V_h/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	1.32	1.51	1.60	1.65	1.74	1.87	2.0	2.2	2.5	2.7

For main local loss coefficient $C_{c,s}$, see values in (3)

(5) Tee, diverging, rectangular (Idelchik, 1986; diagram 7-21)



$\theta = 90°$

$r/W_b = 1.0$

		Branch, $C_{c,b}$								
		$\dot{V}_b/\dot{V}_c$								
A_b/A_s	A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.25	0.25	0.55	0.50	0.60	0.85	1.2	1.8	3.1	4.4	6.0
0.33	0.25	0.35	0.35	0.50	0.80	1.3	2.0	2.8	3.8	5.0
0.5	0.5	0.62	0.48	0.40	0.40	0.48	0.60	0.78	1.1	1.5
0.67	0.5	0.52	0.40	0.32	0.30	0.34	0.44	0.62	0.92	1.4
1.0	0.5	0.44	0.38	0.38	0.41	0.52	0.68	0.92	1.2	1.6
1.0	1.0	0.67	0.55	0.46	0.37	0.32	0.29	0.29	0.30	0.37
1.33	1.0	0.70	0.60	0.51	0.42	0.34	0.28	0.26	0.26	0.29
2.0	1.0	0.60	0.52	0.43	0.33	0.24	0.17	0.15	0.17	0.21
		Main, $C_{c,s}$								
		$\dot{V}_b/\dot{V}_c$								
A_b/A_s	A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.25	0.25	− 0.01	− 0.03	− 0.01	0.05	0.13	0.21	0.29	0.38	0.46
0.33	0.25	0.08	0	− 0.02	− 0.01	0.02	0.08	0.16	0.24	0.34
0.5	0.5	− 0.03	− 0.06	− 0.05	0	0.06	0.12	0.19	0.27	0.35
0.67	0.5	0.04	− 0.02	− 0.04	− 0.03	− 0.01	0.04	0.12	0.23	0.37
1.0	0.5	0.72	0.48	0.28	0.13	0.05	0.04	0.09	0.18	0.30
1.0	1.0	− 0.02	− 0.04	− 0.04	− 0.01	0.06	0.13	0.22	0.30	0.38
1.33	1.0	0.10	0	0.01	− 0.03	− 0.01	0.03	0.10	0.20	0.30
2.0	1.0	0.62	0.38	0.23	0.23	0.08	0.05	0.06	0.10	0.20

Source: *ASHRAE Handbook 1989, Fundamentals*. Reprinted with permission. For details, refer to *ASHRAE Handbook*.

- The smaller the difference in air velocity between the common end and the straight-through end, the lower the $C_{c,s}$ value.
- At a greater ratio of branch duct velocity to main duct velocity v_b/v_c, especially when $v_b/v_c > 1.2$, the $C_{c,s}$ value for the diverging wye of rectangular duct with smooth elbow takeoff is negative.

Return or Exhaust Duct Systems

Behls (1978) compared four designs of a bunker ventilation duct system with 10 risers connected to a main exhaust duct. The results of these four designs are as follows:

	System total pressure loss, in. WC (Pa)	Riser unbalance, in. WC (Pa)
Fixed-diameter riser, tee/diffuser	− 7.39 (− 1837)	6.55 (1628)
Variable-diameter riser, tee/diffuser	− 5.89 (− 1464)	1.77 (440)
Fixed-diameter riser, conical tee	− 5.29 (− 1315)	4.06 (1009)
Variable-diameter riser conical tee	− 4.03 (− 1002)	1.62 (403)

Using a greater branch duct velocity than the main duct, and varying the sizes of the risers, select proper duct fittings to decrease system total pressure loss and system unbalance.

Example 17.2. For a round supply duct system made of galvanized steel with spiral seams, as shown in Fig. 17.20, its operational and constructional characteristics are as follows:

Supply air temperature	60°F (15.6°C)
Kinematic viscosity of air	1.59×10^{-4} ft^2/s (1.46×10^{-5} m^2/s)
Density of supply air	0.075 lb/ft^3 (1.20 kg/m^3)
Absolute roughness	0.0003 ft (0.09 mm)
Local loss coefficients in section FO-1	0.5
Fan total efficiency, average	0.7
Motor efficiency	0.8
Electrical energy cost	\$0.08/kWh
Installation cost of duct	\$3.5/ft^2
CRF	0.10
Annual operating hours	3000

The total supply volume flow rate at the fan discharge is 3000 cfm (1.415 m^3/s), and the adjusted volume flow rates because of duct heat gain for each of the branch takeoffs are illustrated in Fig. 17.20. First, size this supply duct system. Then if each branch takeoff is connected to a VAV box that needs a total pressure loss of 0.75 in. WC (186 Pa) at its inlet, calculate the total pressure loss required for this supply duct system excluding the VAV box and downstream flexible duct and diffuser.

Solution:

1. For the first iteration, start with the round duct section 2-5, which may be the last section of the design critical path. Assume a diameter of 12 in., or 1 ft, and an air velocity of 1200 fpm, or 20 ft/s. Assume also that the local loss coefficient of the straight-through stream of the diverging tee $C_{c,s} = 0.15$, and for the elbow $C_o = 0.22$. The Reynolds number based on diameter D is

$$\mathrm{Re}_D = \frac{vD}{n} = \frac{20 \times 1}{1.59 \times 10^{-4}} = 125{,}786$$

From Eq. (17.42), the friction factor is

$$f = \frac{0.25}{\{\log [\varepsilon/(3.7D) + 5.74/(0.9\mathrm{Re}_D)]\}^2}$$
$$= \frac{0.25}{\{\log [0.0003/(3.7 \times 1) + 5.74/(0.9 \times 125{,}786)]\}^2} = 0.0166$$

From Eq. (17.93), the optimum pressure loss for the round duct section 2-5 is

$$\Delta p_{t2\text{-}5} = 108\left[\frac{LC_{iu}\eta_f\eta_m}{(1/\mathrm{CRF})E_t t_{\mathrm{an}}}\right]^{0.82}\left[\left(\frac{fL}{\sqrt{D}} + \Sigma C\sqrt{D}\right)\rho\right]^{0.18}\left(\frac{1}{\dot{V}^{0.46}}\right)$$
$$= 108\left(\frac{30 \times 3.5 \times 0.7 \times 0.8}{10 \times 0.08 \times 3000}\right)^{0.82}$$
$$\times \left[\left(\frac{0.0166 \times 30}{\sqrt{1.0}} + 0.37\sqrt{1}\right)0.075\right]^{0.18}\left(\frac{1}{1025^{0.46}}\right) = 0.13 \text{ in. WC}$$

And from Eq. (17.90), the diameter of the duct section 2-5 when $\Delta p_t = 0.13$ in. WC is

$$D_{2\text{-}5} = 0.0511\left[\left(\frac{fL}{\sqrt{D}} + \Sigma C\sqrt{D}\right)\rho\right]^{0.22}\dot{V}^{0.44}\Delta p_t^{-0.22}$$
$$= 0.0511\left[\left(\frac{0.0166 \times 30}{\sqrt{1.0}} + 0.37\sqrt{1.0}\right)0.075\right]^{0.22}1025^{0.44}\frac{1}{0.13^{0.22}}$$
$$= 0.927 \text{ ft or } 11.12 \text{ in. } (282 \text{ mm})$$

2. Duct section 1-2 is one of the sections of the critical path. For this section, if we assume that $C_{c,s} = 0.05$ and the diameter is 1.2 ft, then the calculated friction factor $f = 0.0155$ and the optimal total pressure loss is

$$\Delta p_{t1\text{-}2} = 108\left(\frac{20 \times 3.5 \times 0.7 \times 0.8}{10 \times 0.08 \times 3000}\right)^{0.82}$$
$$\times \left[\left(\frac{0.155 \times 20}{\sqrt{1.2}} + 0.05\sqrt{1.2}\right)0.075\right]^{0.18}\left(\frac{1}{2025^{0.46}}\right) = 0.0575 \text{ in. WC}$$

and the sized diameter is

$$D_{1\text{-}2} = 0.051\left[\frac{0.0155 \times 20}{\sqrt{1.2}} + 0.05\sqrt{1.2}\right)0.075\right]^{0.22}2025^{0.44}0.0575^{-0.22}$$
$$= 1.229 \text{ ft or } 14.75 \text{ in. } (375 \text{ mm})$$

3. Duct section FO-1 is one of the sections of the critical path. For this section, $\Sigma C = 0.5$. Let us assume that the diameter is 1.67 ft and the calculated $f = 0.0147$. Then

$$\Delta p_{t\mathrm{FO}\text{-}1} = 108\left(\frac{25 \times 3.5 \times 0.7 \times 0.8}{10 \times 0.08 \times 3000}\right)^{0.82}$$
$$\times \left[\left(\frac{0.147 \times 25}{\sqrt{1.67}} + 0.5\sqrt{1.67}\right)0.075\right]^{0.18}\frac{1}{3000^{0.46}} = 0.0692 \text{ in. WG}$$

and the sized diameter

$$D_{\text{FO-1}} = 0.051\left[\left(\frac{0.0147 \times 25}{\sqrt{1.67}}\right) + 0.5\sqrt{1.67}\right)0.075\Bigg]^{0.22} 3000^{0.44}0.0692^{-0.22}$$

$$= 1.734 \text{ ft or } 20.81 \text{ in. (529 mm)}$$

4. Duct section 2-4 is another leg from junction 2. It must have the same total pressure loss as duct section 2-5. Assume that $C_{c,b} = 1$ and the diameter is 11 in., or 0.917 ft. Then

$$D_{2\text{-}4} = 0.051\left[\left(\frac{0.0166 \times 12}{\sqrt{0.917}} + 1\sqrt{0.917}\right)0.075\Bigg]^{0.22} 1000^{0.44}0.13^{-0.22}$$

$$= 0.978 \text{ ft or } 11.78 \text{ in. (299 mm)}$$

5. Duct section 1-3 is another leg from junction 1. It must have the sum of the total pressure loss of duct sections 1-2 and 2-5, that is, $\Delta p_{t1\text{-}3} = 0.0575 + 0.13 = 0.1875$ in. WC. Assume that $C_{c,b} = 0.8$ and the diameter is also 11 in. Then

$$D_{1\text{-}3} = 0.051\left[\left(\frac{0.0166 \times 12}{\sqrt{0.917}} + 0.8\sqrt{0.917}\right)0.075\Bigg]^{0.22} 975^{0.44}0.1875^{-0.22}$$

$$= 0.858 \text{ ft or } 10.3 \text{ in. (262 mm)}$$

6. The results of the first iteration are listed in Table 17.12. These results are rounded to standard sizes and provide the information for the selection of the diverging tee and wye and the determination of the local loss coefficient.

7. For branch takeoffs 2-4 and 1-3, select proper diverging tees and wyes and, therefore, the $C_{c,b}$ values based on the air velocity of the sized duct sections. Vary the size of the duct sections if necessary so that their Δp_t values are approximately equal to the Δp_t of section 2-5 for branch duct 2-4, and the Δp_t of the sum of sections 2-5 and 1-2 for branch 1-3. Use Eq. (17.88) to calculate the total pressure loss of sections 2-4 and 1-3 according to the rounded diameters.

8. After the diverging tee or wye is selected, recalculate the optimum total pressure losses and diameters for duct sections 2-5, 1-2, and FO-1 from Eqs. (17.93) and (17.90).

9. After two iterations, the final sizes of the duct sections, as listed in Table 17.12, are the following:

Section 2-5, 11 in.; wye, 45°, diverging, round, 30° elbow

Section 1-2, 14 in.; tee, diverging, round, 90° elbow

Section FO-1, 20 in.

Section 2-4, 11 in.

Section 1-3, 12 in.

For duct section 1-2, at $v_s/v_c = 1.2$, $C_{c,s}$ is about 0.07 according to Idelchik (1986), diagram 7-17.

TABLE 17.12 Results of Computations of Duct Sizes and Total Pressure Loss in Example 17.2

		First iteration			Final results									
		Diameter			Air velocity			Velocity ratio						
Duct section	Volume flow V, cfm	ft	in.	Rounded in.	fpm	fps	Friction factor f	v_s/v_c	v_b/v_c	$C_{c,s}$	$C_{c,b}$	C	Diameter	p_t. in. WC
2-5	1025	0.927	11.12	11	1553	25.88	0.0166	0.94		0.011		0.22	11	0.116
1-2	2025	1.229	14.75	15	1894	31.97	0.0155	1.20		0.07			14	0.0645
FO-1	3000	1.734	20.81	20	1374	22.92	0.0147					0.5	20	0.0847
2-4	1000	0.978	11.78	11	1515	25.25			0.80		0.65		11	
1-3	975	0.858	10.3	11	1241	20.69			0.90		1.70		12	

10. Because the local loss coefficient $C_{c,b}$ for the branch stream from the 45° diverging wye at junction 2 can be reduced to 0.45 if a conical branch is used, the critical path consists of duct sections FO-1, 1-2, and 2-5. The total pressure loss of this supply duct system excluding the VAV box and downstream flexible duct and diffuser, from Table 17.12, is

$$\Delta p_t = 0.116 + 0.0645 + 0.0847 = 0.2652 \text{ in. WC (66 Pa)}$$

17.12 DUCT SYSTEMS WITH NEGLIGIBLE PRESSURE LOSS AT BRANCH DUCTS

Supply Duct Systems

When supply outlets are directly mounted on the main duct without branch takeoffs, or the connecting duct between the supply outlet and the main duct is about 2 ft or less, the total pressure loss of the branch duct, excluding the supply outlet, is often very small or negligible. In such circumstances, system balancing of the supply duct system depends mainly on the sizes of the successive main duct sections. If volume dampers are installed just before the outlet or in the connecting duct, the damper modulation will also vary the space diffusion airflow pattern.

Pressure Characteristics of Airflow in Supply Ducts

The rectangular supply duct with transversal slots shown in Fig. 17.25 is an example of a supply duct system with a supply outlet directly mounted on the main duct. The volume flow per ft^2 (m^2) of floor area of this type of duct system may sometimes exceed 8 cfm/ft^2 (40.6 L/s·m^2). This type of supply duct system is often used in many industrial applications.

Consider two planes n and $n + 1$, with a distance of 1 unit length, say, 1 ft (0.305 m), between them. If the space air is at atmospheric pressure, the total pressure loss of the supply air that flows from plane n, turns a 90°, and discharges through the slots is

$$\Delta p_{tn} = p_{tn} = \Delta p_{c,bn} + \Delta p_{ton} = \frac{(C_{c,bn} v_n^2 + C_o v_{on}^2)\rho}{2g_c} \tag{17.99}$$

where $\Delta p_{c,bn}$, Δp_{ton} = dynamic loss of branch stream when it makes a 90° turn and when it discharges from slots, lb/ft^2 or in. WC (Pa)
$C_{c,bn}$ = local loss coefficient of diverging branch stream with reference to velocity at plane n
C_o = local loss coefficient of transversal slots with reference to velocity at slot
v_n, v_{on} = air velocity at plane n and at slot from plane n, ft/s (m/s)

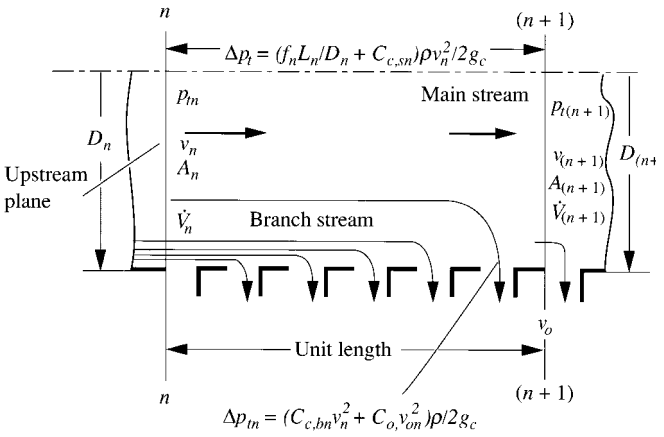
Similarly, at plane $n + 1$ the total pressure loss of the branch stream when it discharges from the slots is

$$p_{t(n+1)} = \frac{(C_{c,b(n+1)} v_{(n+1)}^2 + C_o v_{o(n+1)}^2)\rho}{2g_c} \tag{17.100}$$

The relationship of the total pressure between planes n and $n + 1$ along the airflow is given as

$$p_{tn} = p_{t(n+1)} + \frac{(f_n L_n / D_n + C_{c,sn})\rho v_n^2}{2g_c} \tag{17.101}$$

In Eq. (17.101), f_n, L_n, D_n, and $C_{s,cn}$ indicate the friction factor, length, circular equivalent, and $C_{c,s}$ value between plane n and $n + 1$, respectively. Substituting Eqs. (17.99) and (17.100) into



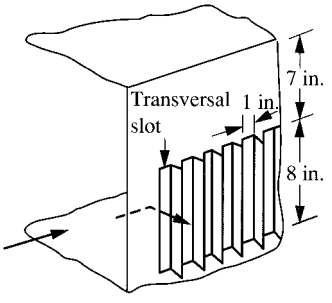


FIGURE 17.25 Rectangular supply duct with transversal slots.

Eq. (17.101), Wang et al. (1984) recommended the following formula to calculate v_{n+1}, in ft/s (m/s), based on the balanced total pressure before the transversal slots:

$$v_{(n+1)} = \left\{\left[(v_{on}^2 - v_{o(n+1)}^2)C_o + \left(C_{c,bn} - C_{c,sn} - \frac{f_n L_n}{D_n}\right)v_n^2\right]\left(\frac{1}{C_{c,b(n+1)}}\right)\right\}^{0.5} \quad (17.102)$$

In Eq. (17.102), v_{on} and $v_{o(n+1)}$, in ft/s (m/s), are the supply air velocities at the slots. For a cold air supply, usually $v_{o(n+1)} > v_{on}$ is desirable because of the effect of the duct heat gain.

For a free area ratio of slot area to duct wall area of 0.5, C_o for a perforated plate including discharged velocity pressure can be taken as 1.5. The v_n, L_n, and D_n are known values during the calculation of v_{n+1}. Local loss coefficients $C_{c,bn}$ and $C_{c,sn}$ can be determined from the experimental curves shown in Figs. 17.26 and 17.27. The coefficient $C_{c,b(n+1)}$ can be assumed to have a value similar to $C_{c,bn}$, and f can be calculated from Eq. (17.42). The mean air velocity in the duct at plane $n + 1$, denoted by $v_{n+1,}$ can then be calculated from Eq. (17.102).

In Figs. 17.26 and 17.27, $\dot{V}_{on}$ indicates the volume flow rate of supply air discharged from the slots per ft length of the duct, in cfm/ft (L/s · m); $\dot{V}_n$ represents the volume flow rate of the supply air inside the duct at plane n, in cfm (m^3/s).

Sizing of this kind of supply duct system starts from the duct section immediately after the fan discharge. This section can be sized from Eqs. (17.90) and (17.93) based on life-cycle cost

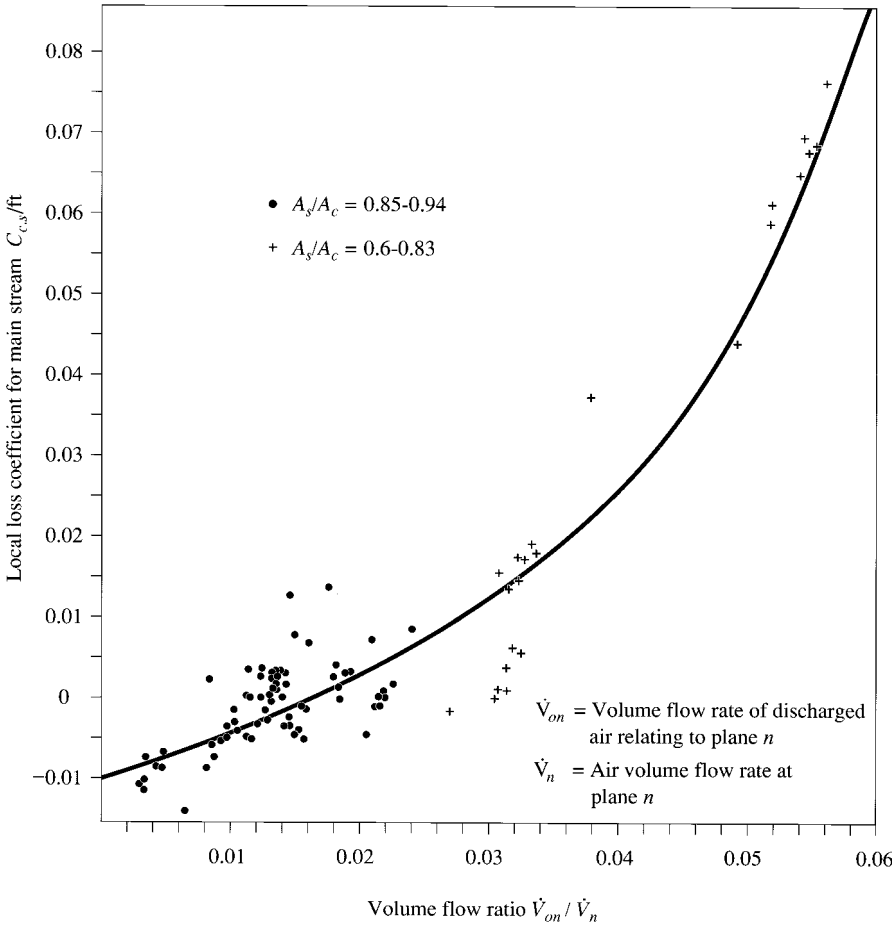


FIGURE 17.26 Local loss coefficient $C_{c,s}$ versus volume flow ratio $\dot{V}_{on}/\dot{V}_n$. (*Source: ASHRAE Transactions, 1984, Part II A. Reprinted with permission.*)

optimization. The designer should check for space available and noise, if sound control is required. The successive duct sections can then be sized by calculating from Eq. (17.102).

If supply duct systems with negligible pressure loss at branch ducts are installed with outlets whose $C_{c,bn}$ and $C_{c,sn}$ are not known, the static regain method is recommended for sizing the supply main duct for a large duct system and the equal-friction method for a small system.

Example 17.3. A galvanized steel rectangular duct section with transversal slots has the following construction and operational characteristics at plane n

Dimension:	4-ft width × 3-ft height (1219 mm × 914 mm)
Absolute roughness ε	0.0005 ft (0.15 mm)
Supply air temperature	65°F (18.3°C)
Volume flow rate $\dot{V}_n$	23,760 cfm (11.21 m^3/s)

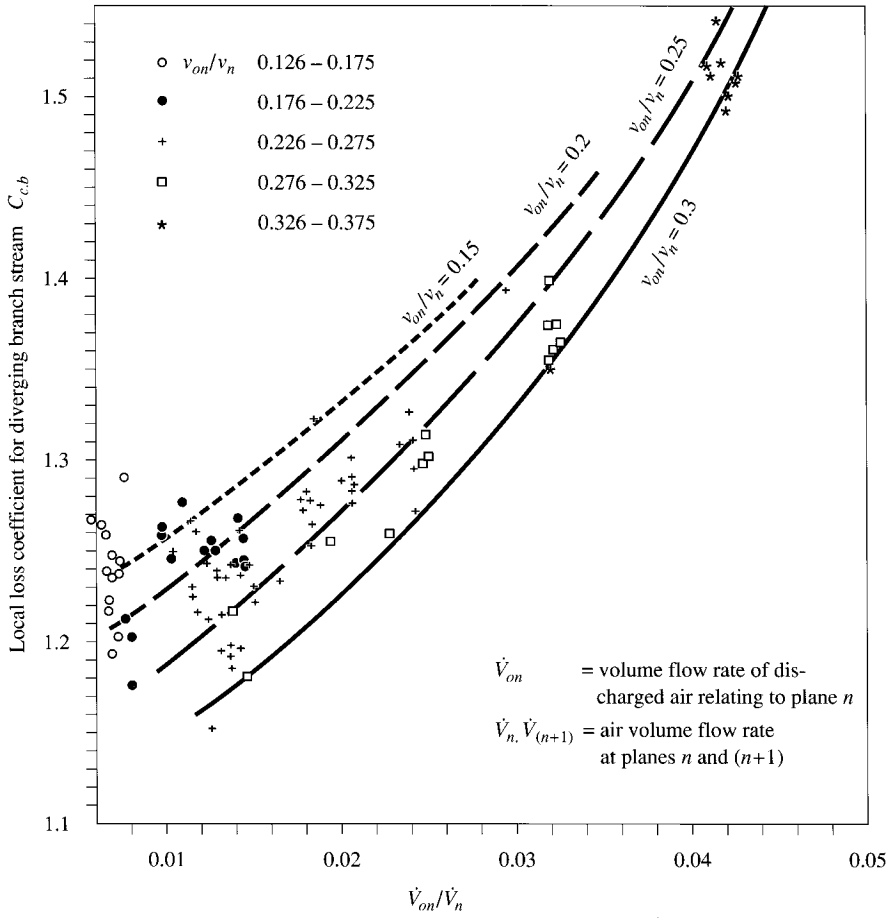


FIGURE 17.27 Local loss coefficient $C_{c,b}$ versus volume flow ratio $\dot{V}_{on}/\dot{V}_n$. (*Source: ASHRAE Transactions, 1984, Part II A. Reprinted with permission.*)

Volume flow discharged from slots related to plane n	270 cfm/ft (417.9 L/s·m)
Discharged velocity at slots related to plane n	7.5 ft/s or 450 fpm (2.29 m/s)
Required discharged velocity at plane $n + 1$	7.6 ft/s or 456 fpm (2.32 m/s)

If the height of the rectangular duct remains the same, size the dimension of this duct at plane $n + 1$, 10 ft (3.05 m) from plane n.

Solution

1. From the information given, the air velocity at plane n is

$$v_n = \frac{23{,}760}{60 \times 4 \times 3} = 33 \text{ ft/s (1980 fpm)}$$

And from Eq. (17.48), the circular equivalent is

$$D_e = \frac{1.3(ab)^{0.625}}{(a + b)^{0.25}} = \frac{1.3(3 \times 4)^{0.625}}{(3 + 4)^{0.25}} = 2.9 \text{ ft}$$

The Reynolds number of the supply air

$$\text{Re}_D = \frac{33 \times 2.9}{1.62 \times 10^{-4}} = 596{,}700$$

The friction factor

$$f = \frac{0.25}{\{\log[\varepsilon/(3.7D) + 5.74/(0.9\,\text{Re}_D)]\}^2}$$
$$= \frac{0.25}{\{\log[0.0005/(3.7 \times 2.9) + 5.74/(0.9 \times 596{,}700)]\}^2} = 0.0139$$

2. From the information given, the volume flow ratio is

$$\frac{\dot{V}_{on}}{\dot{V}_n} = \frac{270}{23{,}760} = 0.0113$$

and the velocity ratio is

$$\frac{v_{on}}{v_n} = \frac{7.5}{33} = 0.227$$

From Figs. 17.26 and 17.27, $C_{c,sn} = -0.0025$ per ft and $C_{c,bn} = 1.22$.

3. Assume that $C_{c,b(n+1)} = 1.23$. Then from Eq. (17.102),

$$v_{(n+1)} = \left\{\left[(v_{on}^2 - v_{o(n+1)}^2)C_o + \left(C_{c,bn} - C_{c,sn} - \frac{fL}{D}\right)v_n^2\right]\frac{1}{C_{c,b(n+1)}}\right\}^{0.5}$$
$$= (7.5^2 - 7.6^2)1.5 + \left[\frac{1.22 - (0.0139 \times 10/2.9) - 0.0025 \times 10)]33^2}{1.23}\right]^{0.5}$$
$$= 29.6 \text{ ft/s } (9.02 \text{ m/s})$$

4. The required volume flow rate from the slots related to plane $n + 1$ is

$$\dot{V}_{o(n+1)} = \frac{9270 \times 7.6}{7.5} = 273.6 \text{ cfm/ft}$$

Then the volume flow rate of air at plane $n + 1$ is

$$\dot{V}_{o(n+1)} = 23{,}760 - 273.6 \times 10 = 21{,}024 \text{ cfm}$$

The duct area required at plane $n + 1$ is

$$A_{n+1} = \frac{21{,}024}{60 \times 29.6} = 11.83 \text{ ft}^2 (1.00 \text{ m}^2)$$

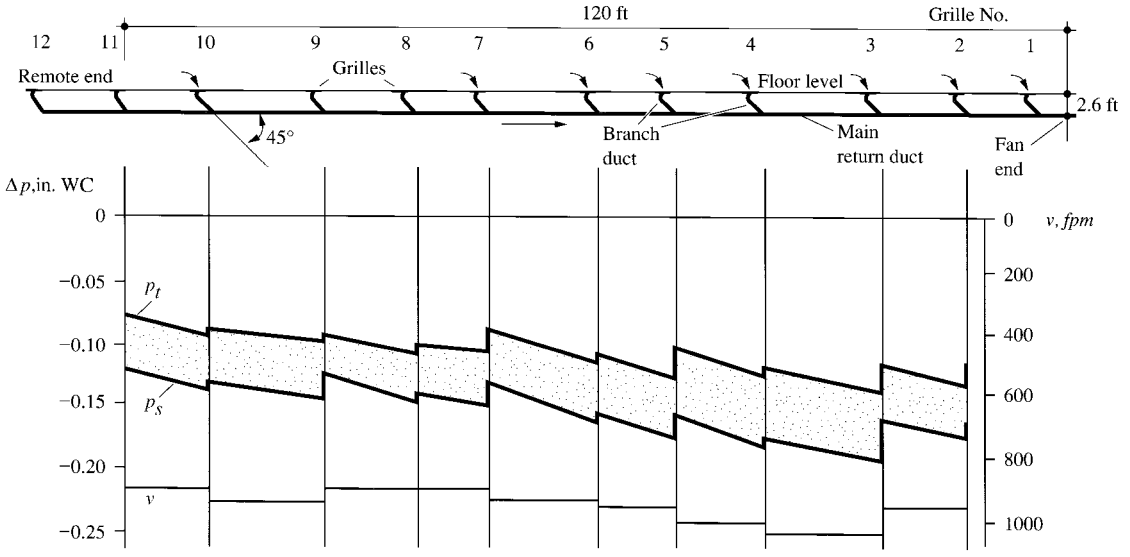


FIGURE 17.28 A return duct system with short connecting ducts for a textile mill.

The width of the duct is

$$W = \frac{10.78}{3} = 3.59 \text{ ft } (1.095 \text{ m})$$

Return or Exhaust Duct Systems

Return duct systems with very short connecting ducts between the return inlet and the main duct are often used in industrial applications. They are often used in large workshops and need nearly equal return volume flow rate for each branch intake. To provide a better system balance, it is necessary to reduce the total pressure difference between the branch inlets at the two ends of the main duct.

For a conditioned space where noise is not a major problem, one effective means of reducing the total pressure loss of a long return main duct is to have a higher branch duct velocity than the velocity in the main duct. Because a total pressure difference between the branch inlets and the main duct at two ends is inevitable, the area of the return grille can be varied along the main duct to provide a more even return volume flow rate. Unlike the supply outlets, the variation of the area of the return grille has a minor effect on the space air diffusion.

A return duct system with short connecting ducts is shown in Fig. 17.28 for a textile mill. Twelve floor grilles are connected to the return main duct at the lower floor. The connecting duct has a floor to main return duct difference of 2.6 ft (0.79 m). The sizes of the branch ducts vary from 16 in. × 16 in. (406 mm × 406 mm) at the remote end to 12 in. × 8 in. (305 mm × 203 mm) near the fan end. The main duct varies from 16 in. × 32 in. (406 mm × 813 mm) to 60 in. × 32 in. (1524 mm × 813 mm). In Fig. 17.28, the variations in total pressure p_t and static pressure p_s along the main duct are shown by the upper and middle curves, in in. WG (Pa); and the velocity in main return duct v, in fpm (m/s), is shown by the lower curve. The velocity in the branches varies from 700 fpm (3.56 m/s) at the remote end to 2000 fpm (10.16 m/s) at the fan end. Note that p_t at grille 6 is − 0.111 in. WG (− 27.6 Pa), − 0.105 in. WG (− 26.1 Pa) at grille 5, and − 0.114 in. WG

(− 28.3 Pa) at grille 2. Such a satisfactory pressure balance along the return duct is mainly due to the higher branch velocity in the grilles near the fan end.

17.13 REQUIREMENTS OF EXHAUST DUCT SYSTEMS FOR A MINIMUM VELOCITY

To transport dust particles or particulate products contained in the air, exhaust duct systems require a minimum velocity in all duct sections of the system. These systems are used in many industrial applications and usually have air velocity ranging between 2400 and 4000 fpm (12.2 and 20.3 m/s). In addition to the variation of the sizes of the branches' ducts, it is essential to select the proper configuration of duct fittings to provide a better system balance and to reduce the total pressure loss of the system (critical path). The following are recommendations for exhaust duct system designs:

- Round ducts usually produce smaller pressure losses; they are more rigid in construction.
- Air velocity inside the duct must not exceed the minimum velocity too much, in order not to waste energy.
- Well-sealed joints and seams are important for reducing air leakage at higher pressure differentials.

17.14 COMPUTER-AIDED DUCT DESIGN AND DRAFTING

According to Amistadi (1993) and "Scientific Computing" (1998), the following computer-aided duct design and drafting (CADD) programs are widely used in air duct design: E20-II Duct Design, Carrier Corp.; Varitrane Duct Designer, The Trane Co.; and Softdesk HVAC. Carrier's E20-II requires a disk space of installation of 4 MBytes and a RAM of 1 MByte, Trane's Varitrane needs a disk space of 13 MBytes and a RAM of 12 MBytes, and Softdesk HVAC needs a disk space of 7 MBytes and a RAM of 8 MBytes. All these three programs will run in Windows 95.

An effective and intelligent computer-aided duct design and drafting program provides the following functions:

Drafting

The computer program provides full, three-dimensional (3D) capabilities and draws round, rectangular, and flat oval ducts and fittings in any cross-sectional plane. The program offers two types of single-line layouts: (1) duct and fitting, which automatically produce elbows, transitions, and reducers from plan or multistory views, and (2) isometric, which allows the designer to produce a 3D drawing in an easy way. A single-line layout can be converted to double-line or 3D ducts and fittings layouts, or the designer can construct the 2D double-line or 3D ducts and fittings drawings directly. Duct layout can be produced in 2D or 3D in different layers simultaneously.

Schedules and Layering

Schedules of a computer-aided duct design program include diffusers, ducts, fittings, and related equipment, such as terminals and air-handling units or packaged units. Duct schedules include the tag number, quantity, size, type, length, gauge, and bill of material. Fitting schedules include the tag number, quantity, type, name, gauge, fitting junction sizes, and cfm (L/s). Values can be totaled and stored for each duct or fitting type. Tag editing capacity includes: creating, moving, and renumbering. The computer program prefers to have a nested-layer hierachy of five levels deep including building information, plan types, layer name and level (elevation), and construction modifier. Build-

ing information layers are grouped by supply, return, and exhaust ducts and subdivided into ducts, diffusers, and fittings. For each of these duct components, layers can be single-line, 2D or 3D construction with tags, labels, dimensions, and accessories. Plan types similar to the drawing information layers include 2D or 3D supply, return, and exhaust ducts. Besides, there are also floor, interior, exterior, and bird's-eye view plans.

A level layer indicates its attribute characteristic and the vertical elevation. Each layer can be edited, modified, and displayed.

Design Interface

Computer-aided duct design software either must provide interfaces with available duct sizing programs or itself must have the software for duct sizing. It is preferable that the duct design software size the ducts by using equal friction, static regain, constant-velocity, and even cost optimiztion T methods. The duct design software extracts system geometry and flow rates from the AutoCAD drawing, exports the information to the selected duct sizing program, and uses the sizes calculated to produce a double-line or 3D drawing to replace the centerline drawing.

Running Processes

Drawing a Centerline Layout. Designers are required to enter a fan and centerline layout in AutoCAD. They have the option to add terminals, diffusers, necessary dampers, fittings, duct sections, and related data to the drawing. Terminal text, line color, and fitting text are interpreted as the airflow, duct shape, and fitting configurations. An ASHRAE Duct Fittings Database with local loss coefficients of 225 types of fittings provides default fitting types and coefficients.

Analyzing the Duct System. With a completed centerline layout, designers are asked to choose the duct sizing software. Default values that are used for the centerline layout, such as duct shape, and airflow rates are passed to the duct-sizing software.

The designer is asked to select the fan location. The computer program connects all lines to the the fan in an attempt to connect each line to the end of the preceding line. The computer program tests the continuity of the duct system by checking that all selected components are connected and all junctions can be found. All lines must have a fan, terminal, or diffuser, or another centerline at each end. The computer program generates nodes and inserts a node tag for the fan, main duct section, and branch takeoff endpoints. Node shape is determined according to the duct shape and class of node. Duct section information is stored in the upstream node. Each main duct node and branch duct node are set with default attributes required by the duct-sizing program.

Editing Default Attributes and Running Duct-Sizing Program. If the designer is satified with all default values, he or she can proceed to run the duct-sizing computer programs. If the designer needs to change any of the default values, she or he can use AutoCAD to edit duct-sizing input default attributes and visually pick the duct sections from the layout for editing before running the duct-sizing program.

When the editing work is done, modified node data are used to update input data for the duct-sizing program. The designer uses the command to run the chosen duct-sizing program.

Converting from Centerline to Double-Line or 3D. Centerline layout or centerline 3D layout can be input into sizing calculations, and the computer program will produce a double-line layout or a 3D duct system layout. The conversion begins with the fan and produces the layout in the following order: fittings, transitions, reducers, duct segments, and dampers.

Total Pressure Loss and Fan-Sizing Calculations. Some computer-aided duct design computer programs have the ability to determine an existing duct layout shape and size in the CAD environment and load it into the duct system total pressure loss and fan sizing calculations.

17.15 DUCT LINER AND DUCT CLEANING

Duct Liner

Duct liners are lined internally on the inner surface of the duct wall. They are mainly used for noise attenuation and form part of the duct insulation layer that helps to reduce the heat transfer between the conditioned air inside the duct and the ambient air surrounding the duct. Fiberglass blanket or boards with a thickness of $^1/_2$ in. (13 mm), 1 in. (25 mm), $1^1/_2$ in. (38 mm), and 2 in. (50 mm) have been used as duct liner for decades and proved to be cost-effective.

In 1994, the U.S. Department of Health and Human Services (DHHS) announced that fiberglass will be listed as a material "reasonably anticipated to be a carcinogen." Although there are many valid arguments concerning mold growth and fiber erosion from fiberglass duct liners, the use of fiberglass in several institutional, educational, and medical projects has been banned or severely limited. Recommendations concerning the use of fiberglass duct liner and alternatives in HVAC&R systems are discussed in Secs. 19.3 and 19.5.

Duct Cleaning

The accumulation of dirt and debris as well as the growth of mold and fungi inside the ductwork normally will not occur in a properly designed (such as equipped with medium- or high-efficiency air filters), installed, operated, and maintained air system including ductwork. When the cause of indoor air pollution is dirty ductwork — accumulation of dirt and debris or mold growth, or both inside the ductwork — source removal or duct cleaning is often the most effective treatment. Spielmann (1997) discussed the most widely used duct cleaning equipment and techniques as follows:

Planning and System Inspection. Drawings are used as a valuable planning tool. Duct cleaning is usually performed in zones of about 25 ft (7.5 m) or less. Contaminated ducts and uncleanable types of ducts, such as flexible ducts and certain lined ducts, that cannot be cleaned without damaging their duct liner should be identified. Inspection should be performed when the area is unoccupied and the air system is turned off.

Starting the Job. Turn off the air system. The return side is often 5 to 10 times dirtier than the supply side and is always cleaned first, before the supply side. Lay drop cloths to protect occupied areas. Duct cleaning is started from return grilles and outdoor air intakes. Existing openings should be used for access whenever possible. Additional access openings often must be cut. Cleaning zones are isolated by using inflatable bladders inserted into the ducts. Duct sections are cleaned along the ductwork toward the air-handling unit or air handler.

Agitation Devices. Contaminants are agitated by cleaning devices so that dirt and debris are loosened drom the duct wall and extracted by the airstream. There are three currently used agitation methods:

- Manual agitation using contact vacuuming requires physical access to all surfaces.
- Use portable compressed-air skipper nozzles to dislodge debris from the duct surfaces.
- Adopt a rotary brush or other cleaning head spinning on the end of a motor-driven flexible shaft. When using a rotary brush in a lined duct, do not allow the brush to remain in the same spot for too long.

Most of the duct cleaning work requires a combination of these agitation methods.

Duct Vacuum. A duct vacuum (negative air machine) is used to extract loosened debris into the airstream of negative pressure so that contaminants will not pollute the conditioned space. A duct

vacuum used in duct cleaning is often portable and consists of a blower, a HEPA filter to collect debris, an outer casing, and connecting flexible duct. A large duct vacuum often has an extracting volume flow rate of 4000 cfm (1.89 m^3/s) and more. For each cleaning zone, there are two openings : one is the access to the agitation device, and the other opening connects the duct under cleaning directly to the portable duct vacuum. A smaller portable duct vacuum is used to extract debris from the fan blades, coils, and condensate drip pans.

Sealing of Access Openings. After the duct cleaning is completed, the cleaned ducts should be isolated from the cleaning zone in which duct cleaning is progressing by means of inflatable bladder. The access opening should be patched using reusable doors, sheet metal, duct tape, and sealants.

17.16 PRESSURE AND AIRFLOW MEASUREMENTS

The total pressure of the airflow is the sum of the static pressure, which acts in all directions, and the velocity pressure, which results from the impact and the inertia of the airflow. The Pitot tube and manometers, shown in Figs. 17.29 and 17.30, are widely used to measure total pressure, static pressure, velocity pressure, and thus airflow inside air ducts.

A standard Pitot tube consists of two concentric tubes having an outside diameter $D = {}^5/_{16}$ in. (8 mm) and an inner tube of diameter of ${}^1/_8$ in. (3.2 mm), as shown in Fig. 17.29. The inner tube opens to the airflow at the nose, and the other end is used as the total pressure tap. Eight equally spaced holes of 0.04-in. (1-mm) diameter allow the air to flow into the hollow space between the outer and inner tubes.

The hollow space is connected to the static pressure tap. Because the small holes are located perpendicular to the centerline of the head, they are able to measure the static pressure when the head of the Pitot tube is placed in a position opposite to the direction of the airflow. The centerlines of the small holes are located at a distance of 8*D* from the nose and 16*D* from the stem. The negative pressure produced at the nose is nearly balanced by the positive pressure at the stem.

The U tube shown in Fig. 17.30*a* is the simplest type of manometer used to measure pressure in air ducts. The vertical difference in the liquid column indicates the pressure reading. For more

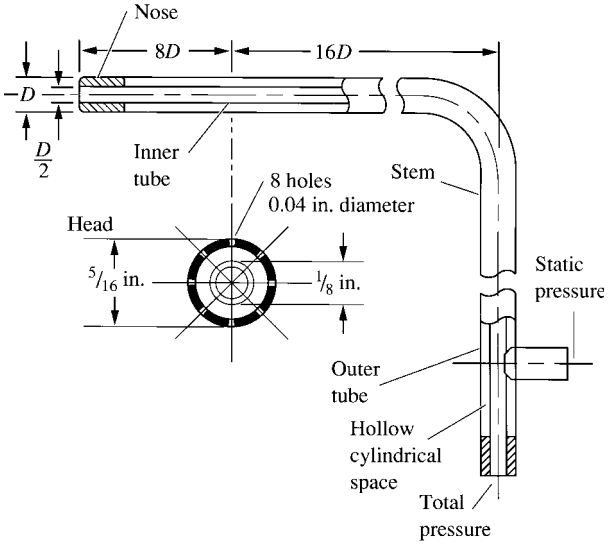


FIGURE 17.29 Pitot tube.

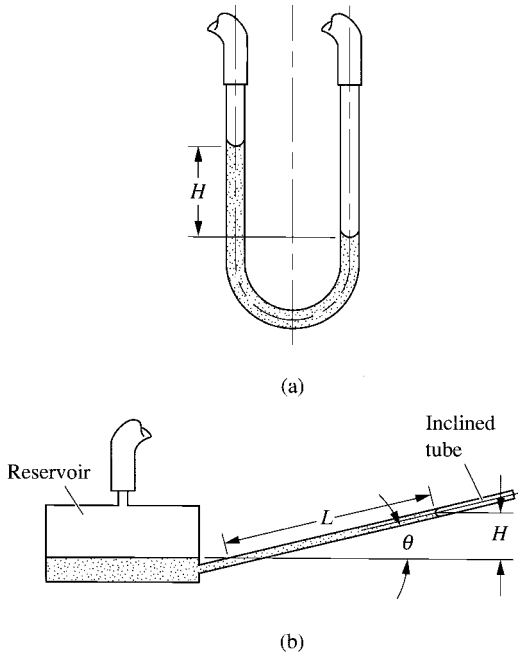


FIGURE 17.30 Manometers: (*a*) *U* tube; (*b*) inclined manometer.

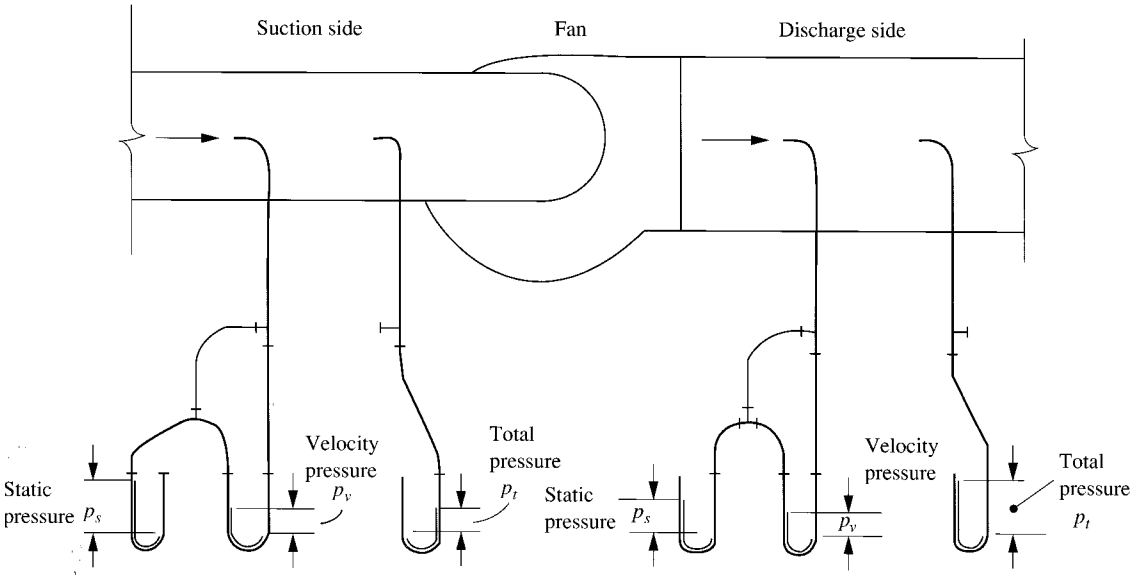


FIGURE 17.31 Pressure measurements in air ducts.

accurate measurement, an inclined manometer (see Fig. 17.30*b*) is often used. The relationship between the height of the liquid column H and the length of the magnified inclined scale is

$$H = L \sin \theta \tag{17.103}$$

where L = length of inclined scale with liquid column, ft (m)
θ = angle of inclination of inclined tube, deg

Because the cross-sectional area of the inclined tube is very small compared with that of the reservoir, the change in the liquid level of the reservoir can be ignored.

The following precautions should be taken when one takes pressure measurements in an air duct such as that shown in Fig. 17.31:

- The nose of the Pitot tube must always be placed opposite to the direction of airflow whether in the suction side or discharge side of the fan.
- When the total pressure or static pressure is measured, one leg of the U tube or inclined manometer must be open to the atmospheric air as the reference datum. The smaller pressure is always connected to the open end of the inclined tube of the inclined manometer.
- For a velocity pressure measurement, the total and static pressure taps must be connected to the two ends of the manometer. The total pressure tap is connected to the reservoir, and the static pressure tap is connected to the inclined tube. Velocity pressure is always positive in the direction of airflow.

Equal-Area Versus Log Tchebycheff Rule

Because velocity is usually not uniform across the cross-sectional area, a transverse is often used to determine the average velocity. The equal-area method was widely used before. Recently ASHRAE

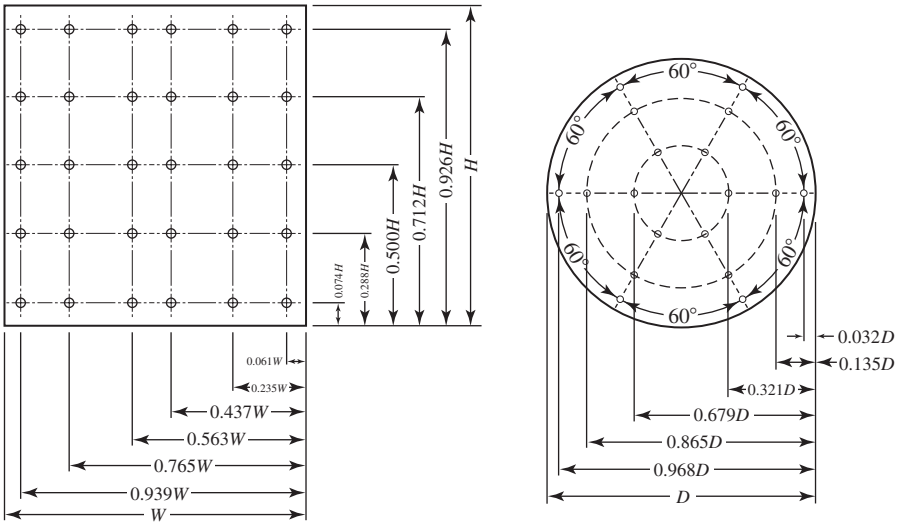


FIGURE 17.32 Measuring points in rectangular and round duct transverse: (*a*) log Tchebycheff rule for rectangular duct transverse; (*b*) log-linear rule for round duct transverse.

Standard 111-88, the Associated Air Balance Council (AABC), and the National Environmental Balancing Bureau (NEBB) all adopted the log Tchebycheff rule for rectangular duct and round duct transverse, as shown in Fig. 17.32*a* and *b*, because this rule provided greatest accuracy. The reason is that the location of transverse points of these rules has taken into account the effect of the duct wall friction and the reduction in velocity near the duct walls. For round duct transverse, the log Tchebycheff rule and log-linear transverse method are similar. Refer to ASHRAE Standard 111-88 for details. The log Tchebycheff rule was developed by a mathematician named Tchebycheff in 1977.

In MacFerran (1999), airflow in ducts measured by the equal-area method was often higher than correct results, from 7 percent higher maybe up to 25 percent higher.

When the velocity pressure is measured, in inches WC (Pa), the air velocity, in fpm, can be calculated from Eq. (17.10). For SI units, the air velocity, in m/s, can be calculated from Eq. (17.9), in which $g_c = 1$.

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CHAPTER 18
AIR SYSTEMS: SPACE AIR DIFFUSION

18.1 PRINCIPLES OF SPACE AIR DIFFUSION 18.2
- Draft and Effective Draft Temperature 18.2
- Air Diffusion Performance Index 18.3
- Space Diffusion Effectiveness Factor 18.4
- Ventilation Effectiveness and Air Exchange Effectiveness 18.4

18.2 AIR JETS 18.5
- Free Isothermal Jets 18.5
- Throw, Entrainment Ratio, and Characteristic Length 18.7
- Confined Air Jets 18.8
- Free Nonisothermal Jets 18.10

18.3 SUPPLY OUTLETS AND RETURN INLETS 18.11
- Grilles and Registers 18.11
- Ceiling Diffusers 18.12
- Slot Diffusers 18.14
- Nozzles 18.16
- Accessories for Supply Outlets 18.17
- Return and Exhaust Inlets 18.17
- Light Troffer Diffuser and Troffer-Diffuser Slot 18.19

18.4 MIXING FLOW 18.20
- Airflow Pattern and Space Air Diffusion 18.20
- Principles and Characteristics of Mixing Flow 18.20
- Mixing Flow Using High Side Outlets 18.21
- Mixing Flow Using Ceiling Diffusers 18.23
- Mixing Flow Using Slot Diffusers 18.24
- Mixing Flow Using Sill or Floor Outlets 18.24
- Stratified Mixing Flow 18.25

18.5 COLD AIR DISTRIBUTION 18.28
- Cold Air Distribution versus Conventional Air Distribution 18.28
- High Induction Nozzle Diffusers 18.28
- Characteristics of Cold Air Distribution Systems 18.29
- Case Study—Florida Elementary School 18.29
- Performance of Ceiling and Slot Diffusers 18.29
- Cold Air Distribution with Fan-Powered VAV Boxes 18.30
- Surface Condensation 18.30

18.6 DESIGN PROCEDURE OF MIXING FLOW AIR DIFFUSION SYSTEM 18.31
- Select the Type of Supply Outlet 18.31
- Volume Flow Rate per Outlet or per Unit Length 18.32
- Choose an Optimum Throw–Characteristic Length Ratio 18.33
- Determine the Design Characteristics of Slot Diffusers in Perimeter Zone 18.33
- Select the Specific Outlet from Manufacturer's Catalog 18.34
- Determination of Final Layout of Supply Outlets and Return Inlets 18.34

18.7 DISPLACEMENT FLOW AND UNIDIRECTIONAL FLOW 18.38
- Displacement Flow 18.38
- Unidirectional Flow 18.38
- Unidirectional Flow for Clean Rooms 18.39
- Ventilating Ceiling 18.40
- Ceiling Plenum and Supply Air Velocity 18.41

18.8 STRATIFIED DISPLACEMENT FLOW 18.42
- Two-Zone Stratified Model 18.42
- Operating Characteristics 18.42
- Comparison of Stratified Displacement Flow and Mixing Flow 18.43

18.9 PROJECTING FLOW—SPOT COOLING/HEATING 18.44
- Benefits of Projecting Flow 18.44
- Industrial Spot Cooling Systems 18.44
- Recommendations in Spot Cooling Design 18.46
- Desktop Task Conditioning Systems 18.46
- Performance of Desktop Task Conditioning Systems 18.47
- Application of Desktop Task Conditioning Systems 18.48

18.10 UPWARD FLOW UNDERFLOOR AIR DISTRIBUTION 18.48
- Upward Flow from Floor Plenum 18.48
- Thermal Storage of Floor Plenum 18.49
- Heat Unneutralized 18.50
- Maintaining a Consistent Access Plenum Temperature 18.50
- Floor Plenum Master Zone Air Temperature Control 18.50

Design Considerations 18.50
Applications of Upward Flow Underfloor Air Distribution 18.51
18.11 COMPUTATIONAL FLUID DYNAMICS 18.51
CFD Becomes More Popular 18.52
Reynolds-Averaged Navier-Stokes Equations 18.52
Numerical Methods 18.52
Conducting CFD Experiments 18.54
REFERENCES 18.54

18.1 PRINCIPLES OF SPACE AIR DIFFUSION

Space air diffusion distributes the conditioned air containing outdoor air to the occupied zone (or a given enclosure) in a conditioned space according to the occupants' requirements. An occupied zone is a space with the following dimensions: (width of room − 1 ft) × (depth of room − 1 ft) × (a height of 6 ft). A satisfactory space air diffusion evenly distributes the conditioned and outdoor air to provide a healthful and comfortable indoor environment for the occupants or the appropriate environment for a specific manufacturing process, at optimum cost.

Because space air diffusion is the last process of air conditioning and takes place entirely within the conditioned space, it directly affects the effectiveness of air conditioning. Because the diffused air and ambient air are transparent, space air diffusion is also difficult to trace.

Draft and Effective Draft Temperature

One of the most common complaints about air conditioned space is draft. *Draft* is defined as an unwanted local cooling of the human body caused by air movement and lower space air temperature. Recent developments by Fanger et al. (1988, 1989) demonstrate that air velocity fluctuations also create drafts, and they investigated the impact of turbulence intensity on the sensation of draft. Figure 18.1 depicts velocity fluctuations in a typical conditioned space. A new dimensionless parameter called the *turbulence intensity* I_{tur} is introduced and expressed as

$$I_{tur} = \frac{\sigma_v}{v_m} \tag{18.1}$$

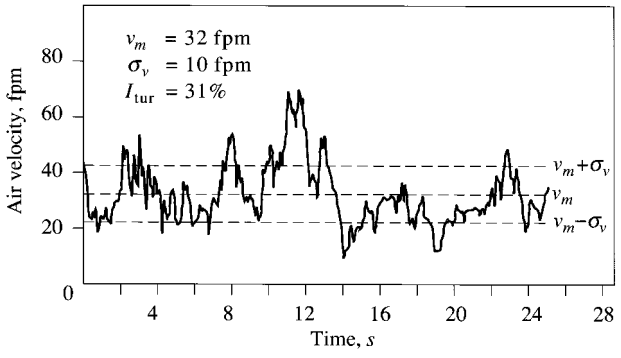

FIGURE 18.1 Fluctuations of air velocity in a typical air conditioned space. (*Adapted with permission from ASHRAE Journal, April 1989, p. 30.*)

where σ_v = standard deviation of air velocity fluctuations, fpm (m/s)
v_m = mean air velocity, fpm (m/s)

Experiments showed that if the space temperature T_x = 73.4°F (23°C) and v_m = 30 fpm (0.15 m/s), the percentage of dissatisfied occupants may increase from about 10 to 15 percent when I_{tur} increases from 0.1 to 0.5. A parameter called the effective draft temperature θ, in °F (°C), which combines the effects of uneven space air temperature and air movement, is often used to assess the deviations of local magnitudes from the mean value, and it is defined as

$$\theta = T_x - T_r - a(v_x - v_{rm}) \tag{18.2}$$

where T_x, v_x = space air temperature and velocity at specific location, °F (°C) and fpm (m/s)
T_r = mean space air temperature or set point, °F (°C)

In Eq. (18.2), a is a conversion constant to combine the effects of space air temperature and air movements; its value is 0.07 when T is expressed in °F and v is expressed in fpm, and 8 when T is expressed in °C and v in m/s.

The desirable mean space air velocity v_{rm}, in fpm (m/s), is closely related to the space air temperature T_r to be maintained, the metabolic rate, and the clothing insulation of the occupant. According to Fig. 4.8, for a space relative humidity φ = 50 percent, $T_{rad} = T_r$, a metabolic rate of 400 Btu/h (117.2 W), and a clothing insulation of 0.6 clo (0.093 $m^2 \cdot °C/W$), their relationship may be given as follows:

T_r, °F	70	72	74	76	78	80
v_{rm}, fpm	16	20	25	32	40	55

Air Diffusion Performance Index

To an extent, the magnitude of effective draft temperature θ also reflects the degree of thermal comfort that can be provided by comfort air conditioning systems. During cooling mode in commercial and public buildings, if the space temperature is maintained between 75 and 78°F (23.9 and 25.6°C), $T_{rad} = T_r$, space air velocity $v_r < 55$ fpm (0.28 m/s), and the space relative humidity is between 30 and 70 percent, then most sedentary occupants feel comfortable when $-3°F < \theta < +2°F$ ($-1.7°C < \theta < +1.1°C$). The air diffusion performance index (ADPI), in percent, which evaluates the performance of space air diffusion, is calculated as

$$\text{ADPI} = \frac{N_\theta \times 100}{N} \tag{18.3}$$

where N_θ = number of points measured in occupied zone in which $-3°F < \theta < +2°F$ ($-1.7°C < \theta < +1.1°C$)
N = total number of points measured in occupied zone

The higher the ADPI, the higher the percentage of occupants who feel comfortable. Maximum ADPI approaches 100 percent.

Because air velocity fluctuations create drafts, therefore, they also affect ADPI. Abu-Ei-Hassan et al. (1996), based on the impact of turbulence intensity Tu recommended by Fanger et al. (1988), conducted tests to determine the effect of turbulence intensity Tu (velocity fluctuations) on the ADPI. Abu-Ei-Hassan et al. took the centerline velocity of the test room, which has nearly the same flow characteristic as the centerline velocity of the air jet, as the air velocities of the local points, and the ADPIs thus calculated as Air Diffusion Performance Index that take into consideration the Tu impact. The results showed that for a sidewall outlet space air diffusion system with a room

cooling load intensity of 10 to 20 Btu/h·ft^2 (31.5 to 63 W/m^2) the Air Diffusion Performance Index considering the turbulence effect were about 10 percent less than those that did not take the impact of Tu into account. At higher cooling loads and airflow rates, the effect of Tu became larger. Actually, the drop in ADPI is less than 10 percent for cooling load intensity between 10 and 20 Btu/h·ft^2 (31.5 and 63 W/m^2) due to the impact of Tu because the points of air velocity located on the centerline of the test room are only a small portion of the total number of measured points.

In cooling mode operation, ADPI is an appropriate index to use to evaluate a space air diffusion system. For heating mode operation, the temperature gradient between two points in the occupied zone may be a better indicator of the thermal comfort of the occupants and the effectiveness of a space air diffusion system. Usually, the temperature gradient is less than 5°F (2.8°C) within an occupied zone.

Space Diffusion Effectiveness Factor

The effectiveness of a space air diffusion system can also be assessed by using a space diffusion effectiveness factor for air temperature ε_T or for air contamination ε_C. Both factors are dimensionless. Effectiveness factor ε_T compares temperature differentials and ε_C compares contamination differentials as follows:

$$\varepsilon_T = \frac{T_{re} - T_s}{T_r - T_s} = \frac{T_{ex} - T_s}{T_r - T_s}$$
$$\varepsilon_C = \frac{C_{ex} - C_s}{C_r - C_s} \tag{18.4}$$

where T = temperature, °F (°C)
C = concentration of air contamination, μg/m^3

In Eq. (18.4), subscript re represents the recirculating air, ex the exhaust air, r the space air or air at the measuring point, and s the supply air. When $\varepsilon \geq 1$, the space air diffusion is considered effective. If $\varepsilon < 1$, a portion of supply air has failed to supply the occupied zone and exhaust through the return or exhaust inlets directly. Parameters in both the numerator and denominator must be in the same units.

Ventilation Effectiveness and Air Exchange Effectiveness

ASHRAE Handbook 1997, Fundamentals, defines ventilation effectiveness as a description of an air distribution system's ability to remove internally generated contaminants from a zone, space, or building. Air change effectiveness ε_{ex} is defined as air system ability to deliver ventilation air to a zone, space, or building. The age of air θ_{age}, in minutes or hours, is the time period that outdoor ventilation air has been in a zone, space, or building, and age can be evaluated for an existing building using the trace gas method. Chamberlin et al. (1999) proposed the following when growth (step-up) and decay (step-down) tracer methods are used:

$$\theta_{age} = (\theta_{end} - \theta_{initial})\left(1 - \frac{C_{avg}}{C_{end}}\right)$$
$$= (\theta_{stop} - \theta_{start})(C_{avg} - C_{start}) \tag{18.5a}$$

where C_{avg} = time-averaged concentration of tracer gas measured in test zone from when tracer gas injections began to when last sample is taken, μg/m^3 or ppm
C_{end}, C_{start} = concentration of tracer gas when last sample was taken at time θ_{end} and when first sample was taken at θ_{start}, μg/m^3 or ppm

$\theta_{end} - \theta_{initial}$ = time elapsed from when tracer gas injection began to when last sample was taken, min or h
$\theta_{stop} - \theta_{start}$ = time elapsed from when first sample was taken to when tracer gas injection stopped, min or h

The youngest air is the freshest air through infiltration. A longer age means a poorer outdoor air delivery.

Another measure of ventilation is the nominal time constant τ_N, in min or h, and it can be calculated as

$$\tau_N = \frac{\dot{V}_s}{V_r} \tag{18.5b}$$

where $\dot{V}_s$ = supply volume flow rate to zone, space, or room, cfm (m^3/s)
V_r = volume of zone, space, or room, ft^3 (m^3)

Air exchange effectiveness is a relative index to measure how well the outdoor air is diffused to various locations in the occupied zone. Nominal air change effectiveness ε_N can be calculated as

$$\varepsilon_N = \frac{\tau_N}{\theta_{age,N}} \tag{18.6}$$

and $\theta_{age,N}$ is evaluated through tracer gas measurement. Normally, for a properly designed, installed, operated, and maintained air distribution system, $\varepsilon_N \approx 1$.

18.2 AIR JETS

An air jet is an airstream that discharges from an outlet with a significantly higher velocity than that of the surrounding air, and moves along its centerline until its terminal velocity reduces to a value that equals or approximately equals the velocity of the ambient air. Because of the turbulence of air particles, air jets tend to spread. They also rise or fall depending on the buoyancy of the airstream.

The outer boundary of an air jet where air is moving at a perceptible velocity, such as 150 fpm (0.75 m/s), 100 fpm (0.5 m/s), or 50 fpm (0.25 m/s), is called the *envelope*. In general, air jets can be classified as free or confined, isothermal or non isothermal, and axial or radial. A free air jet is an ideal air jet whose envelope (outer boundary) is not confined by the enclosure of the conditioned space. A confined air jet's envelope is confined by the ceiling, floor, walls, windows, and furniture of the conditioned space.

According to experimental results, the characteristics of an air jet approach those of a free air jet when the relationship $\sqrt{A_r}/D_o > 50$ is satisfied. Here A_r represents the cross-sectional area of the enclosure perpendicular to the centerline of the air jet, in ft^2 (m^2), and D_o indicates the diameter or the circular equivalent of the supply outlet, in ft (m).

An isothermal air jet is one whose temperature is equal or nearly equal to the temperature of the ambient air. A nonisothermal air jet is one whose temperature is different from that of the ambient air in the conditioned space. An axial air jet projects in one direction, and a radial air jet projects radially in all directions.

Free Isothermal Jets

Along the centerline of a free isothermal jet, four zones can be identified, as shown in Fig. 18.2:

1. *Core zone*. In the core zone, the centerline velocity remains unchanged. This zone extends about $4D_o$ from the surface of the outlet.

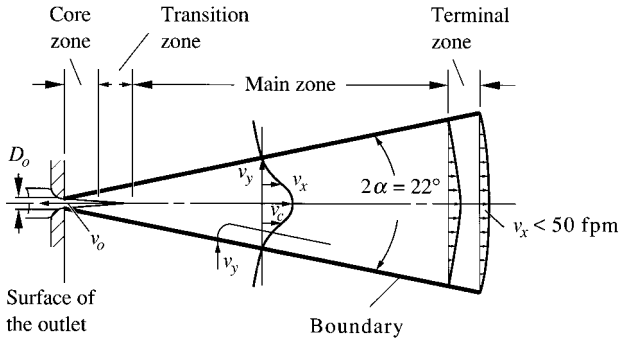

FIGURE 18.2 Four zones of a free, isothermal, axial air jet.

2. *Transition zone*. In the transition zone, the centerline velocity decreases inversely with the square root of the distance from the surface of the outlet. This zone extends about $8D_o$.
3. *Main zone*. In the main zone, turbulent flow is fully developed, and the maximum velocity decreases inversely with the distance from the surface of the outlet. Even when the air jet is discharged from a rectangular outlet, the cross section of the airstream becomes circular in the main zone. This zone extends about 25 to $100D_o$ in length.
4. *Terminal zone*. In the terminal zone, the maximum air velocity decreases rapidly to a value less than 50 fpm (0.25 m/s) within a distance of a few outlet diameters.

In core, transition, and main zones, the fluctuating velocity components transport momentum across the boundary. Therefore, ambient air is induced into the air jet, and the airstream diverges to a greater spread either vertically or horizontally. The angle of divergence 2α of a free isothermal jet discharged from a nozzle is about 22°. For an air jet discharged from a slot, its angle of divergence 2α perpendicular to the slot is about 33°. If the guide vanes at the outlet are deflected at an angle from the straight position, the spread of the air jet is greater.

The velocity profiles at different cross-sectional planes perpendicular to the airflow in the main zone of a free isothermal jet are similar to one another. At any specific cross-sectional plane, the velocity profile can be approximated by the formula

$$\left(\frac{R}{R_{0.5}}\right)^2 = 3.3 \log \frac{v_c}{v} \tag{18.7}$$

where v = air velocity at distance R from centerline of air jet, fpm (m/s)
v_c = centerline velocity, fpm (m/s)
R = radial distance from centerline of air jet, ft (m)
$R_{0.5}$ = radial distance from centerline of air jet to a point where velocity equals $0.5v_c$, ft (m)

Research shows that the law of conservation of momentum can be assumed in the main zone of a free air jet. The application of this law to determine the centerline velocity of a free air jet gives the following results:

$$\frac{v_c}{v_o} = \frac{K}{x/D_o} = \frac{K'}{x/\sqrt{A_o}} = \frac{K}{x/H_o} \tag{18.8}$$

where K, K' = centerline velocity constants, depending mainly on type of outlets; normally, $K' = 1.13K$
H_o = width of radial air jet at outlet or vena contracta, ft (m)
v_o = mean velocity at vena contracta, fpm (m/s)

x = distance from surface of outlet to a cross-sectional plane having centerline velocity v_c, ft (m)

In Eq. (18.8), A_o, in ft^2 (m^2), represents the effective area of the airstream, i.e., the minimum area at the vena contracta. It can be calculated as

$$v_o = v_{\text{core}} C_d R_{\text{fa}}$$
$$A_o = A_c C_d R_{\text{fa}} \tag{18.9}$$

where A_c = core area of outlet, i.e., surface area of opening, ft^2 (m^2)
v_{core} = face velocity at core of outlet, fpm (m/s)
C_d = discharge coefficient, usually between 0.65 and 0.9
R_{fa} = ratio of free area to gross area (free area is net area of opening through which air can pass)

When v_o is 500 to 1000 fpm (2.5 to 5 m/s), for round free openings, $K = 5$; for rectangular openings and linear slot diffusers, $K' = 4.9$. When v_o is 2000 to 5000 fpm (10 to 25 m/s), for round free openings, $K = 6.2$.

When multiple air jets are discharged into a conditioned space at the same level, each air jet behaves independently until the jets meet. From the point where the jets meet, the velocities between the centerlines of the air jets increase until they are equal to the centerline velocities of the air jets.

Throw, Entrainment Ratio, and Characteristic Length

Throw in ft (m), is defined as the horizontal or vertical axial distance from the outlet to a cross-sectional plane where the maximum velocity of the airstream at the terminal zone has been reduced to 50 fpm (0.25 m/s), 100 fpm (0.5 m/s), or 150 fpm (0.75 m/s). The throw is indicated by T_v, and the subscript denotes the terminal velocity for which the throw is measured. For instance, T_{50} ($T_{0.25}$) indicates the throw with a terminal velocity of 50 fpm (0.25 m/s).

From Eqs. (18.8) and (18.9), throw can be calculated as

$$T_v = \frac{K' \dot{V}_s}{v_{t,\,max} \sqrt{A_c C_d R_{\text{fa}}}} \tag{18.10}$$

where $v_{t,\text{max}}$ = maximum velocity of airstream at terminal zone, fpm (m/s)
$\dot{V}_s$ = supply volume flow rate at outlet, cfm (m^3/s)

For a specific configuration of supply outlet, throw T_v depends on both the supply volume flow rate $\dot{V}_s$ and supply outlet velocity v_{core}.

The entrainment ratio R_{en} is the ratio of the volume flow rate of the total air at a specific cross-sectional plane of the air jet $\dot{V}_x$ to the volume flow rate of the supply air $\dot{V}_o$ discharged from the outlet ,which is sometimes called primary air. Total air is the sum of supply air and induced air.

The entrainment ratio is proportional to the distance or square root of the distance from the outlet. For circular jets in the main zone, the entrainment ratio can be calculated as

$$R_{\text{en}} = \frac{\dot{V}_x}{\dot{V}_o} = \frac{2x}{\text{K}\sqrt{\text{A}_\text{o}}} = \frac{2\text{v}_\text{o}}{\text{v}_\text{c}}$$

For a long slot, the entrainment ratio is

$$R_{\text{en}} = \frac{\dot{V}_x}{\dot{V}_o} = \sqrt{\frac{2x}{K' H_\text{o}}} = \sqrt{\frac{2v_\text{o}}{v_\text{c}}} \tag{18.11}$$

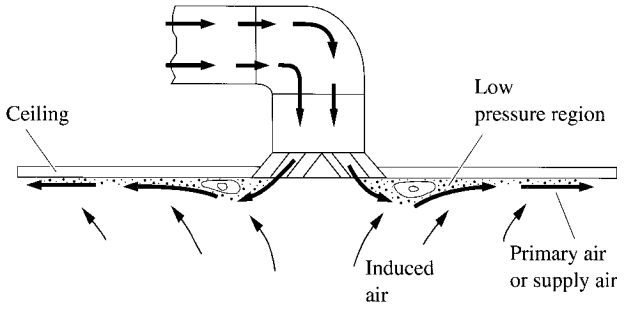

FIGURE 18.3 Surface effect.

The characteristic length L, in ft (m), is the horizontal distance from the surface of the outlet to the nearest vertical opposite wall or the horizontal distance from the surface of the outlet to the midplane between two outlets in the direction of airflow, or the distance to the closest intersection of air jets. The ratio of throw to characteristic length T_v/L is related to the ADPI of various supply outlets and has been used as a parameter in space diffusion design.

Confined Air Jets

In actual practice, most air jets are confined by the boundary of the room or the conditioned space. For a confined air jet, the total momentum of the fluid elements decreases gradually as distance from the outlet increases because of the friction between the airstream and the boundary.

Surface Effect. When a primary airstream discharged from a supply outlet flows along a surface, the velocity of the primary airstream is significantly higher than that of the ambient air, and a lower-pressure region is formed near the surface along the airflow, as shown in Fig. 18.3. Consequently, in-

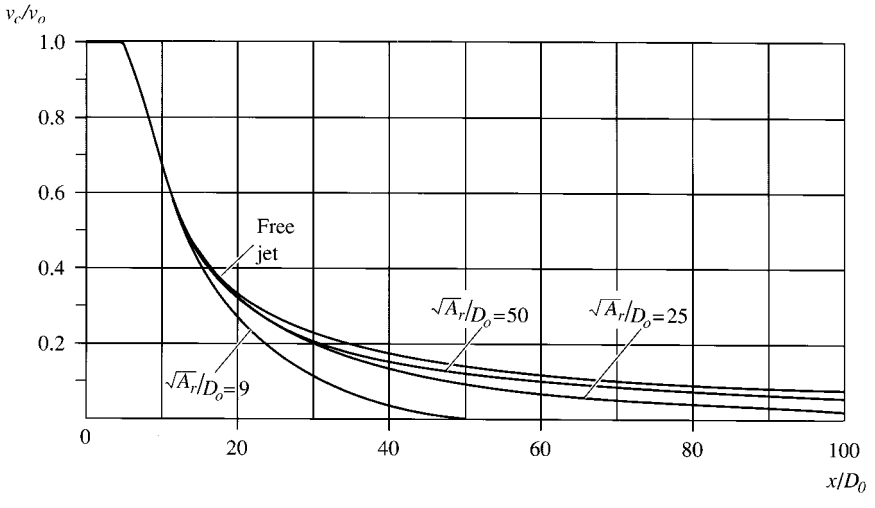

FIGURE 18.4 Centerline velocities of confined air jets.

duced ambient air at a comparatively higher pressure presses the air jet against the surface, even when it is a curved surface. Such a phenomenon is called the surface effect or the Conda effect.

Friction between the air jet and the boundary decreases the centerline velocity of confined air jets, as shown in Fig. 18.4. However, because of the surface effect, the throw of a confined air jet is longer, and the drop from the horizontal axis smaller, than that of a free air jet.

Airflow Pattern. The flow pattern and characteristics of a confined air jet using a sidewall outlet were introduced by Russian scientists during the 1950s. In an air conditioned room whose supply outlet is located above the occupied zone and whose exhaust opening is on the same side of the supply outlet, the supply air jet clings to the surface of the ceiling and mixes with the room air. An induced reverse airstream, with more even velocity and temperature distribution than that of the air jet, covers the occupied zone.

Figure. 18.5 shows the airflow pattern of a typical confined isothermal jet. When supply air is discharged from the circular outlet and moves along the surface of the ceiling, the fluctuating velocity components continue to transport momentum across the boundary of the air jet. Therefore, ambient air is induced into the air jet, and the induced circulating airflow occupies most of the enclosed space.

As the air jet moves forward, its mass flow rate increases and mean air velocity decreases until it arrives at a cross-sectional plane where dimensionless distance $s = 0.22$. Term s is defined as

$$s = \frac{ax}{\sqrt{0.5A_r}} \tag{18.12}$$

where a = turbulence factor; for a circular nozzle $a = 0.076$, and for a rectangular outlet without guide vanes $a \approx 0.15$.

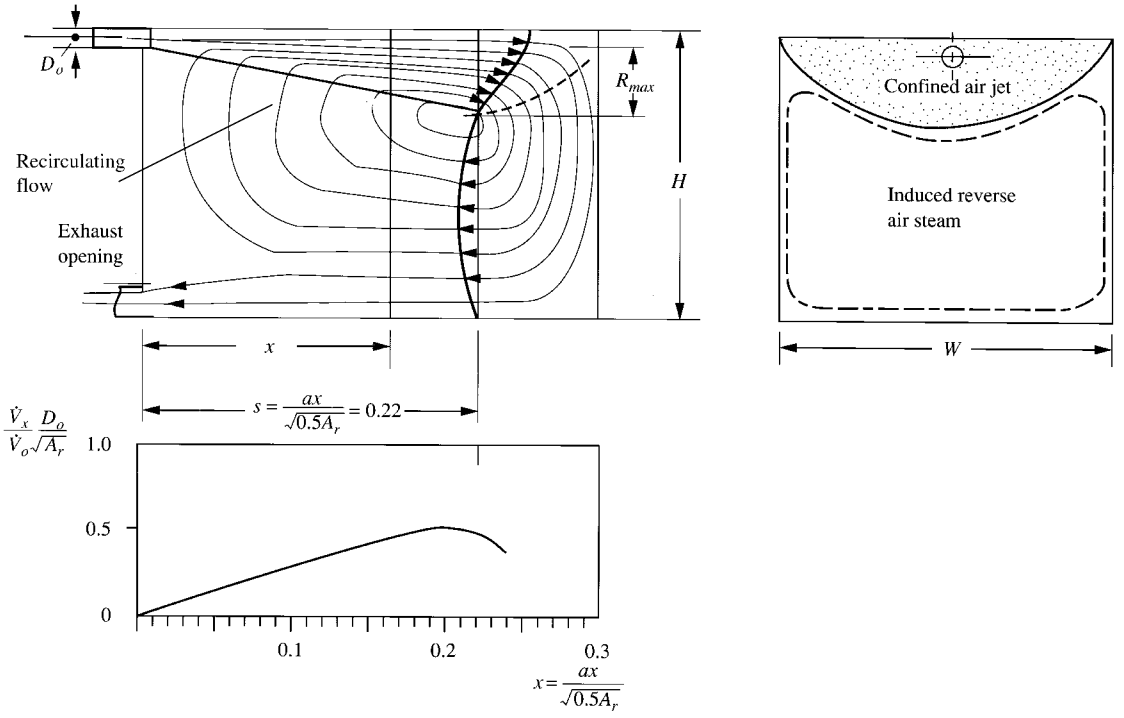

FIGURE 18.5 Airflow pattern of a typical confined air jet.

The air jet terminates beyond $s = 0.22$, and the airstream makes a 180° turn, forming a reverse airstream flowing in the opposite direction. The majority of the reverse airstream turns upward and is induced by the air jet again. Only a portion of it is exhausted outside the room.

The angle of divergence, the velocity profile, and the calculation of the entrainment ratio of confined jets are similar to those of free jets. Based on the principle of continuity of mass at steady state, other characteristics of confined air jets can be summarized as follows:

- If there is no infiltration into or exfiltration from the room, the mass flow rate of supply air is exactly equal to the exhaust air.
- If there are no obstructions in the room, the ideal stream lines of the induced air are closed curves and form recirculating flow.
- If the supply outlet and exhaust inlet are located on the same side of the room, at any cross-sectional plane perpendicular to the horizontal airflow, the mass flow rates of airstreams flowing at opposite directions must be equal.
- The volume flow rate of induced air is equal to or several times greater than that of supply air at the cross-sectional plane where $s = 0.22$. The characteristics of airstreams in the occupied zone depend mainly on the induced reverse airstream. The volume flow rate and the air velocity of the reverse airstream are highest at the cross-sectional plane where $s = 0.22$.

Free Nonisothermal Jets

When the temperature of the conditioned air discharged from a supply outlet is different from that of the ambient air in a conditioned space, the buoyancy of the fluid particles causes the trajectory of the air jet to deviate from the axis of the free isothermal jet. A cold air jet will descend, and a warm air jet will ascend. Figure 18.6 shows a free cold air jet discharged horizontally from a nozzle.

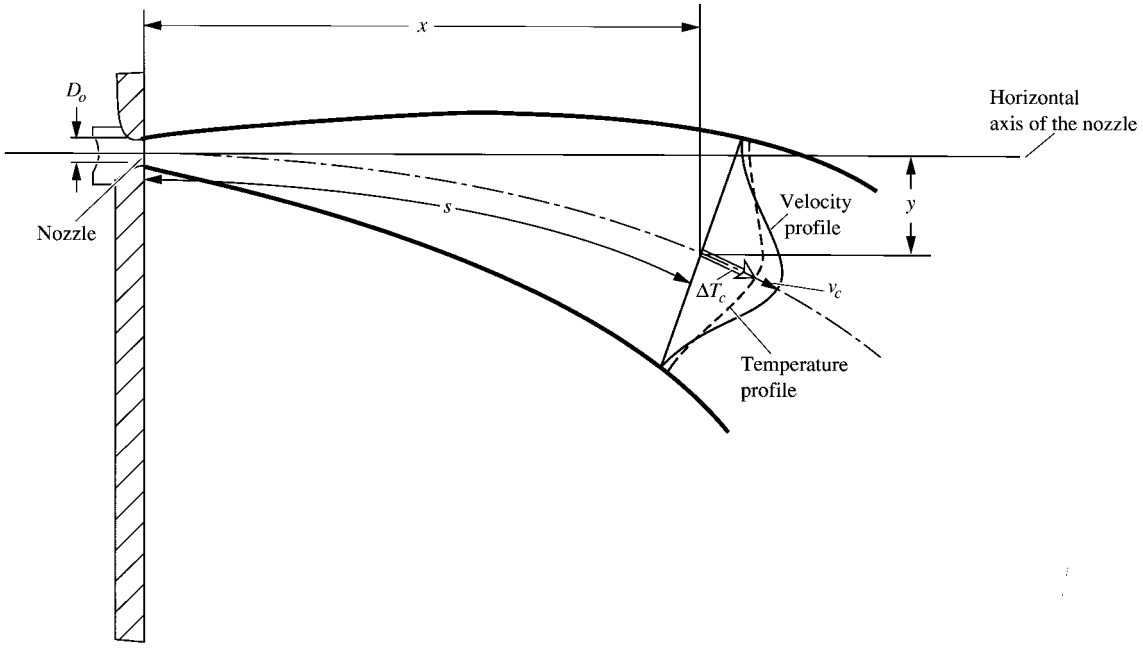

FIGURE 18.6 Path of a cold air jet discharged horizontally from a nozzle.

In Fig. 18.6, x is the horizontal distance between a cross-sectional plane in a cold air jet and the surface of the outlet, and y is the drop of the cold air jet, i.e., the vertical distance between the horizontal axis of the nozzle and the center of the specific cross-sectional plane. The velocity profile at the specific cross-sectional plane is indicated by the solid line, and the temperature profile is indicated by the dotted line.

According to the experimental results of Kostel (1955), the following empirical formula can be used to determine the vertical drop of a cold air jet and the vertical rise of the warm air jet discharged from a nozzle:

$$\frac{y}{\sqrt{A_o}} = \frac{x}{\sqrt{A_o}} \tan\alpha + K\mathrm{Ar}\left(\frac{x}{\sqrt{A_o}} \cos\alpha\right)^3 \tag{18.13}$$

where α = angle between centerline of nozzle and horizontal axis of nozzle, deg
K = constant

Also Ar, known as Archimedes' number, indicates the buoyant forces of the air jet and is given as

$$\mathrm{Ar} = \frac{g\sqrt{A_o}}{v_o^2} \frac{\Delta T_o}{T_{rR}} \tag{18.14}$$

where g = gravitational acceleration, ft/s^2 (m/s^2)
ΔT_o = temperature difference between supply air at outlet and ambient air in conditioned space, °F (°C)
T_{rR} = absolute temperature of space air, °R (K)
v_o = air velocity at supply outlet, f/s (m/s)

For free air jets, Kostel (1955) determined that $K = 0.065$.

For a nonisothermal jet, the relationship between the decay of the centerline velocity and the centerline temperature difference ΔT_c, in °F (°C), can be shown as

$$\frac{\Delta T_c}{\Delta T_o} = \frac{T_c - T_r}{T_s - T_r} = \frac{0.8 v_c}{v_o} \tag{18.15}$$

where subscript c represents centerline, o the outlet, and r the space. The characteristics of isothermal radial air jets are similar to those of axial air jets. When determining the centerline velocity, use x/H_o for radial jets instead of $x/\sqrt{A_o}$ for axial jets.

18.3 SUPPLY OUTLETS AND RETURN INLETS

The proper type of supply outlet for a conditioned space is largely determined by the architectural setup of the room, the airflow pattern needed, the indoor environmental requirements, and the load conditions. Five types of supply outlets are currently used: grilles and registers, ceiling diffusers, slot diffusers, light troffer diffusers, and nozzles. A window sill outlet is actually a type of grille mounted at the top of a fan coil.

Grilles and Registers

A grille or grill, as shown in Fig. 18.7*a*, is an outlet for supply air or an inlet for return air or exhaust air. A register is a grille with a volume control damper. Figure 18.7*b* shows a single-deflection grille. It consists of a frame and one set of adjustable vanes. Either vertical or horizontal vanes may be used as face vanes to deflect the airstream.

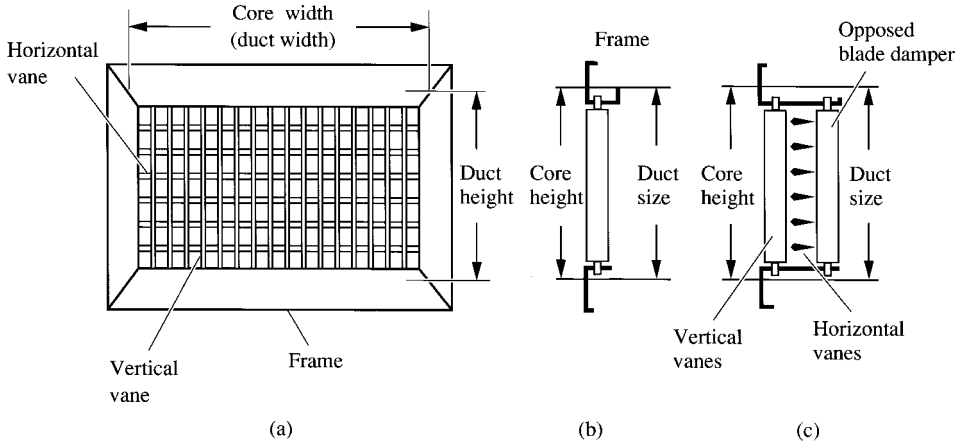

FIGURE 18.7 Supply grille and register: (*a*) grille, front view; (*b*) single-deflection, vertical vanes; (*c*) double-deflection register, with vertical and horizontal vanes.

Figure 18.7*c* shows a double-deflection register. It is able to deflect the airstream both horizontally and vertically. A volume damper is also used to adjust the volume flow through the register. Usually extruded aluminum vanes, aluminum or steel frames, and steel dampers are used. A baked enamel finish gives the grille an attractive appearance.

Grilles have a comparatively lower entrainment ratio, greater drop, longer throw, and higher air velocities in the occupied zone than slot and ceiling diffusers.

In the manufacturer's catalog, the performance of a grille or register is defined by the following parameters:

- Core size or core area A_c, which indicates the total plane area of an opening, ft^2 (m^2)
- Volume flow rate $\dot{V}$, cfm (m^3/s, or L/s)
- Air velocity $v = \dot{V}/A_k$, fpm (m/s)
- Total pressure loss Δp_t, in. WC (Pa)
- Throw at terminal velocities of 50 fpm (0.25 m/s), 100 fpm (0.5 m/s), and 150 fpm (0.75 m/s) with horizontal vanes deflected at 0°, 22.5°, and 45°
- Noise criteria curve

Here A_k is the area factor representing the net or corrected area, in ft^2 (m^2), which is the unobstructed area of a grille, register, or diffuser measured at the vena contracta.

For Air Diffusion Council–certified performance data, throw is tested on isothermal jets in order to avoid the influence of buoyancy forces at different supply temperature differential ΔT_s. During testing, the difference between supply and room air temperatures cannot exceed $\pm$ 2 °F (1.11°C).

Ceiling Diffusers

A ceiling diffuser consists of a series of concentric rings or inner cones made up of vanes arranged in fixed directions and an outer shell or frame, as shown in Fig. 18.8*a*. Ceiling diffusers can be round, square, or rectangular. Square diffusers are most widely used. Supply air is discharged through the concentric air passages or directional passages in one, two, three, or in all directions by using different

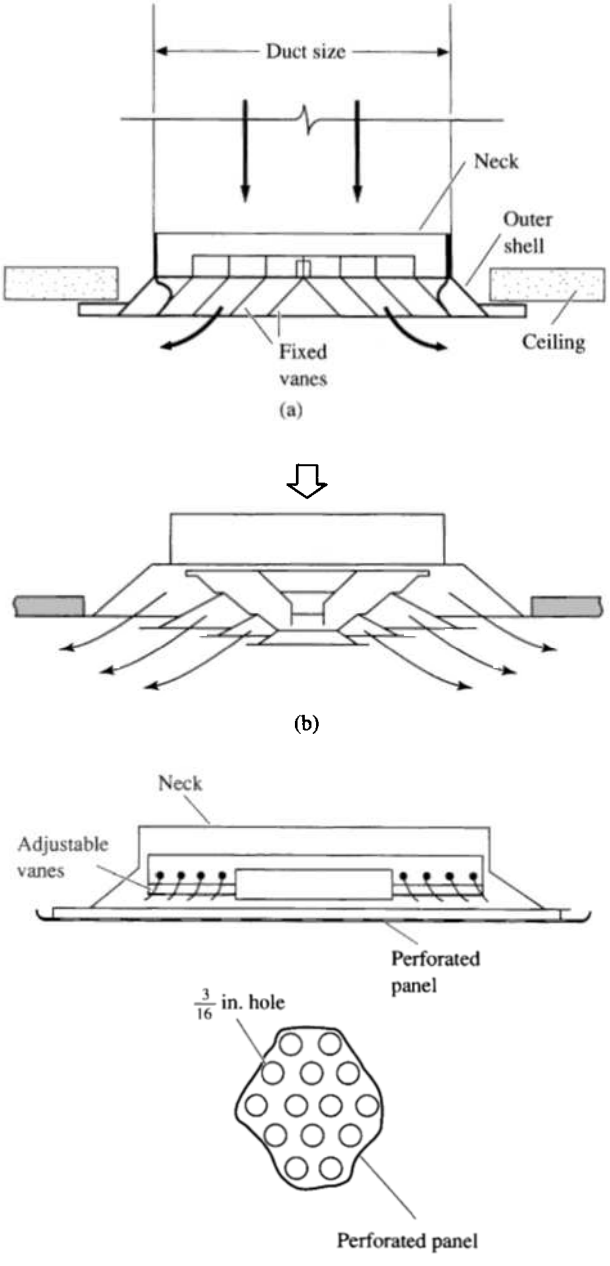

FIGURE 18.8 Ceiling diffusers: (*a*) square and rectangular ceiling diffusers; (*b*) removable inner-core ceiling diffuser; (*c*) perforated ceiling diffuser.

types of inner cone and vanes. The air discharge pattern of many ceiling diffusers may be changed from horizontal to vertical by means of adjustable inner cones or special deflecting vanes. Ceiling diffusers are often mounted at the center of the conditioned space and discharge air in all directions.

An important characteristic of a ceiling diffuser is its induction effect. Induction is the volume flow rate of space air induced into the outlet by the primary airstream. It is also called aspiration. The ratio of the volume flow rate of induced air to the volume flow rate of supply air $\dot{V}_{\text{ind}}/\dot{V}_s$, both in cfm (L/s), is called the induction ratio. A ceiling diffuser with a higher induction ratio (see Fig. 18.8*b*) is beneficial for a higher supply temperature differential ΔT_s for cold air distribution in an ice storage system, which is discussed later in Sec. 18.5 and in Chap. 31.

Figure 18.8*c* shows a perforated ceiling diffuser. A perforated surface increases the induction effect. It is also used to match the appearance of the ceiling tile. The diameter of the holes on the perforated panel is typically $\frac{3}{16}$ in. (4.8 mm). Some ceiling diffusers are designed to discharge air evenly in all directions so that the surrounding ceiling surfaces are less likely to be smudged.

The size of a ceiling diffuser is determined by its neck size and is therefore closely related to the neck velocity of the diffuser. Ceiling diffusers are made of an outer shell and a removable inner core. The opposed-blade damper with an equalizing device of parallel blades mounted within a frame is often used to adjust the volume flow and to provide a more uniform airflow.

Ceiling diffusers are usually made of aluminum-coated heavy-gauge steel, extruded aluminum, or teak wood. In addition to high induction ratios, ceiling diffusers have a good surface effect and a shorter throw and can supply air in all directions. Horizontally projected ceiling diffusers are suitable for conditioned spaces with low headroom.

The performance data of a ceiling diffuser are defined by the following parameters:

- Neck velocity v_{neck}, that is, mean air velocity at the neck of the ceiling diffuser, fpm (m/s)
- Volume flow rate $\dot{V}_s = A_k\, v_s$, in cfm (L/s); area factor A_k, in ft^2 (m^2), is determined at the testing laboratory, and the average supply velocity v_s is usually measured at a specified position at the outlet by a deflecting vane anemometer, often called a velometer
- Total pressure loss Δp_t, in in. WC (Pa)
- Throw, ft (m)
- Noise criteria curve

Slot Diffusers

A slot diffuser consists of a plenum box with single or multiple slots and air deflecting vanes, as shown in Fig. 18.9. The slots are available in 0.5-in. (13-mm), 0.75-in. (19-mm), 1-in. (25-mm), and 1.5-in. (38-mm) widths. The 0.75- and 1-in. slot widths are most commonly used. Slots are typically 2, 3, 4, 5, or 6 ft (600, 900, 1200, 1500, or 1800 mm) long. Air discharged from a slot diffuser can be projected horizontally or vertically. With a single-slot diffuser, air is always discharged in one direction. With multiple slots, air can be horizontally discharged either left or right, or a combination of both, or one slot can discharge vertically while another discharges horizontally. Slot diffusers are often called T-bar diffusers, as they are compatible with standard ceiling T bars.

Air enters the plenum box from the round inlet at one side. The function of the plenum box is to distribute the air more evenly at the slot. The plenum box is sometimes insulated internally. After passing the deflecting vanes, the air is discharged from the slot horizontally (left or right) or vertically according to the required airflow pattern. Figure 18.9 shows the airstreams discharged from slot diffusers in various directions. Different manufacturers use their own patented air deflecting vanes. Deflecting vanes can also be used to block the airflow entirely if necessary.

Extruded aluminum is widely used as the material for the structural member of the slot, and a galvanized sheet is used for the plenum box. The performance of a slot diffuser is defined by the following parameters:

- Width of slot, number of slots, and length of slots

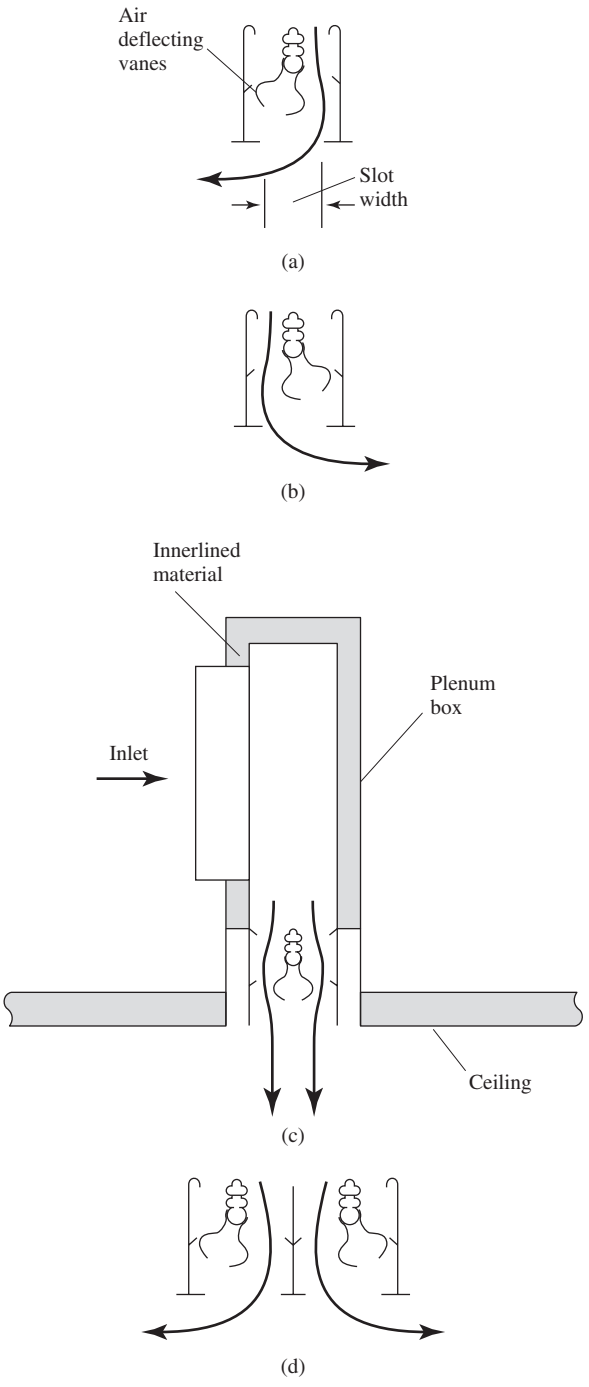

FIGURE 18.9 Slot diffusers: (*a*) horizontal left; (*b*) horizontal right; (*c*) vertical; (*d*) two-slot, two-way.

TABLE 18.1 Performance Data for Typical 0.75-in. (19-mn) Width Slot Diffuser with Plenum

Slots	Total pressure, in.WC		.004	.015	.032	.058	.091	.125	.175	.230
		V	.003	.011	.024	.044	.067	.095	.125	.170
1	Cfm/Ft		6	12	18	24	30	36	42	48
	Throw, ft	*H*	1,1,3	1,3,9	3,6,11	5,9,13	8,10,15	9,11,16	10,12,17	11,13,19
		V	2	6	10	12	14	15	16	17
	NC		—	—	—	18	24	30	35	39
2	Cfm/Ft		12	24	36	48	60	72	84	96
	Throw, ft	*H*	1,1,5	2,5,13	5,10,16	9,13,19	11,15,21	13,16,23	14,17,25	15,19,27
		V	4	9	14	17	19	21	23	24
	NC		—	—	13	23	29	35	40	44
3	Cfm/Ft		18	36	54	72	90	108	126	144
	Throw, ft	*H*	1,2,8	4,8,16	7,12,20	11,16,23	14,18,26	16,20,28	18,22,30	19,23,32
		V	6	11	17	21	23	26	28	30
	NC		—	—	16	26	32	38	43	47

Source: TITUS. Reprinted by permission.

- The volume flow rate per ft (m) of slot, or slot intensity, in cfm/ft (L/s· m)
- Throw, total pressure loss, and noise criteria curve

Table 18.1 lists the performance data of a typical slot diffuser of 0.75-in. (19-mm) slot width, taken from a manufacturer's catalog.

Nozzles

A nozzle is a round supply outlet (see Fig. 18.10). The airstream discharged from a nozzle is contracted just before the outlet, which results in a higher flow velocity and more even distribution. The purpose of using a nozzle instead of other types of supply outlets is to provide a longer throw and a smaller spread. Round nozzles (Fig. 18.10*a*) are usually used in air diffusion systems with large

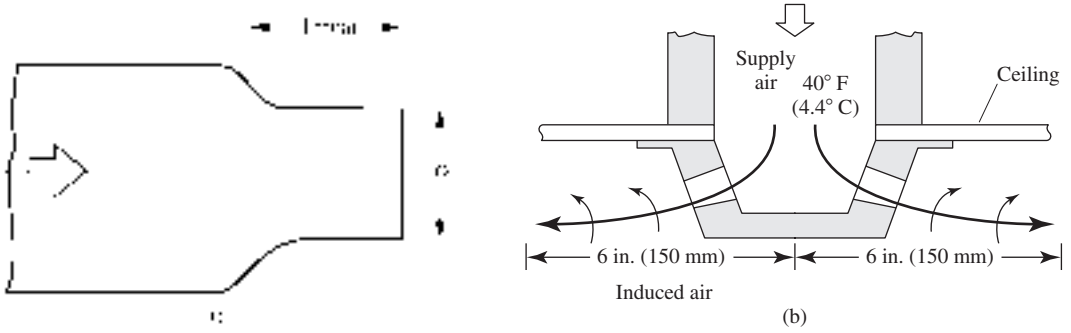

FIGURE 18.10 Round nozzle and nozzle diffuser (*a*) Round nozzle and (*b*) nozzle diffuser.

space volume or in spot cooling/heating, which is covered in detail in Sec. 18.9. Nozzle diffusers (Fig. 18.10*b*) have a high induction ratio and are used in cold air distribution, which is discussed in Sec. 18.5.

Accessories for Supply Outlets

Grilles, registers, and diffusers may require certain accessories to modulate the volume flow rate of each supply outlet and to distribute the air more evenly at the outlet. Currently used accessories include

- **Opposed-blade dampers**. Usually they are attached to the register upstream from the vanes, or installed at the outlet collar leading to a ceiling diffuser, as shown in Fig. 18.11*a*.
- **Split dampers**. A piece of sheet metal is hinged at one end to the duct wall, as shown in Fig. 18.11*b*. An equalizing grid is often used at the same time to distribute the supply air more evenly to the outlets. A split damper is also often installed at the junction of two main ducts, to adjust the volume flow rates of the supply air between these ducts.
- **Gang-operated turning vanes**. These vanes are always installed before the takeoff to the outlet, as shown in Fig. 18.11*c*. The tips of the turning vanes through which air enters the takeoff remain parallel to the direction of the airflow. Gang-operated turning vanes are especially suitable for adjusting the airflow rate for registers that are mounted on the air duct with a very short outlet collar.

Return and Exhaust Inlets

Various types of return inlets are used to return space air to the air-handling unit or packaged unit in the fan room. Exhaust air inlets are usually connected to the exhaust duct. Before contaminated air is exhausted outside, it is usually cleaned and treated with special air cleaning equipment.

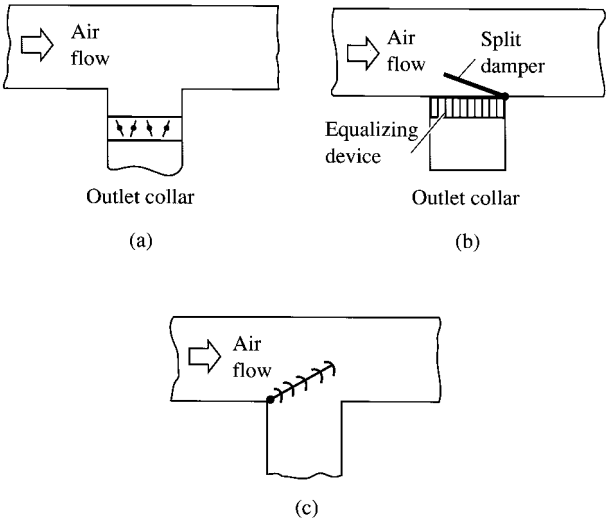

FIGURE 18.11 Accessories for supply outlets: (*a*) opposed-blade damper; (*b*) split damper with equalizing device; (*c*) gang-operated turning vanes.

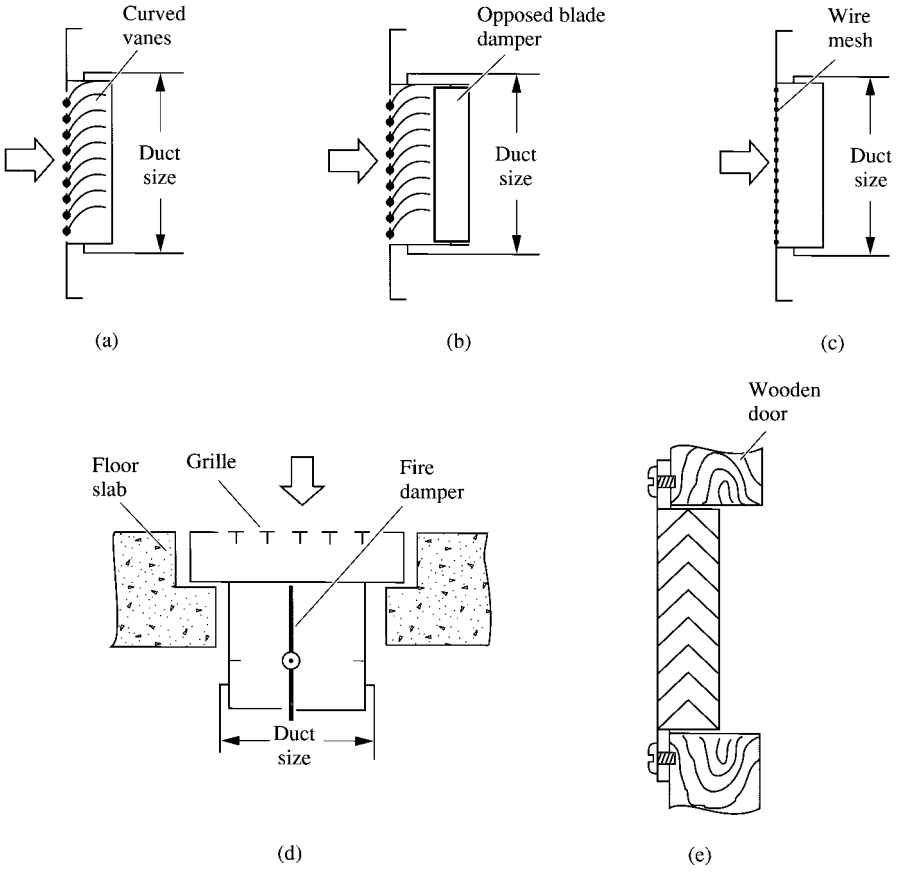

FIGURE 18.12 Various types of return grilles and registers: (*a*) return grille with curved vanes; (*b*) return register; (*c*) wire-mesh grille; (*d*) floor-type return grille; (*e*) door louvers.

Return and exhaust inlets can be classified as return or exhaust grilles, return or exhaust registers, return slots, and ventilated light troffers.

Return grilles and registers (see Fig. 18.12) are similar in shape and construction to supply grilles and registers. In Fig. 18.12*a*, only one set of vanes is required to direct the air into the return duct, so vision-proofed curved vanes are often used instead of the straight vanes. Vanes can be fixed or adjustable. An opposed-blade damper is sometimes installed behind the curved vanes in a return register, as shown in Fig. 18.12*b*. Wire-mesh grilles, as shown in Fig. 18.12*c*, are simple and inexpensive. They are often installed in places where the appearance of the grille is not important. In many industrial buildings, floor grilles are used when the return duct is below the floor level, as shown in Fig. 18.12*d*. When room air is required to return to the fan room through a corridor and a return duct, a door louver is often used to transport return air through the door. Vision-proofed, right-angle louvers are generally used for this purpose, as shown in Fig. 18.12*e*. Local codes should be consulted before the door louvers are installed.

When a ceiling plenum is used as a return plenum (as in many commercial buildings), return slots are used to draw the return air through the ceiling. Return slots are similar to slot diffusers. They are usually made with single or double slots, as shown in Fig. 18.13. The width of the slot

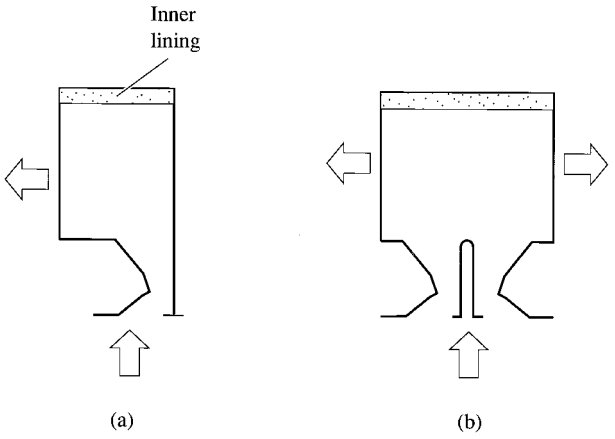

FIGURE 18.13 Return slots: (*a*) single-slot; (*b*) double-slot.

varies from 0.5 to 1.75 in. (13 to 45 mm). Slots are usually available in lengths of 2, 2.5, 3, 4, and 5 ft (600, 750, 900, 1200, and 1500 mm).

The capacity of return slots is also expressed in cfm/ft (L/s·m) of slot length. If a return slot has the same configuration, slot width, and slot intensity as the supply slot diffuser, then its total pressure loss, in in. WC (Pa), is nearly the same as the vertical projection total pressure loss of the supply slot diffuser. Return slots are usually inner-lined with sound absorption material. This material prevents sound transmission between two rooms via the return slots and the ceiling plenum.

Exhaust inlets are similar to the return inlets but they often have a simpler design.

In return and exhaust inlets, air velocity decreases very sharply as the distance from the surface of the inlet increases. This phenomenon is the major difference in airflow characteristics between a return or an exhaust inlet and a supply outlet.

Light Troffer Diffuser and Troffer-Diffuser Slot

Light troffer diffuser combines a fluorescent light troffer and slot diffuser, sometimes in saddle type, as shown in Fig. 18.14*a*. The slot can be used as a supply outlet or return inlet. A troffer-diffuser slot is a combination of light troffer, slot diffuser, and return slot, as shown in Fig. 18.14*b*. Light troffer diffusers and troffer-diffuser slots serve two main purposes:

1. To maintain a lower air temperature in the light troffer, which increases the luminous efficiency of fluorescent lamps
2. To form an integrated layout of light troffer, diffuser, and return slots on suspended ceilings

In addition to these benefits, a combination of a light troffer and return slot also reduces the space cooling load because return air absorbs part of the heat released by the fluorescent lights.

In a light troffer diffuser, surface slots are often used. Air can be discharged horizontally or vertically. Troffers and slots are available in 2-, 3-, and 4-ft (600-, 900-, and 1200-mm) lengths.

To prevent the deposit of dust particles on the surface of the fluorescent tube, the light troffer should be designed so that return air does not come in direct contact with the fluorescent tube. The result is a high ventilation efficiency and a neat appearance.

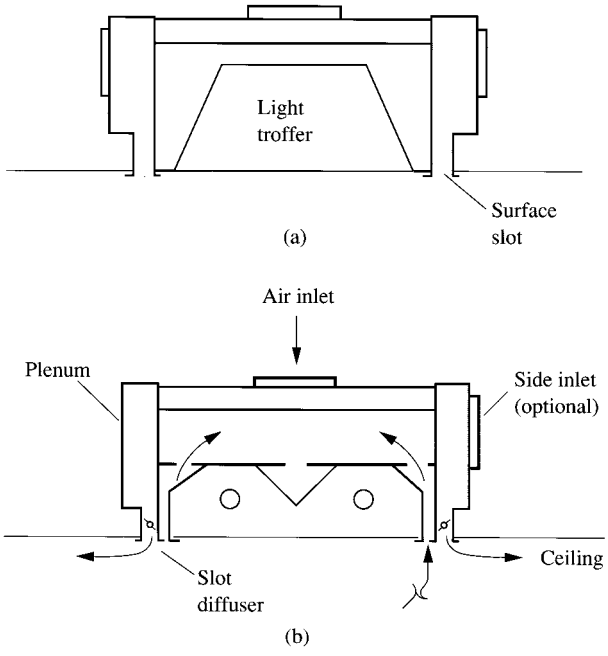

FIGURE 18.14 Light troffer, slot diffuser, and return slot combination: (*a*) light troffer diffuser; (*b*) troffer-diffuser slot.

18.4 MIXING FLOW

Airflow Pattern and Space Air Diffusion

Airflow pattern determines the performance of space air diffusion in the occupied zone of commercial buildings or in the working area of a factory. The optimum airflow pattern for an occupied zone depends mainly on the indoor temperature, relative humidity, and indoor air quality requirements; the outdoor air supply; and the characteristics of the building. Four types of airflow patterns are currently used in air conditioned space:

1. Mixing flow
2. Displacement flow
3. Projecting flow
4. Upward flow

Principles and Characteristics of Mixing Flow

The airflow pattern of an air conditioned space is said to be in mixing flow only when the supply air is thoroughly, or nearly thoroughly, mixed with the ambient air, and the occupied zone or working area is dominated by the induced recirculating flow.

In many commercial and industrial buildings, cold supply air at 50 to 60°F (10 to 15.6°C) and a velocity of 400 to 800 fpm (2 to 4 m/s) must be nearly thoroughly mixed with the ambient air first. The temperature of the mixture is raised to 72 to 74°F (22.2 to 23.3°C), and its velocity drops to less than 70 fpm (0.35 m/s) before it enters the occupied zone and working area, or drafts occur. Mixing flow is often the best choice for comfort air conditioning systems when the supply temperature differential is greater than 15°F (8.3°C) and the supply air velocity exceeds 300 fpm (1.5 m/s).

Straub and Chen (1957) tested mixing flow patterns by using five types of supply outlets. These airflow patterns have the following characteristics:

Induction of Space Air to the Air Jet. Mixing of induced air and the air jet reduces the differences in air velocity and temperature between the air jet and the ambient air to acceptable limits.

Reverse Airstream in the Occupied Zone. The induced reverse airstream covers the occupied zone with a more even temperature and velocity distribution.

Minimize the Stagnant Area in the Occupied Zone. A stagnant area is a zone with natural convective currents. Air velocities in most of the stagnant zones are lower than 20 fpm (0.1 m/s). Air stratifies into layers, with a significant temperature gradient from the bottom to the top of the stagnant zone. If the recirculating flow fills the entire occupied zone, no space remains for stagnant air.

Types and Locations of Return and Exhaust Inlets. Because the return or exhaust airflow rate is only a fraction of the maximum total airflow of the supply air jet or the maximum airflow of the reverse airstream, the location of the return or exhaust inlet does not significantly affect the airflow pattern of mixing flow in the occupied zone. However, the location of the return and exhaust inlets does affect the thermal effectiveness factor ε_T of space air diffusion. Therefore, for an induced recirculating flow pattern, the return intake should be located near the end of the reverse airstream, in the vicinity of a heat source, or within the stagnant zone.

For a conditioned space in which the space air is contaminated by dust particles, toxic gases, or odors, several evenly distributed return or exhaust grilles should be installed. Collective return grilles, i.e., one or two large return inlets instead of many inlets, impair indoor air quality. They are only suitable under the following circumstances:

- When the conditioned space is small
- When it is difficult to provide a return duct
- When lower initial cost is of prime importance

Types and Locations of Supply Outlets. The type and the location of supply outlets affect the mixing flow pattern and its performance. Currently, mixing airflow incorporates the following types of supply outlets:

- High side outlets
- Ceiling diffusers
- Slot diffusers
- Sill and floor outlets
- Outlets from stratified mixing flow

Mixing Flow Using High Side Outlets

A high side outlet may be a sidewall outlet mounted at the high level of a conditioned space, or an outlet mounted directly on the supply duct. Figure 18.15 shows a mixing flow using a high sidewall

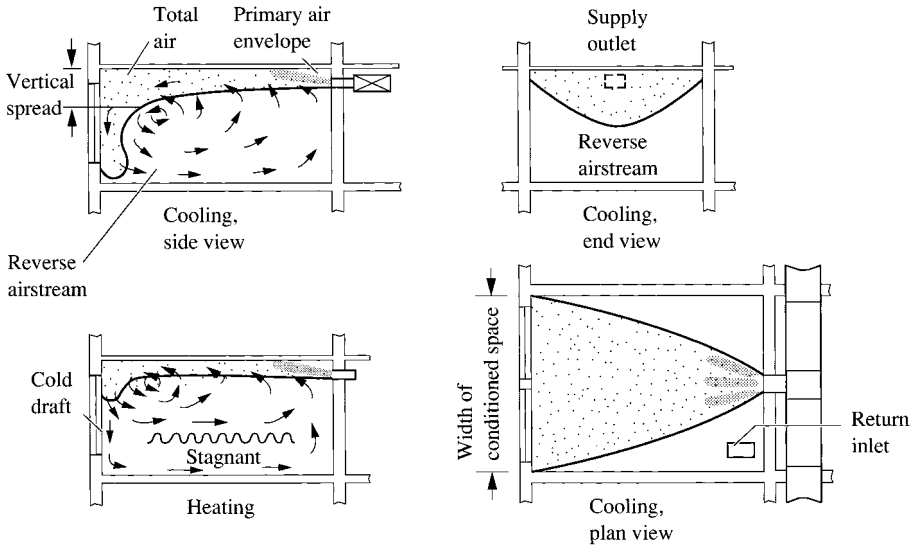

FIGURE 18.15 Mixing flow using high side outlets. (*Adapted with permission from ASHRAE Handbook 1989, Fundamentals*.)

outlet. In Fig. 18.15, the shaded black envelope shows the primary air envelope of the air jet, and the dotted envelope shows the total air envelope.

As the air jet discharges from the high sidewall outlet, the surface effect tends to keep the air jet in contact with the ceiling. During cooling, the air jet induces the ambient air and deflects downward when it strikes on the opposite wall. As soon as the maximum air velocity of the air jet drops to about 50 fpm (0.25 m/s), the air jet terminates as it flows along the opposite wall. The induction of space air from the occupied zone into the air jet forms the reverse airstream and fills the occupied zone.

If the throw of a high sidewall outlet is longer than the sum of the length of the room and the height of the opposite wall, then the air jet is deflected by the opposite wall and floor and enters the occupied zone with excessive air velocity. Both the air jet and the reverse airstream fill the occupied zone with a higher air velocity and greater temperature difference.

If the throw is too short, the air jet drops and may enter the occupied zone directly. This causes higher air velocity and lower temperature (cold draft) in the occupied zone.

During heating, warm air tends to rise and results in a shorter throw. As the induced airstream rises, a stagnant zone may form between the floor and the induced airstream. A warm air jet with sufficient velocity and a longer throw can reduce or even eliminate the stagnant zone.

If the airstream at the terminal zone of the warm air jet comes in contact with the cold inner surface of the window, it converts to a cold airstream and forms a cold draft when it flows downward and enters the occupied zone. The installation of a baseboard heater under the window sill can prevent the formation of this cold draft and reduce the vertical temperature gradient in the occupied zone.

The most suitable location for a return inlet is on the ceiling outside the air jet. Return air flows through the return inlets and often uses the ceiling plenum as a return plenum. Such an arrangement often results in a higher effectiveness factor ε_T and a lower installation cost for the return system.

For a room without a suspended ceiling, a cold or warm air jet has a shorter throw because of lack of surface effect. The buoyancy force causes a cold air jet to drop sooner, and it may enter the occupied zone directly. The higher rise of a warm air jet forms a greater stagnant zone above the floor level.

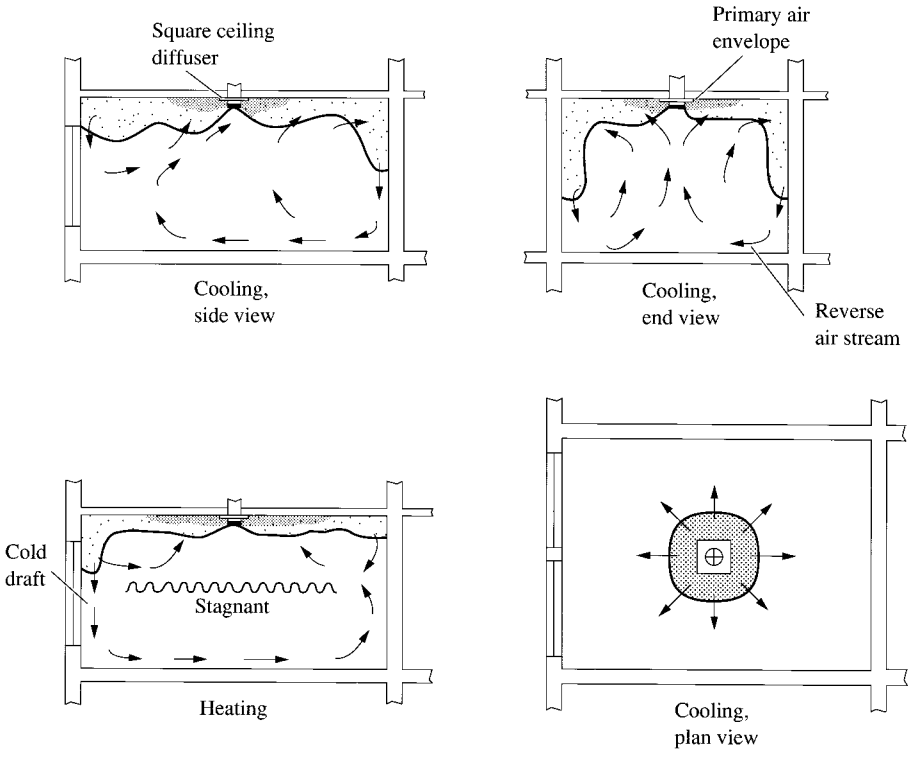

FIGURE 18.16 Mixing flow using ceiling diffusers. (*Adapted with permission from ASHRAE Handbook 1989, Fundamentals*.)

Mixing flow using high side outlets usually gives a longer throw and a higher air velocity in the occupied zone than other supply outlets. Drop must be checked to prevent the air jet from entering the occupied zone directly in a large conditioned space.

Mixing Flow Using Ceiling Diffusers

The airflow pattern of mixing flow using ceiling diffusers is similar to that of high side outlets. Ceiling diffusers produce a better surface effect, a shorter throw, a lower and more even distribution of air velocity, and a more even temperature in the occupied zone.

Figure 18.16 shows mixing flow using ceiling diffusers. During cooling, the cold air jet usually maintains contact with the ceiling, producing a better surface effect than sidewall outlets. The reverse airstream fills the occupied zone. During heating, thinner vertical spread and shorter throw create a larger, higher stagnant zone. The best location for return inlets is on the ceiling within the terminal zone of the air jet.

When two air jets discharge horizontally from ceiling diffusers or other outlets in opposite directions along the ceiling, with a ceiling height of 10 ft (3.05 m), the air velocity where two air jets collide should be less than 120 fpm (0.6 m/s) in order to maintain an air velocity below 50 fpm (0.25 m/s) 5 ft (1.5 m) above the floor. Mixing flow using ceiling diffusers is widely used for conditioned spaces with limited ceiling height. Many ceiling diffusers are designed to have a high

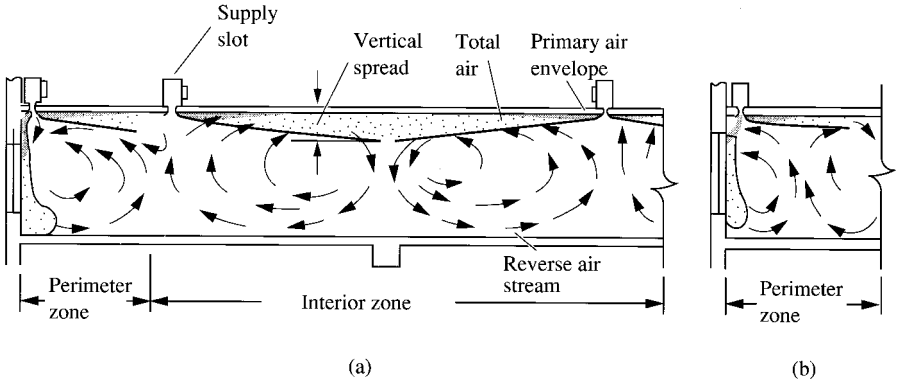

FIGURE 18.17 Mixing flow using slot diffusers: (*a*) perimeter and interior zone, cooling; (*b*) perimeter zone, heating.

induction ratio and to produce a very good surface effect. These ceiling diffusers are often used in variable-air-volume systems.

Mixing Flow Using Slot Diffusers

Slot diffusers are narrower and longer than ceiling diffusers and grilles. When supply air is discharged horizontally from a ceiling slot diffuser, it has a thinner vertical spread and an excellent surface effect. Figure 18.17 shows mixing flow using slot diffusers discharging vertically downward into the perimeter zone and also discharging horizontally along the ceiling in both the perimeter and interior zones of a large office.

Because of its superior surface effect, the horizontally discharged cold air jet remains in contact with the suspended ceiling even when the supply air volume flow rate is reduced in a variable-air-volume system. The occupied zone is then filled with the reverse airstream at a more uniform temperature and air velocity.

In order to counteract natural convection along the inner surface of a window in the perimeter zone, the air jet is often projected downward toward the window. A sufficient throw is important to produce a downward air jet that offsets the cold draft at the inner surface of the window during winter heating. The occupied zone fills with induced airflow at an air velocity lower than 50 fpm (0.25 m/s).

For a mixing flow using slot diffusers, the location of return slot is preferably aligned with the supply slots.

In addition to their excellent surface effect in mixing flow, slot diffusers have a linear appearance that can be coordinated easily with electric lights and ceiling modular arrangements. Therefore, they are widely used in buildings using VAV systems with moderate loads and normal ceiling height.

Mixing Flow Using Sill or Floor Outlets

The purpose of using a sill or floor outlet is to counteract the cold draft flowing downward along the inner surface of the window when the outdoor temperature is below 30°F (− 1.1°C). Figure 18.18 shows a mixing flow using a sill outlet.

During cooling, a stagnant zone may form above the eddies beyond the terminal zone of the air jet if the supply air velocity is not high enough. Meanwhile the space under the cold air jet is filled

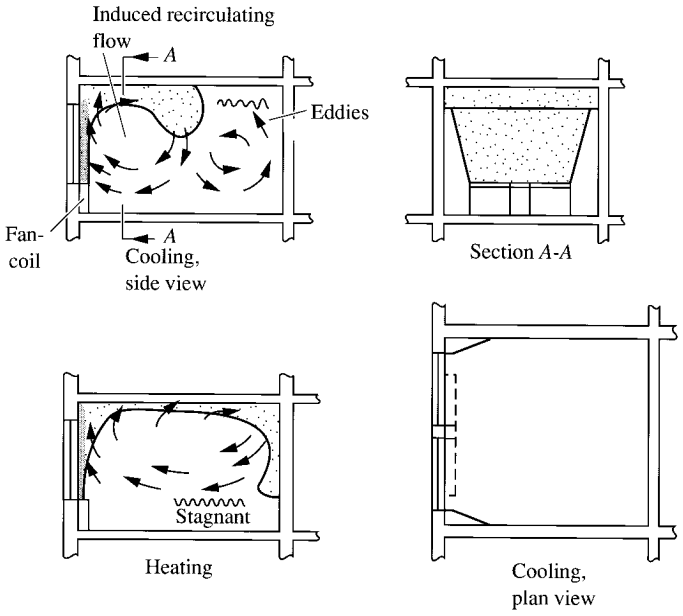

FIGURE 18.18 Mixing flow using sill outlet.(*Adapted with permission from ASHRAE Handbook 1989, Fundamentals*.)

with reverse airstreams and cooled. During heating, a stagnant zone may form below the ascending induced airstream.

Mixing flow with sill or floor outlets has been widely used in buildings with large window areas or in raised-floor offices. The direction and amount of airflow from floor diffusers usually can be adjusted according to the requirements of the occupants.

Stratified Mixing Flow

In a building with a high ceiling, it is more economical to stratify the building vertically into two zones (the stratified upper zone and the cooled lower zone) or three zones (upper, transition, and lower zones) during cooling. The upper boundary of the lower zone is at the level of the supply outlet where the air jet projects horizontally.

Gorton and Sassi (1982) and Bagheri and Gorton (1987) performed a series of model studies and experiments about stratified space air diffusion. Figure 18.19*a* shows the elevation view of the nuclear reactor facility in which the experiments were conducted. The ceiling height was 41 ft (12.5 m). An air-handling unit was located at floor level, and two diffusers were mounted on the supply riser 16 ft (4.9 m) above the floor level. The supply velocity at the outlet was about 1000 fpm (5 m/s), and the supply temperature was around 60°F (15.6°C).

Figure 18.19*b* shows the space air temperature profile at 4 p.m. at various height levels during cooling. This temperature profile could be divided into

- An upper zone, from the ceiling down to about 22 ft (6.7 m) from the floor level, in which air temperature varied from 82.5 to 79°F (28 to 26.1°C)
- A transition zone, between 16 and 22 ft (4.9 and 6.7 m) from the floor, in which air temperature varied from 79 to 74.5°F (26.1 to 23.6°C)

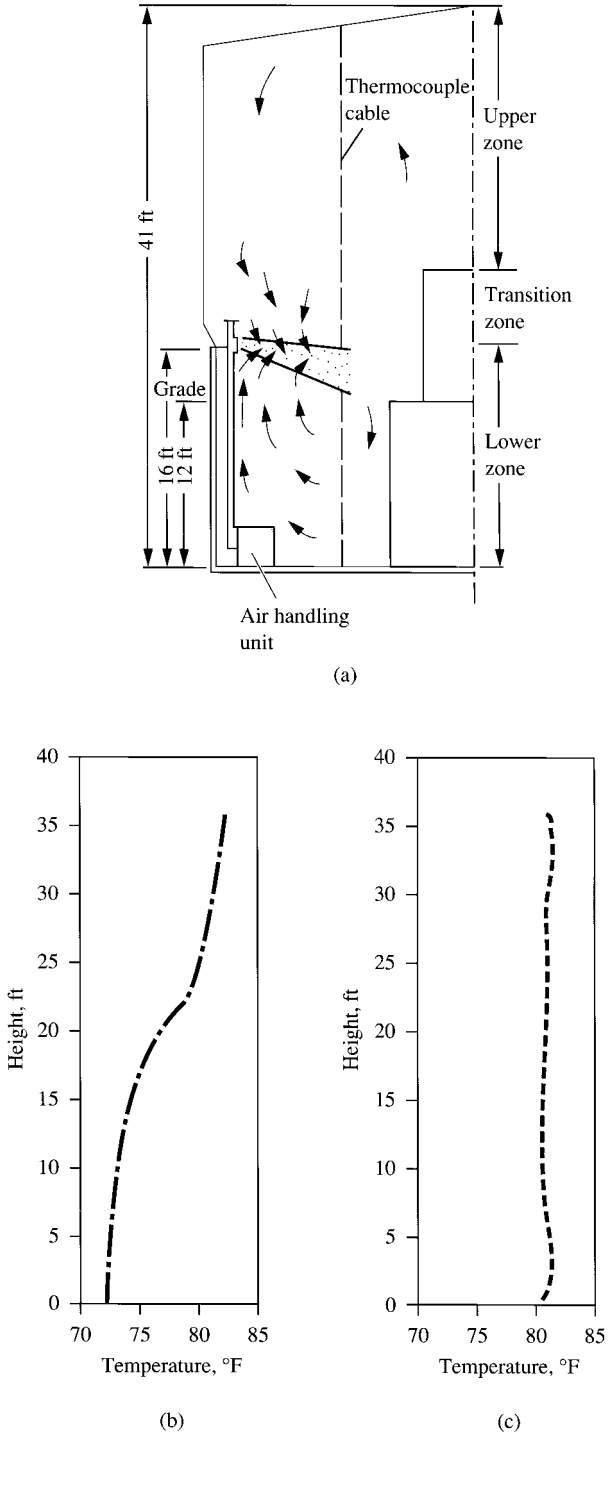

FIGURE 18.19 Stratified mixing flow in a nuclear reactor facility: (*a*) elevation view of a nuclear reactor facility; (*b*) temperature profile during cooling; (*c*) temperature profile during heating.

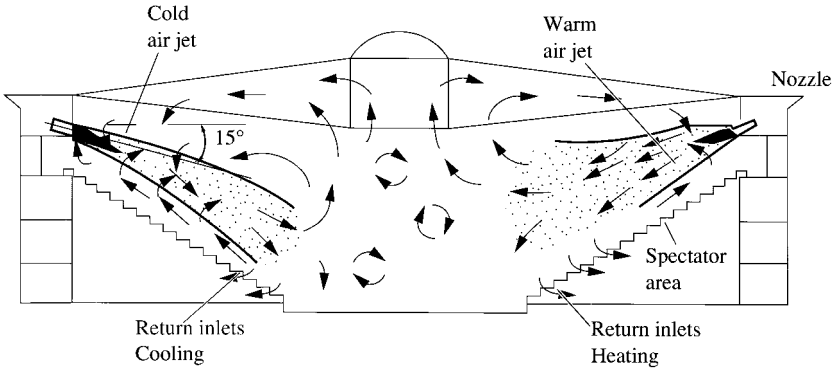

FIGURE 18.20 Stratified mixing flow in a large indoor stadium using supply nozzles.

- A lower zone below 16 ft (4.9 m), in which air temperature varied between 72 and 72.5°F (22.2 and 22.5°C)

The induced recirculating flow under the cold air jet formed the lower zone. Most of the lower zone was filled with reversed airstreams. The upper zone and the transition zone were formed because of the entered heat gains at the upper level and a weaker recirculating induced flow above the cold air jet. Because of the drop of the cold air jet, a greater amount of recirculating flow was induced in the cold air jet in the lower zone than in the upper zone. Consequently, most of the cooling occurred in the lower zone. The formation of a stagnant layer near the ceiling also retarded airflow and heat transfer between outdoor and indoor air in the upper zone.

Figure 18.19*c* shows the space temperature profile at various height levels at 4 p.m. during heating. The supply air temperature was about 85°F (29.5°C) and the outside temperature 40°F (4.4°C). The air temperature varied from 80 to 81.5°F (26.7 to 27.5°C) from a height of 0 to 36 ft (0 to 11.0 m) from floor level. There was no evidence of significant thermal stratification.

Figure 18.20 shows a stratified mixing flow in a large, high-ceiling indoor stadium using supply nozzles. During summer cooling, the cold air jet creates two induced recirculating airflows: a cooled lower occupied zone and a stratified upper zone. If a higher air velocity is required in the lower occupied zone (the spectator's area), the supply nozzles should project at an inclined angle. Otherwise, the cold air jet should project horizontally.

During heating, if the throw of the warm air jet is long enough to arrive at the lowest spectator seat, the whole spectator area is filled with the reversed airstream. No stagnant zone is formed in the lower occupied zone. Return inlets should be located in the lower cooling zone. They should be evenly distributed under the spectator seats.

Characteristics of the stratified mixing flow during summer cooling mode operation are as follows:

- Convective heat transfer from the hot roof is effectively blocked by the higher-temperature or stagnant air layer in the upper zone.
- Cooling loads that occur in the lower zone (windows, walls, lights, occupants, and equipment) should be offset by cold supply air.
- Radiant heat from the roof, upper external walls, and electric lights in the upper zone enters the occupied zone and converts to cooling load.
- Although supply airflow rate and supply temperature affect the throw and drop of the air jet, the induced recirculating airflow pattern and the stratified upper and lower zones remain the same.
- The height of the supply air jet determines the upper boundary of the lower zone.

- The location of return inlets influences the cooling load only when they are located in the upper or transition zone. A portion of exhaust air should be extracted at a higher level.

18.5 COLD AIR DISTRIBUTION

Cold Air Distribution Versus Conventional Air Distribution

Because of the lower chilled water discharge temperature of 34 to 38°F (1.1 to 2.2°C) from ice storage systems, the supply air temperature of the air system can often be reduced to 40 to 45°F (4.4 to 7.2°F) and results in a cold air distribution (low-temperature distribution) system instead of a conventional system of 55°F (12.8°F) supply temperature. Dorgan and Elleson (1993) and Scofield (1993) summarized that a cold air distribution system has the following advantages when compared with a conventional space air diffusion system:

- Due to the low dew point of supply air, space relative humidity can be maintained between 35 and 45 percent. Cold air distribution can increase the space temperature slightly without causing discomfort.
- For a cold air distribution system using fan-powered VAV boxes, T_s is equal to the air temperature discharged from the fan-powered unit during summer cooling mode operation if the duct heat gain between the fan-powered VAV box and the supply outlet is ignored. If $T_s = 44$°F (6.7°C) and space air temperature $T_r = 78$°F (25.5°C), then the supply temperature differential for cold air distribution $T_r - T_s = 78 - 44 = 34$°F (18.9°C). Compared with a supply temperature differential $T_r - T_s = 20$°F for a conventional system, cold air distribution systems reduce the design supply volume flow rate by 44 percent.
- With cold air distribution, fans, ducts, and the associated air system components can be downsized. Smaller ducts require less space clearance between the ceiling and structural members, and thus a reduction of floor-to-floor height in high-rise buildings. Also there is a significant increase in rental space because of the reduction of the mechanical rooms and duct shafts.
- Fan energy use can save as much as 40 percent.
- Savings are seen in construction cost because of the reduction of the structural height and smaller ductwork and air system components (sometimes it is the primary consideration).
- There are reduced fan sound levels and, therefore, less sound attenuation for a specified space *NC* rating.
- Cold air distribution is often beneficial for a retrofit project in which the existing air distribution system may be used for increased space cooling loads.

The major reservation of using cold air distribution, especially for VAV systems in large buildings, is the justification of a reduction of 40 to 45 percent of supply air at a time of increased concern over IAQ and the dumping of the cold air jet at reduced volume flow.

High Induction Nozzle Diffusers

Two types of space diffusion systems are used in cold air distribution: (1) Cold supply air from the air-handling unit or packaged unit is supplied directly to the conditioned space by means of high induction diffusers; (2) fan-powered VAV boxes are used to mix low-temperature supply air with return air, and the mixture at a temperature between 55 and 60°F (13 and 16°C) is then supplied to the conditioned space.

Fields and Knebel (1991) and Knebel and John (1993) introduced a high induction nozzle diffuser used in cold air distribution, as shown in Fig. 18.10*b*. A typical 4-ft- (1.22-m-) long high induction nozzle diffuser has 57 0.75-in. (19-mm) nozzles on each side of the diffuser. At an airflow rate of 350 cfm (165 L/s), it has a supply air velocity of about 1000 fpm (5 m/s), a total pressure loss at the supply outlet of 0.213 in. WC (53 Pa), a throw $T_{50} = 33$ ft ($T_{0.25} = 10$ m), and a drop of about 0.8 ft (0.24 m).

Characteristics of Cold Air Distribution Systems

A satisfactory cold air distribution system often has the following characteristics:

- A higher $\sqrt{A_r}/D_o$ value, as defined in Sec. 18.2, to provide adequate surrounding spaces for the supply air jets to induce sufficient ambient air and fully mix the ambient air with the 40 to 45°F (4.4 to 7.2°C) low-temperature supply air
- A higher supply air velocity and jet turbulence (depending on the configuration of the air diffusion system) to provide momentum for the induced airstreams
- An excellent surface effect with adequate throw and small drop to prevent supply air jet from entering the occupied zone directly
- An ADPI $\geq$ 80 percent at both design and reduced airflow

Case Study—Florida Elementary School

According to Knebel and John (1993), in a Florida elementary school, the HVAC&R system consists of a 40,000 cfm (18,880 L/s) blow-through AHU with airfoil fan and an ice storage system. The DDC controlled VAV terminals are used to throttle the airflow and reset the supply air temperature. The classroom that uses cold air distribution is 27 ft (8.3 m) wide by 36 ft (11.0 m) long. Three 4-ft nozzle-diffusers, each with 114 nozzles, were set in one row, designed to produce a throw of 18 ft (5.4 m) on each side. Each nozzle-diffuser had a design airflow rate of 200 cfm (94 L/s) and a supply velocity of 500 fpm (2.5 m/s) at the nozzle outlet. With this velocity, the 50 fpm (0.25 m/s) air jet envelope was 30 in. (0.76 m) from the nozzle. The air motion in the space was between 0 and 10 fpm (0 and 0.05 m/s).

The diffuser supplier modified the number of active nozzles in each diffuser. The supply velocity at the nozzle outlets was raised to 2000 fpm (10 m/s) at the design airflow and to 1000 fpm (5 m/s) at 50 percent part-load operation. For an outlet velocity of 2000 fpm (10 m/s) and a supply air temperature of 40°F (4.4°C), space air velocity had been raised to 15 to 45 fpm (0.075 to 0.23 m/s) and the measured ADPI was 100 percent. At part-load operation, when the outlet velocity dropped to 1000 fpm (5 m/s) with an supply air temperature of 41°F (5°C), the space air velocity varied from 10 to 40 fpm (0.05 to 0.20 m/s) and the measured ADPI was 96.2 percent. The estimated $\sqrt{A_r}/D_o$ value for this cold air distribution system in this classroom is about 37.

Performance of Ceiling and Slot Diffusers

Zhang et al. (1994) conducted ventilation performance tests for two square diffusers (three concentric inner cones discharged in all directions, which is similar to the ceiling diffuser shown in Fig. 18.8*b*) and one-slot diffusers (horizontally projected) using 38°F (3.3°C) supply air for cold air distribution and 55°F (12.8°C) supply air for conventional air distribution. Test results were as follows:

- Cold and conventional air distribution systems performed equally well in delivering the outdoor ventilation air into the occupied zone.

- The average air exchange effectiveness (AEE) of the small square diffuser using cold air distribution was 1.01, and using conventional air distribution was 0.99. The average AEE for the slot diffuser using cold air distribution was 0.99, and using conventional air distribution was 1.00. The high magnitudes of AEE (> 0.90) showed that almost all the supply air was delivered into the occupied zone.
- The ADPI for the large square diffuser using cold air distribution was 92.5 percent, for the small square diffuser was 96.7 percent, and for the slot diffuser was 96.7 percent. The cold air distribution system for the slot diffuser has a $\sqrt{A_r}/D_o$ value of 22. The ADPI for small square diffuser using conventional air distribution was 96.7 percent and for the slot diffuser was 90.0 percent.
- Cold air distribution with these three diffusers provided satisfactory ventilation performance. There was no appreciable difference between the cold and conventional air distribution systems.

Cold Air Distribution with Fan-Powered VAV Boxes

There are two types of fan-powered VAV box used in cold air distribution: parallel fan-powered box and series fan-powered box. In a parallel fan-powered box, the cold primary air from the AHU does not flow through the VAV box fan but instead mixes with the recirculating air after the box fan outlet, whereas in a series fan-powered box, the cold primary air flows through the box fan. Parallel and series fan-powered boxes are discussed in Sec. 21.5. For 44°F (6.7°C) cold primary air and 78°F (25.6°C) ceiling plenum air, 40 percent of 78°F (25.6°C) return air is often extracted to the fan-powered box and is mixed with 60 percent of 44°F (6.7°C) cold primary air from the AHU, resulting at a 58°F (14.4°C) mixture, the same as in the conventional air distribution system supplied to the space. A fan-powered box has the benefits of providing space air motion.

Elleson (1993) compared the energy use with and without the fan-powered mixing boxes in cold air distribution. In general, the energy required for continuous mixing box fan operation is greater than the supply fan energy savings resulting from the reduced supply temperature.

Box fan energy use can be reduced by operating the box fan only when space air motion falls below a minimum level. Supply air temperature reset can further reduce fan energy consumption. Direct supply of cold primary air to the conditioned space entails the lowest fan energy use. Furthermore, compared to the direct supply of cold primary air, cold air distribution using fan-powered boxes has a higher noise level and initial cost and requires more site maintenance. Direct supply of low-temperature cold primary air to the conditioned space is recommended.

Surface Condensation

For a space air temperature of 75°F (23.9°C) and a relative humidity of 50 percent, the dew point of space air is 55°F (12.8°C). During cold air distribution, the space relative humidity can be reduced to 40 percent, and the dew point is then about 49.5°F (9.7°C). For a cold air distribution with a supply air of 40°F (4.4°C), the temperature differential between the supply air inside the ducts and the ambient air increased to $75 - 40 = 35$°F (19.4°C) instead of $75 - 55 = 20$°F (11.1°C) in a conventional air distribution system. Harmon and Yu (1993) reported that the actual space relative humidity levels measured were 3 to 5 percent above those projected by calculations. In cold air distribution systems, in order to prevent surface condensation, the AHU or PU, ducts, duct access doors, VAV boxes, flexible ducts, and diffusers should be insulated with a layer of fiberglass protected by a vapor barrier, or a closed-cell elastomeric or cellular glass insulation, with a thickness of 2 in. (50 mm). It is preferable to select VAV boxes with thermally isolated inlet connections and properly insulated diffusers.

Plenum space must be properly sealed to prevent outside air infiltration. Gradually cool the supply air and the space air during each early morning in summer to form a soft start, so that the outside surface will not suddenly be cooled below the dew point of the ambient air.

18.6 DESIGN PROCEDURE OF MIXING FLOW AIR DIFFUSION SYSTEM

The supply volume flow rates of various control zones at summer and winter design conditions are usually determined before a mixing flow air diffusion system is designed. Because cooling mode operations are used in both perimeter and interior zones in summer and often in interior zones even in winter, heating mode operations often occur only in the perimeter zone during winter. Therefore, cooling mode operation is the basic consideration in space air diffusion design. In the perimeter zone, heating mode space air diffusion must be considered, and a compromise may be necessary. The design procedure for such an air diffusion system follows.

Select the Type of Supply Outlet

Selection of the supply outlet depends on the following:

- *Requirements of indoor environmental control*. If the conditioned space needs controlled air movements, such as for a badminton and table tennis tournament, or precise air temperature control, a high side outlet is not the right choice.
- *Shape, size, and ceiling height of the building*. For buildings with limited ceiling height, ceiling and slot diffusers are often the best choice. For large buildings with high ceilings, high side outlets mounted at high levels to form stratified induced recirculating flow patterns are recommended. In a perimeter zone, an overhead two-way slot diffuser projected down toward the window and horizontally projected to the room, a ceiling diffuser with a throw to the inner surface of the window glass, or a sill outlet should be used.
- *Surface effect*. A good surface effect is especially important to the VAV system because it allows the supply volume flow rate to be reduced to half or even 30 percent of the design flow.
- *Volume flow per ft² of floor area*. Sidewall outlets are limited to a lower volume flow per ft² of floor area, in cfm/ft² ($L/s \cdot m^2$), because of the higher air velocity in the occupied zone. The slot diffuser has a narrower slot width and can only project in one or two directions. Therefore, the volume flow per ft² for a slot diffuser is smaller than that of a ceiling diffuser. Table 18.2 lists the volume flow per ft² for various types of supply outlet recommended by *ASHRAE Handbook 1992, HVAC Systems and Equipment*.
- *Appearance*. The shape and configuration of outlets and inlets are closely related to the interior appearance of the building and should be coordinated with inlets and lighting troffers.

TABLE 18.2 Volume Flow per ft² Floor Area

Type of outlet	Supply air density, cfm/ft² of floor space	Approximate maximum air changes per hour for 10-ft (3-m) ceiling
Grille	0.6–1.2	7
Slot diffuser	0.8–2.0	12
Perforated panel	0.9–3.0	18
Ceiling diffuser	0.9–5.0	30
Ventilating ceiling	1.0–10.0	60

Source: *ASHRAE Handbook 1992, HVAC Systems and Equipment*. Reprinted with permission.

- *Cost*. In many commercial buildings, cost is often an important factor in determining the type of supply outlet.

Volume Flow Rate per Outlet or per Unit Length

For a specific supply volume flow rate in a conditioned space, the volume flow rate per supply outlet $\dot{V}_{s,\text{out}}$, in cfm (L/s), determines the number of outlets in the conditioned space. As defined in Eq. (18.10), both $\dot{V}_{s,\text{out}}$ and supply outlet velocity $v_{s,o}$ or v_{core}, in fpm (m/s), affect the throw of the supply airstream. However, $\dot{V}_{s,\text{out}}$ has a significantly greater influence than $v_{s,o}$, especially for slot diffusers.

The volume flow rate per supply outlet depends mainly on the throw required to provide a satisfactory space air diffusion system design. Load density, space air diffusion system characteristics, sound control requirements, and cost considerations are also factors that determine $\dot{V}_{s,\text{out}}$. In a VAV system in a high-rise office building, avoiding the discharge of cold air into the space directly at a reduced volume flow rate is one of the primary considerations.

For a space air diffusion system using slot diffusers, the supply volume flow per ft (m) length of slot diffuser (the slot intensity), in cfm/ft (L/s·m), is often an important index, especially in the return slots. It is usually 15 to 40 cfm/ft (23 to 62 L/s·m). All return slots mounted on the same

TABLE 18.3 Relationship between ADPI and T_{50}/L and T_{100}/L

Terminal device	Load density, Btu/h·ft^2	T_{50}/L for max. ADPI	Maximum ADPI	For ADPI greater than	Range of T_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5–2.2
	40	1.6	78	70	1.2–2.3
	20	1.5	85	80	1.0–1.9
Circular ceiling diffusers	80	0.8	76	70	0.7–1.3
	60	0.8	83	80	0.7–1.2
	40	0.8	88	80	0.5–1.5
	20	0.8	93	90	0.7–1.3
Sill grille straight vanes	80	1.7	61	60	1.5–1.7
	60	1.7	72	70	1.4–1.7
	40	1.3	86	80	1.2–1.8
	20	0.9	95	90	0.8–1.3
Sill grille spread vanes	80	0.7	94	90	0.8–1.5
	60	0.7	94	80	0.6–1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3–0.7
	60	0.3	88	80	0.3–0.8
	40	0.3	91	80	0.3–1.1
	20	0.3	92	80	0.3–1.5
Light troffer diffusers	60	2.5	86	80	< 3.8
	40	1.0	92	90	< 3.0
	20	1.0	95	90	< 4.5
Perforated and louvered ceiling diffusers	35–160	2.0	96	90	1.4–2.7
				80	1.0–3.4

Source: Adapted with permission from *ASHRAE Handbook 1989, Fundamentals*. Reprinted with permission.

return ceiling plenum must have the same total pressure loss, or using same type of return slot, the same slot intensity.

In a closed office of an area of 150 ft^2 (14 m^2) or less with only one side of external wall, usually one ceiling diffuser is sufficient.

Choose an Optimum Throw–Characteristic Length Ratio

For most types of supply outlets, the selection of an optimum throw–characteristic length ratio T_{50}/L ($T_{0.25}/L$) or T_{100}/L ($T_{0.5}/L$) determines the layout and grid of the supply outlets. An optimum T/L ratio should meet the following requirements:

- Selected T_{50}/L or T_{100}/L values should have an ADPI greater than 70 and 80 percent, as listed in Table 18.3.
- The spread at the end of the supply air jet covers or almost covers the width of the conditioned space before the air jet enters the occupied zone as shown in Fig. 18.15.

Miller (1976) and others conducted many experiments and determined the relationship between the ADPI and T_{50}/L or T_{100}/L. They found that the cooling load density is also a factor. Table 18.3 lists the relationship between ADPI and T_{50}/L or T_{100}/L at various load densities. The characteristic length of a sill grille is defined as the length of the room or space in the direction of jet flow. In Table 18.3, T_{50} and T_{100} ($T_{0.25}$ and $T_{0.5}$) are throws of isothermal jets given in most of the manufacturers' catalogs for selection convenience.

Generally, high sidewall outlets have a longer throw and therefore a higher T_{50}/L ($T_{0.25}/L$) range than do ceiling diffusers. Slot diffusers have a wider T_{100}/L ($T_{0.5}/L$) range in which ADPI exceeds 80 percent than do high sidewall outlets. This characteristic makes them suitable for use at reduced volume flows. Square ceiling diffusers have a T_{50}/L ($T_{0.25}/L$) range similar to that of the circular ones. Higher cooling load density usually results in a lower ADPI.

For VAV systems, T_v/L must be selected within the satisfactory range, that is, ADPI $>$ 80 percent, for both maximum and minimum airflow.

Determine the Design Characteristics of Slot Diffusers in Perimeter Zone

In the perimeter zone, overhead slot diffusers and ceiling diffusers are widely used. The characteristic length of an overhead slot diffuser in the perimeter zone has not been clearly defined. Based on tests performed by Straub and Cooper (1991), Lorch and Straub (1983), and Int-Hout (1998), the design characteristics of overhead slot diffusers in the perimeter zone are as follows:

- Two-slot, two-way overhead slot diffusers should be used. The suitable distance between the overhead slot diffuser and the external wall is about 1 ft (0.3 m). One of the slots near the window is set to blow down, and the other slot blows horizontally into the room. Such an arrangement offsets the cold draft along the window in winter heating.
- During cooling mode operation, ADPI exceeding 80 percent occurs within a wider range of slot intensities, load densities, and spacing between the diffuser and external wall than during heating mode. Heating mode operation requires careful consideration.
- According to ASHRAE Standard 55-1992, the criteria for an acceptable space air diffusion in heating mode operation are based on the requirement that the vertical temperature gradient in the occupied zone measured between 4-in. (0.1-m) and 67-in. (1.7-m) levels not exceed 5°F (3°C).
- According to *ASHRAE Handbook 1997, Fundamentals*, the supply temperature differential for warm air during heating mode $T_s - T_r$ should not be greater than 15°F (8.3°C), to prevent excessive buoyancy.

- A terminal velocity of 150 fpm (0.75 m/s) is recommended at the 5-ft (1.5-m) level near the window, to offset cold drafts in winter.
- An evenly distributed air volume flow rate between the vertically and horizontally discharged slots is recommended.

Select the Specific Supply Outlet from Manufacturer's Catalog

After an optimum T_{50}/L ($T_{0.25}/L$) or T_{100}/L ($T_{0.5}/L$) and characteristic length L are determined from the preliminary layout, the supply outlet can be selected from the manufacturer's catalog from known T_{50} ($T_{0.25}$) or T_{100} ($T_{0.5}$) and supply volume flow rates, after the following parameters have been checked:

Sound Level. The combined sound level of terminal and outlet should be at least 3 dB lower than the recommended *NC* criteria in the conditioned space. For optimum noise control, the recommended air velocities at the supply outlet are as follows:

Residences, apartments, churches, hotel guest rooms, theaters, private offices	500 to 750 fpm (2.5 to 3.75 m/s)
General offices	500 to 1250 fpm (2.5 to 6.25 m/s)

The outlet velocity for the ceiling diffuser can be calculated by dividing the volume flow by area factor A_k, given in the manufacturer's catalog.

Drop of Cold Air Jet. Drop of a cold air jet should be checked if the cold jet enters the occupied zone directly. Figure 18.21*a* shows the drop and other data of typical sidewall outlets mounted within 1 ft (0.3 m) of the ceiling with an adjustable vane deflection of vertical 5° up and horizontal 0°. Figure 18.21*b* shows the drop for sidewall outlets without surface effects and a vane deflection of vertical 15° up and horizontal 0°. Both are cold air jets based on a supply temperature differential of 20°F (11.1°C).

Total Pressure Loss of the Supply Outlet. The total pressure loss of supply air when it flows through a slot diffuser with a slot width of 0.75 in. (19 mm) is usually between 0.05 and 0.20 in. WC (12 and 50 Pa). For a ceiling diffuser, it is between 0.02 and 0.2 in. WC (5 and 50 Pa). A total pressure loss higher than 0.20 in. WC (50 Pa) is unsatisfactory.

Determination of Final Layout of Supply Outlets and Return Inlets

The determination of the volume flow rate per outlet, the number of outlets, T_{50}/L ($T_{0.25}/L$), and the selection of specific outlets from the manufacturer's catalog is often an iteration procedure. After that, final layout of supply outlets and return inlets can be determined.

In the perimeter zone, the high side wall outlet in recently constructed hotel guest rooms are often located on the partition wall adjacent to the window. In the interior zone, ceiling and slot diffusers are usually located directly above the center of the conditioned space.

As mentioned in Sec. 18.4, if a ceiling plenum is used as return plenum, the return inlets should be located outside the supply air jet, above the return airstream, or near a concentrated heat source for a better effectiveness factor ε.

The recommended face velocities for return inlets are as follows:

Above the occupied zone	800 to 1000 fpm (4 to 5 m/s)
Within the occupied zone	400 to 600 fpm (2 to 3 m/s)
Door louvers	300 to 500 fpm (1.5 to 2.5 m/s)

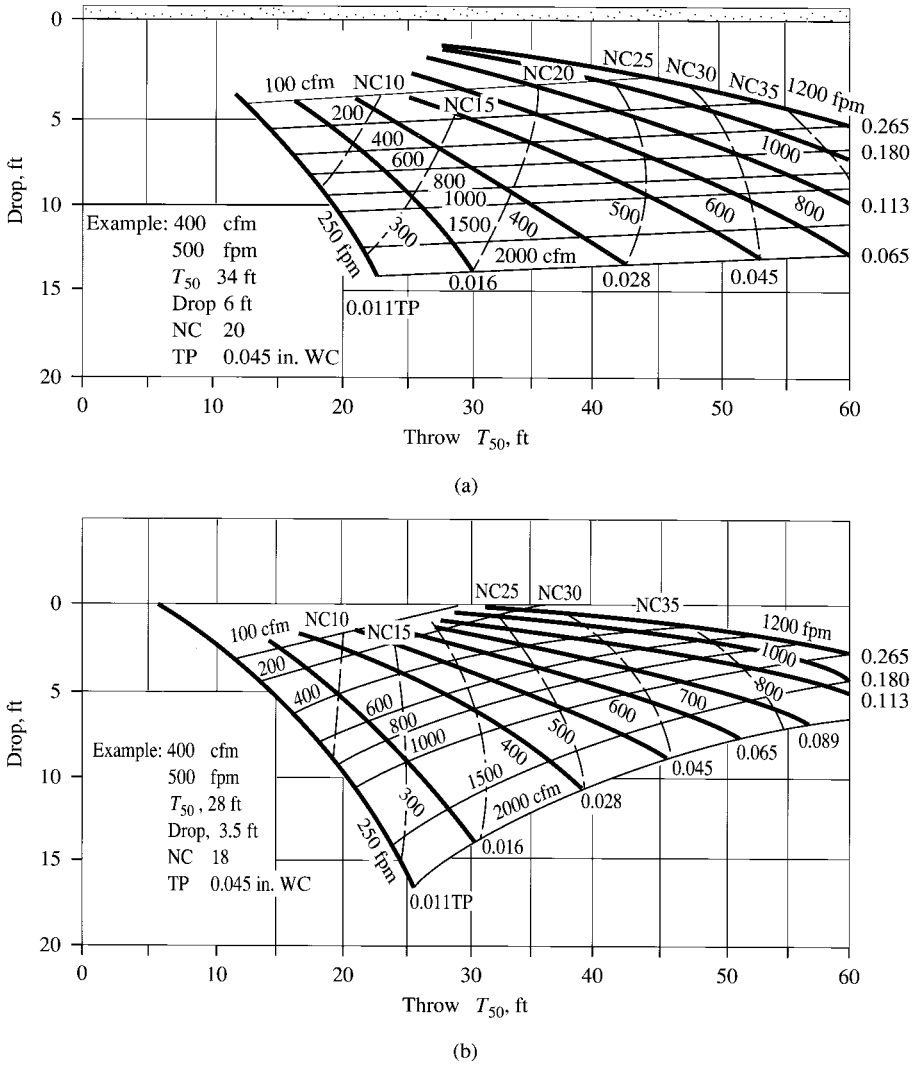

FIGURE 18.21 Performance of cold air jets based on a supply temperature differential of 20°F. (*a*) Performance of cold air jets from typical grilles and registers mounted within 1 ft below ceiling; deflection: vertical 5° up, horizontal, 0°. (*b*) Performance of free cold air jets from typical grilles and registers; deflection: vertical 15° up, horizontal, 0°. (*Source: Titus Products. Reprinted with permission.*)

Example 18.1. A large, open office has a perimeter length of external wall of 30 ft (9.1 m) and a depth of 35 ft (10.7 m) of which 15 ft (4.5 m) is the perimeter zone, as shown in Fig. 18.22. The supply air volume flow rates and load densities for both perimeter and interior zones are as listed below:

Design supply volume flow rate, cfm (L/s)

Perimeter zone	960 (453)
Interior zone	500 (236)

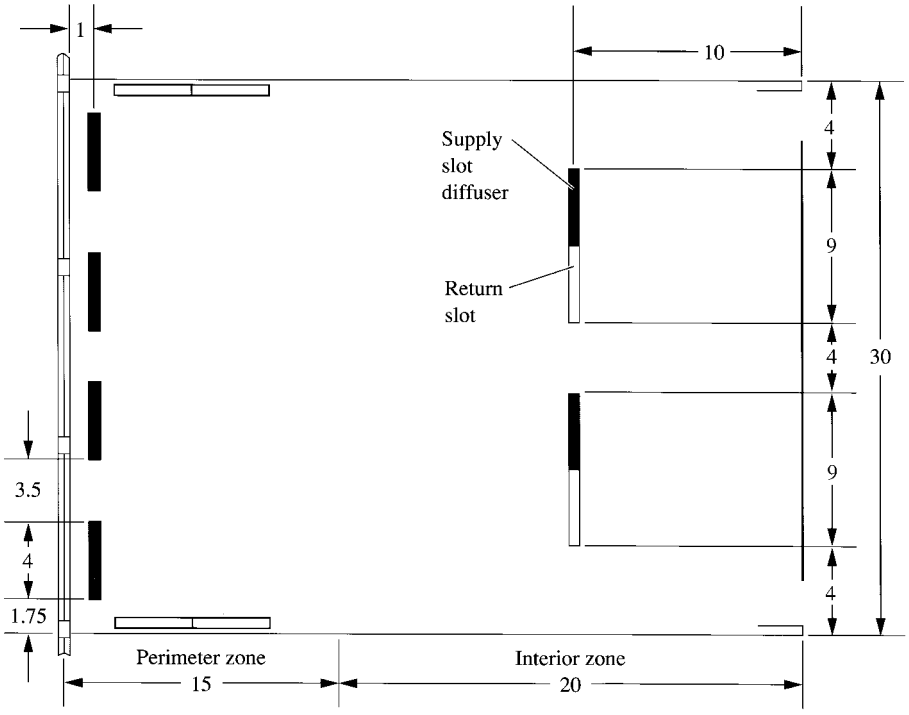

FIGURE 18.22 Layout of slot diffusers for a large open office in Example 18.1. All dimensions in feet.

Supply temperature differential	15°F (8.3°C)
Sound level	*NC* 40
Cooling load density in perimeter zone	50 Btu/h· ft^2 (158 W/m^2)
Linear density of heating load	300 Btu/h· ft (288 W/m)

Design a space air diffusion system for this office for year-round operation using a VAV system. Its minimum supply volume flow rate is 50 percent of the design volume flow at part-load operation. The ceiling plenum is used as a return plenum.

Solution

1. For a large, open office using a VAV system, a space air diffusion system of mixing airflow pattern using ceiling slot diffusers is a suitable choice because of its good surface effect and wider T_{100}/L ($T_{0.5}/L$) range.

2. For the perimeter zone, the overhead two-way two-slot diffuser should be located on the suspended ceiling parallel to the external wall and window glass. One of the slots should discharge downward toward the window glass. The space between the slot diffusers and the window glass is 1 ft (0.3 m). Another slot discharges horizontally inward from the window glass.

3. In Table 18.3 the range of T_{100}/L for slot diffusers having ADPI $>$ 80 percent at a load density of 50 Btu/h· ft^2 is between 0.3 and 0.95. Considering the reduction of supply volume flow rate at 50 percent load, select a T_{100}/L ratio around 0.7. Because the characteristic length L in the perimeter zone is $L = 15 - 1 = 14$ ft, the required throw for the slot diffuser in the perimeter zone

is $T_{100}/L = 0.7$, or

$$T_{100} = 0.7L = 0.7 \times 14 = 9.8 \text{ ft}$$

From Table 18.1, two-way two-slot diffusers with a length of 4 ft (1.2 m) and a slot width of 0.75 in. (19 mm) are selected. At a slot intensity of 30 cfm/ft (46 L/s·m), its performance is as follows:

Throw	
Horizontal projection, ft (m)	8-10-15 (2.4-3-4.6)
Vertical projection T_{50} ($T_{0.25}$), ft(m)	14 (4.3)
Total pressure loss	
Horizontal, in. WC (Pa)	0.091 (23)
Vertical, in. WC (Pa)	0.067 (17)
Sound level, *NC* curve	29

4. If the supply volume flow rate in the perimeter zone is evenly split between horizontal and downward airstreams, the number of slot diffusers is $0.5(960)/(4 \times 30) = 4$, so four slot diffusers should be used. For the downward discharge slot, the vertical throw $T_{150} \approx 0.5\ T_{50} \approx 0.5 \times 14 = 7$ ft, which is greater than the required throw T_{150} (ceiling height 9 − distance from floor level 5) = 4 ft. The terminal velocity is higher than 150 fpm (0.75 m/s) near the window glass at a level of 5 ft from the floor level.

For the horizontally discharged slot, T_{100} is 10 ft. Because the slot is installed 1 ft away from the window, $T_{100}/L = (10 + 1)/15 = 0.73$. From Table 18.3, at a load density of 50 Btu/h·ft^2, the range of T_{100}/L ratio for ADPI > 80 percent is between 0.3 and 0.95; $T_{100}/L = 0.73$ is a suitable setup.

5. Heating load in the perimeter zone is

$$Q_{rh} = 30 \times 300 = 9000 \text{ Btu/h}$$

If the supply volume flow in heating mode operation is still 960 cfm, and if the density of supply air is 0.072 lb/ft^3, then the supply differential can be calculated as

$$T_s - T_r = \frac{Q_{rh}}{60\dot{V}_s \rho_s C_{pa}}$$

$$= \frac{9000}{60 \times 960 \times 0.072 \times 0.243} = 8.9°\text{F } (5°\text{C})$$

This value is far smaller than 15°F (8.3°C).

As the terminal velocity is approximately equal to 150 fpm (0.75 m/s) near the window glass at a level 5 ft (1.5 m) from the floor level, and a far lower $T_s - T_s$, cold draft does not occur at the inner glass surface at a heating load linear density of 300 Btu/h·ft (288 W/m^2).

6. Four 4-ft two-slot, of 0.75-in. slot width, return slots of the same configuration as the supply slot diffusers with a slot intensity of $900/(2 \times 4 \times 4) = 28$ cfm/ft (43 L/s·m) are installed in the perimeter zone. Their locations are arranged on two sides of the room in a direction perpendicular to the supply outlets shown in Fig. 18.22 for a better effectiveness factor ε.

Because the total pressure TP loss of return slots is nearly equal to the TP of the vertical projection of the same type of supply slot diffusers, from Table 18.1, for a slot intensity of 28 cfm/ft, TP for selected return slots is 0.06 in. WC (15 Pa).

7. The depth of the interior zone is 20 ft. It is better to select two 4-ft two-way two-slot supply slot diffusers with 0.75-in. slot width and located at the midpoint of the interior zone. Each one-way single-slot diffuser needs a throw of $20/2 = 10$ ft. From Table 18.1, for a slot intensity of $500/(2 \times 4 \times 2) = 31$ cfm/ft, horizontal throw T_{100} is about 10 ft and the ratio $T_{100}/L = 10/10 = 1.0$. Such a setup is beneficial during part-load operation.

The return slots in the interior zone must have the same total pressure loss TP = 0.06 in. WC, i.e., the same slot intensity, as in the perimeter zone. This is because both perimeter and interior zone inlets are installed on the same ceiling and use the ceiling plenum as the return plenum.

For the interior zone, return slots with a slot intensity of 28 cfm/ft = 500/28 = 18 ft (5.4 m) must be installed. Select two 5-ft two-way two-slot return slots with a slot width of 0.75 in. The diffusers will be located on the ceiling at a position midway from the depth boundaries of the interior zone, as shown in Fig. 18.22.

8. In the perimeter zone, during cooling mode operation of 50 percent part-load operation, the throw–characteristic length ratio may reduce to

$$\frac{T_{100}}{L} = \frac{4.5}{14} = 0.32$$

In the interior zone,

$$\frac{T_{100}}{L} = \frac{4.5}{0.5 \times 20} = 0.45$$

Both are still within 0.3 to 1.4 at a load density 25 Btu/h· ft^2.

9. During heating mode part-load operation, the supply air temperature differential, the buoyancy effect, and the vertical temperature gradients will be smaller than those at design load.

18.7 DISPLACEMENT FLOW AND UNIDIRECTIONAL FLOW

Displacement Flow

Displacement flow is a flow pattern in an air conditioned space in which cold supply air, at a velocity nearly equal to the required velocity in the space, enters the occupied zone or working area and displaces the original space air with a pistonlike airflow without mixing the supply air and the original space air.

Compared with mixing flow, displacement flow provides a better indoor air quality in the occupied zone. If cold air is supplied at a velocity nearly equal to the mean air velocity in the occupied zone with a small supply temperature differential for comfort air conditioning systems, there will be low turbulence intensities and fewer draft problems. On the other hand, displacement flow usually requires a greater supply volume flow rate and a higher construction cost.

There are three types of displacement flow: unidirectional flow, downward uniform flow, and stratified displacement flow.

Unidirectional Flow

In a unidirectional flow, the conditioned supply airstream flows in the same direction as a uniform airflow and showers the entire working area or occupied zone. Supply airstreams occupy the entire working area or occupied zone of the conditioned space.

Types of unidirectional flow currently used in clean rooms include downward unidirectional flow and horizontal unidirectional flow, as shown in Fig. 18.23. Unidirectional flow refers to airflow patterns in which the stream lines of airflow are uniform and move in the same direction. Because of the unidirectional flow, contaminants generated in the space cannot move laterally against the downward airflow, and dust particles will not be carried to higher levels by recirculating flow or large eddies.

Such an airflow pattern in clean rooms was formerly known as laminar flow. Laminar flow is often confused with the types of fluid flow distinguished by Reynolds number in fluid mechanics.

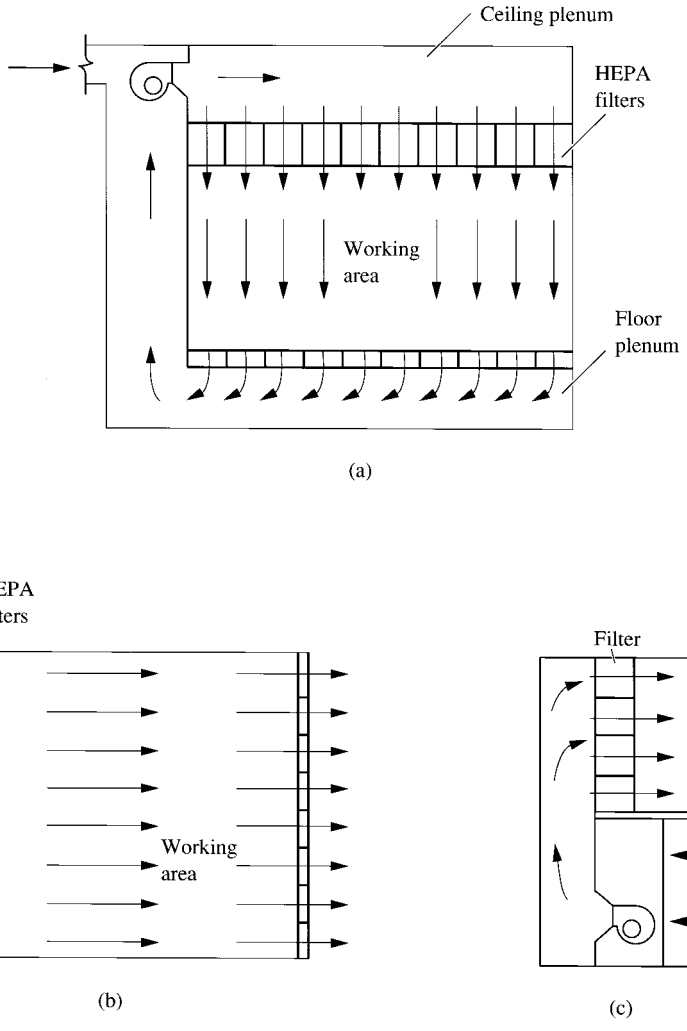

FIGURE 18.23 Unidirectional flow for clean rooms: (*a*) downward unidirectional flow for clean room; (*b*) horizontal unidirectional flow for clean room; (*c*) unidirectional flow for clean workstation.

Most forced airflows driven by fans in air conditioned space are turbulent flow except the laminar flow in the boundary layers adjacent to the surface of the building envelope. The Reynolds number of forced airflow, even at very low air velocity, is usually greater than 1×10^4. As of 1991, *ASHRAE Handbook 1991, HVAC Applications*, refers to this flow pattern as unidirectional flow.

Unidirectional Flow for Clean Rooms

Figure 18.23*a* shows a downward unidirectional flow for clean rooms. After flowing through the high-efficiency particulate air (HEPA) filters or ultralow penetration (ULPA) filters, ultraclean air discharges uniformly downward in parallel stream lines and enters the working area. It then

flows through the raised floor grating and is returned to the recirculating air unit at the top of the clean room.

Figure 18.23*b* shows a horizontal unidirectional flow for a clean room. Instead of flowing downward, the clean airstream discharges horizontally from the HEPA filters on one side of the room and flows through the working area. The contamination level near the return inlets of a horizontal unidirectional flow may be higher than those of a downward unidirectional flow.

Figure 18.23*c* shows a clean workstation that provides a small noncontaminated working area for a single worker by means of horizontal or downward unidirectional flow and HEPA filters. Clean workstations can achieve a high degree of contamination control over a limited area for many practical applications.

To provide parallel stream lines, air velocity of 60 to 90 fpm (0.3 to 0.45 m/s) is required. Unidirectional flow provides a direct and predictable path of submicrometer-size dust particles and minimizes the opportunities for these particles to contaminate working parts. It also captures internally generated dust particles and carries them away. Most dust particles in unidirectional flow reestablish their parallel stream lines after the downstream eddies of an obstruction.

The supply temperture differential T_r-T_s for unidirectional flow for clean rooms depends mainly on the required space velocity and the cooling loads to be removed within the working area. A case study of a class 10 clean room is discussed in Chap. 30.

Ventilating Ceiling

A ventilating ceiling is sometimes called a perforated ceiling. It creates a downward uniform flow similar to the downward unidirectional flow for clean rooms. In most cases, ventilating ceilings discharge conditioned air through the entire ceiling to form a downward uniform flow, except in the area occupied by light troffers. The primary differences between unidirectional flow for clean rooms and downward uniform flow from ventilating ceilings are as follows:

- Unidirectional flow requires a 60 to 90 fpm (0.3 to 0.45 m/s) air velocity and ultraclean air in the working area, and ventilating ceilings usually have a mean air velocity of less than 15 fpm (0.075 m/s) of conditioned air.
- There is no mixing of supply and space air in unidirectional flow; whereas just below the perforated ceiling, supply air is mixed with the ambient air at a vertical distance of less than 1 ft (0.3 m) in downward uniform flow from the ventilating ceiling.

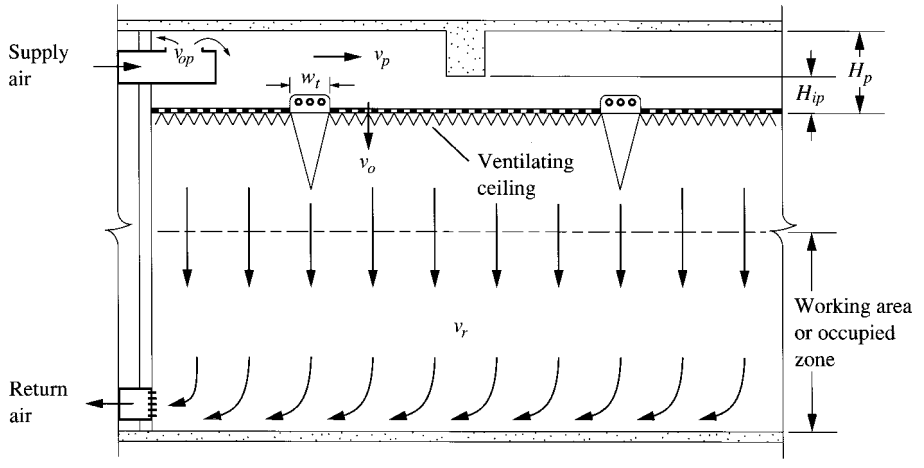

FIGURE 18.24 Ventilating ceiling.

Figure 18.24 shows a downward uniform airflow. In Fig. 18.24, conditioned air is first supplied to the ceiling plenum through supply outlets inside the plenum. It is then squeezed through the holes or slots of the ventilating ceiling and discharged to the conditioned space in a downward uniform flow. When parallel airstreams combine and flow through the working area, they arrive at the floor level, turn to the side return grilles at low levels, and then return to the air-handling unit or packaged unit in the fan room.

Downward uniform airflow patterns from ventilating ceilings have the following characteristics:

- There is no mixing or induced recirculating flow in the working area or occupied zone, so dust particles contained in the space air at a lower level are not carried to a high level to enter the working area again.
- A very low air velocity can be maintained in the working area or the occupied zone even when the volume flow per ft^2 $\dot{V}/A_{fl}$, in cfm/ft^2 (L/s·m^2) is very high. Here $\dot{V}$ represents the supply volume flow rate, in cfm (m^3/s), and A_{fl} represents the floor area, in ft^2 (m^2).
- When $\dot{V}/A_{fl} > 3$ cfm/ft^2 (15 L/s·m^2) the downward uniform flow becomes prominent.

Ceiling Plenum and Supply Air Velocity

To create a more uniform supply air velocity v_o, in fpm (m/s), at the perforated holes or slots, the maximum velocity of the airstream crossing the ventilating ceiling inside the ceiling plenum v_p, in fpm (m/s), should be low. Usually, $v_p \leq 0.5v_o$. Moreover, the clearance between the beam and the ventilating ceiling H_{ip}, in ft (m), should be always greater than 8 in. (200 mm) to prevent a greater pressure loss against the airflow crossing the ventilating ceiling inside the plenum. The greater the supply volume flow rate, the higher H_{ip}. If the ceiling plenum is of sufficient height H_p and few small obstructions other than the beams are present, distributing ductwork inside the ceiling plenum is not necessary to provide a uniform air supply.

Supply air discharged from the outlets into the ceiling plenum always moves in an upward direction so that the velocity pressure of the discharged airstream does not affect the supply air velocity at the perforated openings. The construction of the ceiling plenum must meet the requirements of the National Fire Protection Association (NFPA) and local fire protection codes. Ventilating ceilings are always made of noncombustible materials such as metal strips or mineral acoustic tiles. As a supply plenum, the ceiling plenum must be insulated against heat gain and loss, and condensation on its inner surface if it is adjacent to an area that is not air conditioned.

Supply air velocity at the perforated openings v_o must be optimal. When v_o exceeds 1000 fpm (5 m/s), objectionable noise is generated. A higher v_o also means higher pressure at the ceiling plenum and, therefore, greater air leakage. On the other hand, a lower v_o may cause unevenness in the supply airflow rate because of a higher v_p/v_o ratio. Supply air velocity v_o is usually between 200 and 700 fpm (1 and 3.5 m/s). When v_o has been determined, the perforated area A_o, in ft^2, (m^2) can be calculated as

$$A_o = \frac{\dot{V}_s}{v_o} \tag{18.16}$$

where $\dot{V}_s$ = supply volume flow rate, cfm (m^3/s). Generally, the diameter of the circular holes or the slot width is less than 0.25 in. (6.5 mm). The number of holes or slots can then be determined accordingly. Based on test results, the local loss coefficient of the perforated holes and slots C_o including discharge air velocity pressure at the perforated openings is about 2.75.

Ventilated ceilings are often used in industrial applications where very low air movements or precise control of space temperature up to 68 ± 0.1°F (20 ± 0.056°C) is needed. They are also used in indoor sport stadiums for badminton and table tennis tournaments, where an air velocity of less than 40 fpm (0.2 m/s) should be maintained in the arena.

18.8 STRATIFIED DISPLACEMENT FLOW

Two-Zone Stratified Model

Stratified displacement flow supplies conditioned cold air (in Scandinavian countries usually 100 percent outdoor air at lower velocity) at a low-level supply outlet, as shown in Fig. 18.25. The cold supply air, with a volume flow rate of $\dot{V}_s$, in cfm (L/s), flows in a thin layer along the floor. Above the heat and contaminant sources, heated air containing contaminants rises upward because of its buoyancy effect. Supply air is then entrained into the upward convective flow with a volume flow rate of $\dot{V}_{conv}$, in cfm (L/s). When upward convective flow arrives at a height where its volume flow rate is equal to the supply volume flow rate, that is, $\dot{V}_{conv} = \dot{V}_s$, this height is recognized as the stationary level z_{stat}, in ft (m). Above z_{stat}, ambient air is induced into the upward convective flow until it reaches the ceiling. Upward convective flow with induced air spreads laterally along the ceiling. An amount nearly equal to the supply volume flow rate exhausts or returns to the fan room through the exhaust or return inlet near the ceiling. The remaining portion containing the contaminant descends to the stationary level to be induced into the upward convective flow and recirculated.

The stratified level divides the room vertically into two zones: an upper zone and a lower zone. In the lower zone, only the supply air is induced into the upward convective flow, as its volume flow rate is smaller than the supply volume flow rate $\dot{V}_{conv} < \dot{V}_s$. In the upper zone, the portion of air and contaminants that is greater than the supply flow rate $\dot{V}_{conv} > \dot{V}_s$ recirculates. For cold air supply, the mean temperature of the upper zone is usually 1 to 2°F (0.56 to 1.1°C) higher than that of the lower zone.

Operating Characteristics

Stratified displacement flow (or displacement ventilation) was introduced to Scandinavian countries in the early 1970s as a means of improving general ventilation in industrial applications. For comfort systems in public and office buildings, it was adopted in the early 1980s.

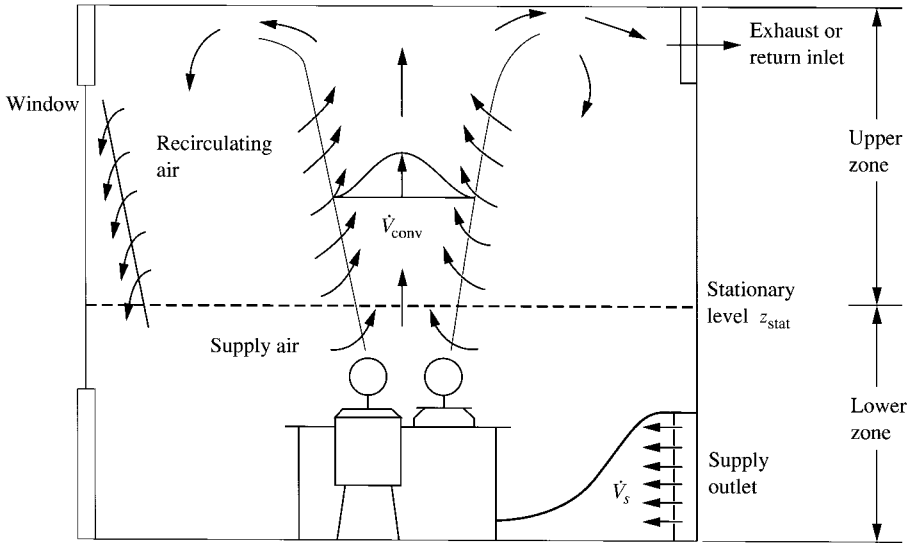

FIGURE 18.25 Stratified displacement flow in a typical room.

According to Svensson (1989), Mathisen (1989), Sandberg and Blomqvist (1989), and Seppänen et al. (1989), stratified displacement flow has the following characteristics:

- Cold air supply of usually 100 percent outdoor air is used to remove cooling loads in conditioned space. Heating is usually provided by radiant panels or a baseboard heater under the windows or on the walls.
- Air must be supplied at low velocity, usually less than 60 fpm (0.3 m/s) and at a height often less than 1.8 ft (0.54 m) from the floor level.
- Cold air is generally supplied at a temperature 5 to 9°F (2.8 to 5°C) lower than that of the air in the occupied zone or working area.
- The height of the lower zone, or z_{stat}, generally should be higher than the breath line of a seated occupant (4.5 ft or 1.4 m). Stationary level depends mainly on the supply volume flow rate $\dot{V}_s$.
- In the lower zone, all air is supply air, theoretically, except the downward cold drafts. Supply air is supplied to the occupied zone directly without mixing with the ambient air.
- Because of the small supply temperature differential, the maximum cooling load that can be removed from the room is considerably less than mixing flow. Some scientists recommend a maximum cooling load of 13 Btu/h·ft^2 (41 W/m^2) at a ceiling height of 9 ft (2.7 m). For greater cooling loads, additional cooling panels mounted on the ceiling or part of space air mixing should be used.
- Return or exhaust inlets are located near the ceiling level.
- According to the required thermal comfort of the occupant, the vertical temperature difference between the 0.3- and 5.5-ft (0.09- and 1.7-m) level as specified in Sec. 4.8 should not exceed 5°F (2.8°C).

Busweiler (1993) reported the use of a combination of radiant cooling, displacement ventilation, and desiccant cooling during the retrofit of the HVAC&R system of a conference room in a hotel in Bremen, Germany. Because of the limited space in the ceiling plenum, air ducts for a conventional air conditioning system could not be installed. The cooled ceiling was used to remove the sensible heat and controlled the room temperature. Displacement ventilation supplied 100 percent outdoor air from wall outlets near to the floor level at a velocity of 40 fpm (0.2 m/s). Desiccant-based cooling was sometimes used to cool and dehumidify the outdoor air and maintain a room temperature of 68°F (20°C) and a relative humidity of about 60 percent. The first-year operation in the Bremen hotel reveals that these systems are a good and practical combination.

Comparison of Stratified Displacement Flow and Mixing Flow

Compared with mixing flow space diffusion systems, stratified displacement flow has the following advantages:

- Contamination level in the occupied zone is lower, and therefore indoor air quality is better.
- Space diffusion effectiveness factors ε_T and ε_C are higher.
- Turbulence intensities I_{tur} are lower, and therefore fewer draft problems occur even at higher mean air velocities.

The disadvantages of the stratified displacement flow are as follows:

- Initial cost is significantly higher if cooling load density is greater than the recommended maximum 13 Btu/h·ft^2 (41 W/m^2) and if cooling panels are added.
- Energy cost is comparatively higher.
- Stratified displacement flow is for space cooling only.

18.9 PROJECTING FLOW—SPOT COOLING/HEATING

Benefits of Projecting Flow

In a projecting flow air pattern, the cold or warm air jet is deliberately projected into part of the occupied zone or working area, which is often called the target zone. The result is control of the environment in a small or localized area, or microenvironmental control. This projecting airflow pattern is used in spot cooling/heating, task air conditioning, or controlling personal environments.

Spot cooling/heating using projecting flow has many advantages over conventional space air diffusion:

- It provides better control of temperature, air cleanliness, and air movements in a localized environment. Spot cooling improves the occupants' thermal conditions significantly, and their heat strain decreases.
- There is greater direct outdoor air supply.
- There is direct and efficient handling of local loads.
- Occupants can have greater control of their own microenvironment or personalized environment.

The main disadvantages of spot cooling/heating include draft discomfort or discomfort due to the pressure of the air jet, limited area of environmental control, and a more complicated space air diffusion system design.

Air jets in projecting flow are usually free jets with high entrainment ratios because of their large contact area with ambient air. Long-throat round nozzles are often used as supply outlets for spot cooling in large spaces.

Two types of projecting flow are currently used today: industrial spot cooling systems and desktop task air conditioning systems.

Industrial Spot Cooling Systems

In an industrial spot cooling system, as shown in Fig. 18.26*a*, the temperature differential between the target zone and the room air (ambient air) is often 5°F (2.8°C) or greater. Melikov et al. (1994) summarized the following based on their and other research workers' experimental results:

Distance Between Target Zone and Supply Outlet. The core zone of a free air jet (from surface of outlet to $x/D_o = 4$) and transition zone (x/D_o between 4 and 12) had a stronger cooling capacity than full developed main zone. This was due to the intensive mixing of the cold jet air with the room air in the main zone. The difference between the target temperature T_j and the outlet temperature T_o increased as the distance between the target zone and the outlet x and the ambient air temperature T_a were increased.

For a target temperature of 77°F (25°C) and an ambient (room) temperature of 86°F (30°C), if $x/D_o = 2$, the required outlet temperature T_o was about 71.6°F (22°C). If $x/D_o = 8$, T_o was about 64.4°F (18°C). The size of the target zone can be estimated since the angle of divergence of the air jet $\alpha = 22°$ and the distance from the outlet x are known. High turbulence intensity caused intensive mixing of the cold air jet and, therefore, decreased the cooling capacity of the air jet.

Vertical Versus Horizontal Jet. Many reseachers preferred a vertical air jet because a horizontal air jet required a large diameter of the jet's outlet and the worker's body was not exposed symmetrically to the cold air shower. In Melikov's experiments, when the room temperature was 82.4°F (28°C), the target velocity v_j preferred by the subjects participating in the experiments was almost identical with vertical and horizontal cooling jets. However, when the room temperatures were 91.4°F (33°C) and 100.4°F (38°C), experimental results confirmed that a vertical jet cooled the subjects more than a horizontal jet.

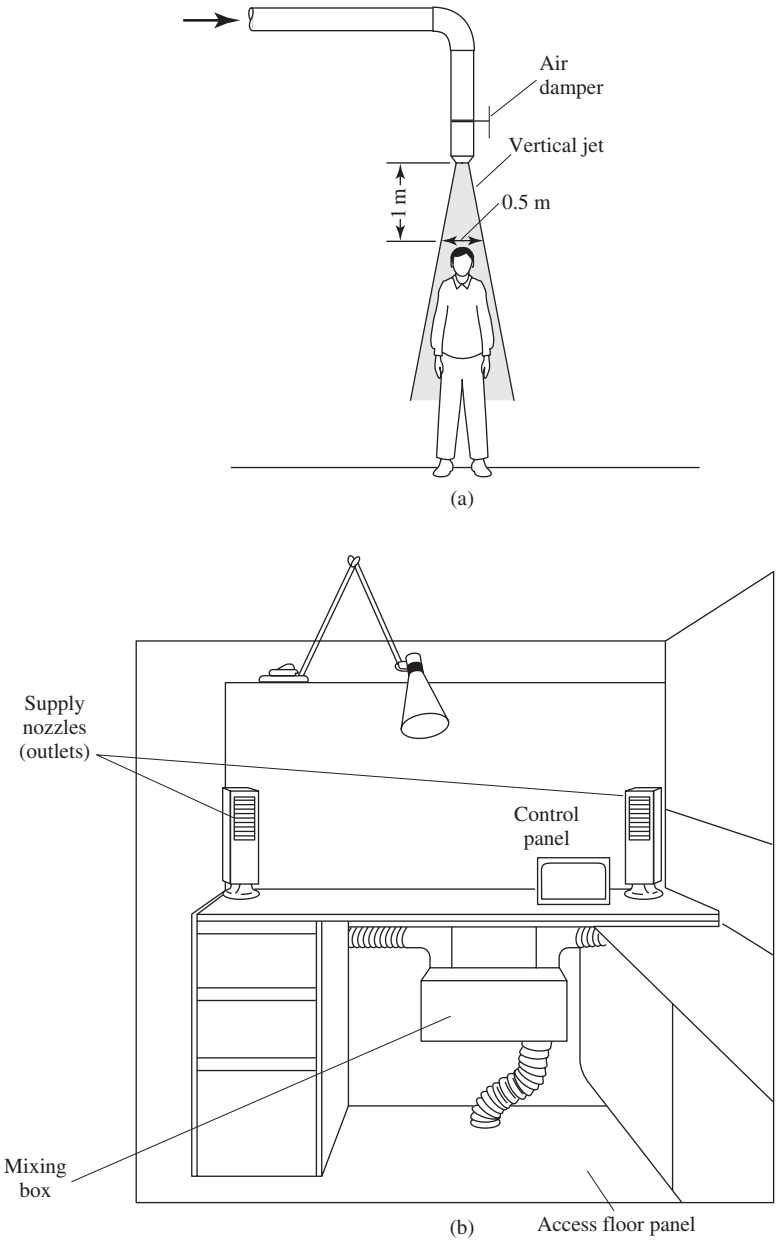

FIGURE 18.26 Projecting flow: (*a*) an industrial spot cooling system; (*b*) a desktop task conditioning system.

Target Velocities. The preferred target velocities v_j of participating subjects at various conditions were as follows:

Study	T_a, °F(°C)	T_j, °F(°C)	v_j, fpm (m/s)	Thermal sensation	Jet type
Melikov et al. (1994)	91.4 (33)	82.4 (28)	280 (1.4)	Slightly warm	Vertical
					Horizontal
Olesen and Nielsen	86 (30)	75.2 (24)	320 (1.6)	Slightly warm	Vertical
(1983)	86 (30)	75.2 (24)	400 (2.0)	Slightly warm	Horizontal
Melikov et al. (1994)	86 (30)	75.2 (24)	440 (2.2)		Vertical
					Horizontal
Fanger et al. (1974)	82.4 (28)	82.4 (28)	160 (0.8)	Neutral	Horizotal
					Whole body
Melikov et al. (1994)	82.4 (28)	82.4 (28)	300 (1.5)		Vertical
	82.4 (28)	82.4 (28)	300 (1.5)		Horizontal

The preferred target velocities show substantial differences from different studies. The amount of the body exposed to the air jet is also an important factor that affects v_j.

Thermal Sensation. Large differences occurred in the local thermal sensation because of the velocity and temperature distribution of the airflow around the human body. Under these circumstances, an occupant's acceptance of the thermal environment showed a compromise between the thermal sensation of the whole body (general thermal sensation) and the thermal sensation for individual parts of the body (local thermal sensation).

When air velocities were above 200 fpm (1 m/s), the airflow exerted a physical pressure on the skin that produced a discomfort. This pressure is the sum of the static and velocity pressure, the total pressure which must take into account the direction of airflow. At high local air temperatures when $T \geq 82.4$°F (28°C), a subject participating in the tests preferred the slight warmth instead neutral sensation because of the annoying effect of the pressure at the high-velocity air jet. At low jet temperatures, local discomfort created by draft was mostly at the head and neck for a vertical jet and at the neck, face, and left shoulder for a left horizontal jet.

Recommendations in Spot Cooling Design

Melikov et al. (1994) and Brown (1988) recommend the following for spot cooling design:

- Allow the occupant to have individual control of the jet velocity and temperature. Spot cooling systems that permit the occupant to adjust the distance between the target zone and the outlet x and the direction of the air jet are preferrable.
- The target zone should be located in the core or transition zone of a cooling jet; typically, $x/D_o =$ 4 to 5 from the nozzle outlet.
- The maximum target velocity (mean value) should not exceed 440 fpm (2.2 m/s), and the maximum local velocity should not exceed 840 fpm (4.2 m/s).
- When ambient air temperature is above 95°F (35°C), use of a combination of a vertical jet, a helmet, and a clothing ensemble to protect the neck from draft is required.

Desktop Task Conditioning Systems

System Description. Bauman et al. (1993) reported the results of laboratory and field measurements investigating the performance of a desktop task conditioning system. A typical desktop task conditioning system, as shown in Fig. 18.26*b*, consists of a self-powered mixing box, two small

fans, two desktop supply outlets, flexible ducts, and a control panel. Inside the mixing box, there is a variable-speed fan to extract primary air (cold conditioned air from the AHUs) either from the low-pressure floor plenum or from the supply duct in the ceiling plenum by means of connecting flexible ducts. Another fan pulls recirculating air from the knee space under the desk. The primary air and recirculating air are mixed together according to the required fractions by regulating the opening of their dampers in each of the transporting ducts. The mixture is then drawn through an electrostatic air filter. The primary air damper is never allowed to close completely, to ensure that the minimum amount of outdoor ventilation air is supplied all the time. The mixing box is hung on the back or on the corner of the desk and is connected to the two supply nozzles (outlets) on the top of the desk by flexible ducts.

Two supply nozzles may be rotated 360° on the horizontal plane. They are installed with horizontal guide vanes which are adjustable ± 30° in the vertical plane.

Control Panel. A desktop DDC unit controller uses adjustable sliders to control the variable-speed fan and modulate the volume flow of the supply air from the nozzles. The supply air temperature is adjusted by varying the opening of the primary and recirculating dampers, and thus the ratio of primary to recirculating air. A 200-W electric radiant heating panel in the knee space is employed during winter heating. The dimming of the occupant's task light controls the lighting luminance and a white noise generator which has a continuous frequency spectrum with equal energy/Hz over a specified frequency range.

A motion detector–based occupancy sensor sends a signal to the DDC unit which shuts the desktop system off when the workstation is unoccupied for a specified time.

Operating Characteristic. Each desktop unit can provide 40 to 250 cfm (20 to 70 L/s) of supply air from the supply nozzles. Even when the two fans in the desktop unit are shut off, 40 cfm (20 L/s) of primary air will still be delivered to the conditioned space for the minimum outdoor ventilation air requirement.

The maximum outlet velocity measured at the face of the 2.3- by 4-in. (58- by 100-mm) supply outlet during laboratory tests varied between 6.5 and 24.5 fps (2 and 7.5 m/s). For a typical desktop system, the primary air temperature supplied from a variable-air-volume (VAV) AHU was 55°F (12.8°C). After mixing with recirculating air, the supply air temperature from the desktop outlet was 65°F (18.3°C).

The supply nozzles are 5 ft (1.52 m) wide by 30 in. (0.76 m) deep at the back corners of the desk. Air is supplied from these two nozzles toward a focus point near the center of the front edge of the desk. For isothermal air jets of horizontal supply with a volume flow of about 90 cfm (43 L/s), the average speed at the work location in front of the desk was 3.3 fps (1 m/s).

Performance of Desktop Task Conditioning Systems

Based on the laboratory investigation and field-measured results, the performance of desktop task conditioning systems was summarized in Bauman et al. (1993) as follows:

- The desktop unit can be controlled to produce a wide range of thermal conditions, allowing occupants the opportunity to fine-tune the local environment to their individual comfort preferences.
- Under the influence of different load density of the adjacent workstation, or under a warm average room air temperature, a desktop unit is capable of being fine-tuned to maintain a nearly comfortable environment, or to maintain a temperature of 1 to 3°F (0.56 to 1.7°C) lower than that of the workstation without the desktop unit, even at relatively low supply air volume flow rates.
- At the same supply volume flow rate, a larger supply nozzle was shown to deliver lower air velocity than the smaller nozzle. A larger nozzle reduces the potential to produce draft while maintaining improved ventilation performance at moderate to high supply volume flow rates.
- Turning off the desktop unit whenever the workstation is unoccupied significantly reduces its energy use.

Application of Desktop Task Conditioning Systems

The first large installation of the desktop system was recently completed in a new office building for an insurance company in West Bend, Wisconsin. This building has 370 desktop units and was fully occupied in July 1991. Reseachers had tracked the productivity of more than 100 employees before and after they moved to the new building. The study has concluded that the desktop system does have a positive impact on worker productivity.

The desktop systems provide excellent individual thermal control for occupants and a direct supply of primary air including outdoor ventilation air to occupants. However, desktop systems often need a floor plenum for primary air supply and a ceiling plenum for return air.

Desktop units and task air conditioning for microenvironmental (personalized environmental) control have great potential because of the smaller controlled environment and therefore lower energy use. Air conditioning integration with furniture may be important in the future.

18.10 UPWARD FLOW UNDERFLOOR AIR DISTRIBUTION

Upward Flow from Floor Plenum

Upward flow from a raised floor has been successfully used for space air diffusion in computer rooms and other industrial applications with high cooling load density, such as 60 to 300 $Btu/h \cdot ft^2$ (189 to 946 W/m^2). Figure 18.27 shows an upward flow space air diffusion from a raised floor plenum—an underfloor air distribution system (an air distribution system consists of ductwork distribution, terminals, and space air diffusion) with the following characteristics:

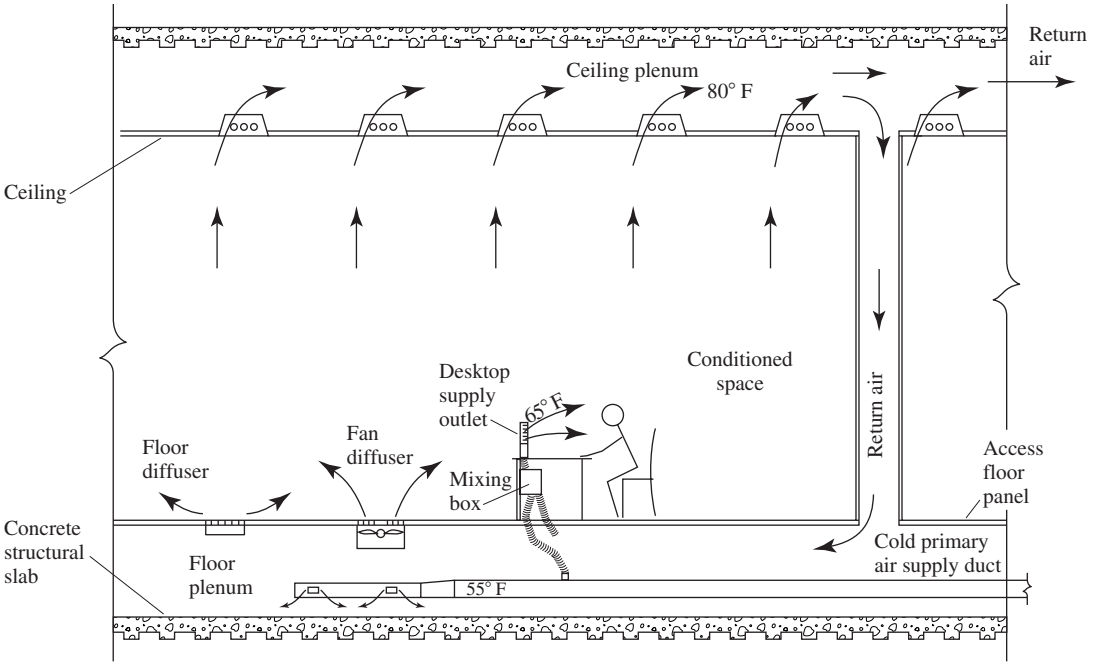

FIGURE 18.27 Upward flow underfloor air distribution system.

- Conditioned air is supplied to an access floor plenum. The floor is usually raised from the structural floor 12 to 18 in. (0.30 to 0.45 m) depending on whether fan-coil units (FCUs) or water source heat pumps (WSHPs) are installed inside the floor plenum. Cable services are usually provided below the access floor.
- Types of supply outlets include floor diffusers, fan-driven floor diffusers, desktop units, and supply outlets from FCUs and WSHPs. The floor plenum can be pressurized at a positive pressure of usually less than 0.08 in. WG (20 Pag) if conditioned air is squeezed out from the floor plenum through floor diffusers directly. For fan diffusers, desktop units, grilles from FCUs, and WSHPs, conditioned air is forced to the occupied zone directly by individually installed small fans. Under these conditions, the floor plenum pressure should be maintained at slightly negative pressure between -0.02 and -0.05 in. WG (-5 and -12.5 Pag).
- In the occupied zone, the airflow patterns of the supply air discharged from the floor diffusers, supply outlets of the FCUs, and WSHPs are mixing flow, and the air discharged from the desktop nozzles is projecting flow. However, from the floor level up to the ceiling plenum, the upward flow is the dominating airflow pattern.
- Cool primary air (including outdoor ventilation air) from the air-handling unit is supplied at neutral pressure to the floor plenum at a temperature of 45°F (7.2°C) for cold air distribution and 55°F (12.8°C) for conventional air distribution. Because all the supply outlets of the underfloor air distribution system are located in the occupied zone, the cool primary air must be mixed with the recirculating air by using fan diffusers, desktop units, FCUs, and WSHPs. The temperature of the mixture at the supply outlet (supply air temperature) is often between 63 and 65°F (17 and 18.5°C).
- After the space cooling load is absorbed by the supply air, the bouyant space air rises. The temperature of the return air entering the light troffers is usually between 80 and 82°F (26.7 and 27.8°C). Return air then enters the ceiling plenum. A part of it is returned to the floor plenum through the return air shafts along the columns, and the remaining part is returned to the fan room through a return system.
- FCUs or WSHPs are often installed inside the access floor plenum and are used to offset the building envelope heat losses and heat gains.

In recent years, there have been rapid developments in office automation. Underfloor air distribution is a new air distribution system that intends to keep in pace with these advances. Shute (1995) summarized the experience of an integrated access floor plenum prototype project which has been in full operation for 4 years as follows:

Thermal Storage of Floor Plenum

The distribution of cold primary air in the floor plenum is not fully ducted. If the floor plenum is composed of a 4-in. (100-mm) concrete structural slab and a concrete-filled access floor, the cold primary air within the floor plenum is in direct contact with a density of 50 lb/ft^2 (244 kg/m^2) thermally active concrete and the floor panel which are potentially thermal storage media.

During a winter night, the mass may provide after-hours low-grade night setback space heating, and creates a cooled reservoir to assist in offsetting a part of the cooling load the next day. On a winter day, part of the internal heat gain is stored, and the recovered heat is used to assist the following night's setback space heating. During the summer, from early morning (4.00 a.m.), the system is started to subcool the mass, which helps to reduce the electric peak demand.

Observations with a laboratory experiment have indicated that a 4-in. (100-mm) structural slab plus the total mass of the access floor panel could participate in a 6 to 7°F (3.3 to 3.9°C) temperature variation cycle within a 24-h period. This allows storage of up to 30 percent of a typical daily cooling load.

Heat Unneutralized

When the supply air from the floor level is returned at the ceiling level, upward airflow lifts the heat gains in the upward flow direction and returns some heat unneutralized to the ceiling. Furthermore, the upward airflow captures most of the lighting heat gain in the return airflow. Research has demonstrated that for a typical office, 15 percent of space heat gain and most of ceiling-level lighting heat gain can be considered unneutralized because of the localized thermal buoyancy. In offices where equipment is concentrated, convective plumes, upward warm expanding airflow above the heat sources, can capture an even greater percentage of space load by the floor-to-ceiling airflow, and results in greater capability to exhaust heat through ceiling return than the conventional space air diffusion system.

Maintaining a Consistent Access Plenum Temperature

Because of the smaller supply air temperature differential $T_r - T_s$ in underfloor air distribution, if the underfloor plenum has hot or cold areas, the occupants' comfort will not be satisfied by pulling air from these localized areas of temperature inconsistency. The following rules are based on test results:

- Supply air colder than 63°F (17.2°C) is perceived as too cold by occupants. Blend air at the floor outlets and at other supply units.
- Primary air degradation becomes unacceptable when it travels a distance exceeding 30 ft (9 m) from the point of air introduction within the access floor plenum. Provide sufficient insulated ductwork to distribute primary air.
- A variation of more than 3°F (1.7°C) across a master control zone may exceed the air transfer fan adjustment range to compensate for this variation. Practical experience limits the master zoning (a zone corresponds to an area served by a VAV box or other terminals) to about 3000 ft^2 (279 m^2) if primary air is supplied from a central riser with VAV control, or a maximum of 10,000 ft^2 (929 m^2) if primary air is supplied from an on-floor fan room.

Floor Plenum Master Zone Air Temperature Control

For a VAV system, as space load varies, the supply volume flow rate supply to each master control zone through an individually controlled VAV box will be varied accordingly. Because of the thermal storage effect, the floor plenum air temperature does not respond as quickly as required. It takes approximately 20 min before the plenum temperature falls sufficiently to call for proper action.

As the space load reduces, more recirculating air is pulled to mix with the cold primary air. The primary air rate becomes proportionally greater in the floor plenum. This effect increases the pressure on the floor plenum, and its pressure becomes positive. The pressurized cold primary air overpowers the normal downward return airflow. If a temperature sensor is located in the downward airflow path, this cold air will be sensed by the sensor, with a resulting rapid reponse to cut the primary air supply flow rate.

Design Considerations

- A structural concrete slab of 4-in. (100-mm) thickness that separates the supply raised floor plenum and the return ceiling plenum is recommended. An attempt to baffle supply and return sections and integrate them in the plenum of the same level has inherent difficulties due to air leakages through plenum baffles, short-circuiting of air in the space, and limiting of the supply air temperature range.

- The diameter of the floor diffuser is usually 8 to 15 in. (200 to 375 mm). The mean air velocity at the outlet is often 200 to 400 fpm (1 to 2 m/s). For a 12-in. (300-mm) floor diffuser, the maximum supply volume flow rate $\dot{V}_{max} \leq 150$ cfm (71 L/s).
- The mean air velocity of a radial air jet of a floor diffuser in which the supply air is discharged in a radial direction along the floor decays more rapidly than a vertical air jet of the same size and supply volume flow rate.
- According to Sodec and Craig (1990), the attenuation of the ductwork between two floor diffusers in adjacent rooms is usually greater than 29 dB. Therefore, conversation transmission through floor diffusers is negligible.

Applications of Upward Flow Underfloor Air Distribution

Compared with a conventional ceiling supply mixing flow air diffusion system, the primary advantages of an upward flow underfloor air distribution system are as follows:

- It keeps pace with the requirements of the raised floor plenum in office buildings of high automation.
- Conditioned air (including outdoor ventilation air) is supplied to the occupied zone directly, which results in a better indoor air quality.
- When air is supplied from the floor diffusers and discharged and returned from the ceiling plenum, usually there is no stagnant air in the occupied zone.
- Upward airflow lifts some unneutralized heat and carries most of the lighting heat gains at high level to the ceiling plenum; these result in a higher return air temperature and, therefore, a higher space diffusion effectiveness factor and a low space cooling load.
- It utilizes the mass heat storage of the access floor panels, and part of the structural concrete slab subcools the floor plenum and reduces the summer peak electric demand charge.

Disadvantages of upward flow underfloor air distribution include the following:

- Valuable floor space may be occupied by the floor diffusers and the buffer zones between the air jets and the workstation.
- Both floor and ceiling plenums are required, and the initial investment is higher than that for conventional ceiling supply mixing flow systems.

For offices and applications other than computer rooms, upward flow underfloor air distribution for commercial buildings was first developed in Germany, and many underfloor air distribution systems have recently been adopted in South Africa, Hong Kong, and Japan. In the 1990s, some modern upward flow underfloor air distribution systems were completed in Canada and the United States. One, an underfloor air distribution system using 370 desktop units for an office building in West Bend, Wisconsin, was fully occupied in July 1991 (Bauman et al., 1993).

Upward flow underfloor air distribution systems are still in the developmental stage. Where a floor plenum is needed, an underfloor air distribution system might be a suitable choice.

18.11 COMPUTATIONAL FLUID DYNAMICS

Computational fluid dynamics (CFD) including heat transfer is a kind of computing technique for the quantitative prediction of air motion and heat transfer, and it presumes that the physical behavior of airflow and thermal systems can be predicted by approximated Navier-Stokes and thermal equations.

CFD Becomes More Popular

HVAC&R engineers traditionally conducted scale-model physical experiments to study air motion and temperature distribution by maintaining similarity through a high Re number and an appropriate Ar corresponding to the adopted scale. CFD was first developed in Europe and Japan. In the late 1980s, ASHRAE organized a research project which thoroughly investigated many issues associated with CFD simulation. In the late 1990s, CFD became more popular in the HVAC&R industry for the determination of locations of supply outlets and return inlets, airflow patterns, air velocity and temperature distributions, and contaminant concentration and removal in an HVAC&R system.

Baker et al. (1997) and Ladeinde and Neavon (1997) summarized the reasons for CFD's popularity:

- CFD has the ability to establish firm quantitative data regarding air motion and can predict fluid characteristics and pressure differentials to very low levels that are essentially impossible during experiments.
- The cost of CFD is lower than that of scale-model experiments.
- Earlier CFD codes were developed on supercomputers. Today computing software is available for simulations to be performed on personal computers.
- People are slowly getting over the fear of using CFD.

Reynolds-Averaged Navier-Stokes Equations

The mathematical expressions describing the basic space-time relationship between mass, velocity, and temperature are expressed in partial differential equations called Navier-Stokes (NS) equations. These equations may be only directly applied to laminar flow fields. Baker et al. (1997) pointed out that CFD modeling based on NS equations requires a large number of assumptions and approximations. These approximations are closely related to the CFD computing results.

For predicting turbulent flow, an approximation called Reynolds averaging (Ra) is used to convert this time-unsteady flow to a mean velocity presentation. A principal assumption for CFD is the turbulence closure model to govern the Reynolds-averaged Navier-Stokes partial differential equations. The selected CFD has been a two-equation turbulent kinetic energy (TKE) model. The TKE model assumes a fully turbulent flow existing everywhere for room airflow prediction. Actually, at normal air change rates per hour, fully turbulent flow occurs only in supply ducts, air jets, downstream of the edge of the obstacles. Elsewhere, the flow is more likely weakly turbulent and actually time-unsteady. A quantitative measurement of the degree of turbulence of an airflow is the turbulence Reynolds number (Re^t) which is defined as the ratio of the turbulence eddy viscosity to the kinematic viscosity: $Re^t = \nu^t/\nu$. For laminar flow, $Re^t = 0$, and for fully turbulent flow $50 < Re^t < 10^3$.

Numerical Methods

Because these Reynolds-averaged partial differential equations using the turbulence Reynold number Re^t are highly nonlinear, they are not solvable by explicit, analytical methods. In these partial differential equations, the velocities u, v, w; and pressure p; temperature T; and some scalar φ are dependent variables to be calculated. Space displacements x, y, z and time t are independent variables. Initial and boundary conditions must be specified. Numerical solution (approximate methods) such as via the finite element and finite volume methods is more popular in CFD simulation.

Ladeinde and Neavon (1997) illustrated the finite element procedure during the calculation of airflow in a duct section. Figure 18.28*a* shows this duct section that contains an inlet and an outlet. The interior of the duct that constitutes the region of airflow (domain of the CFD model)

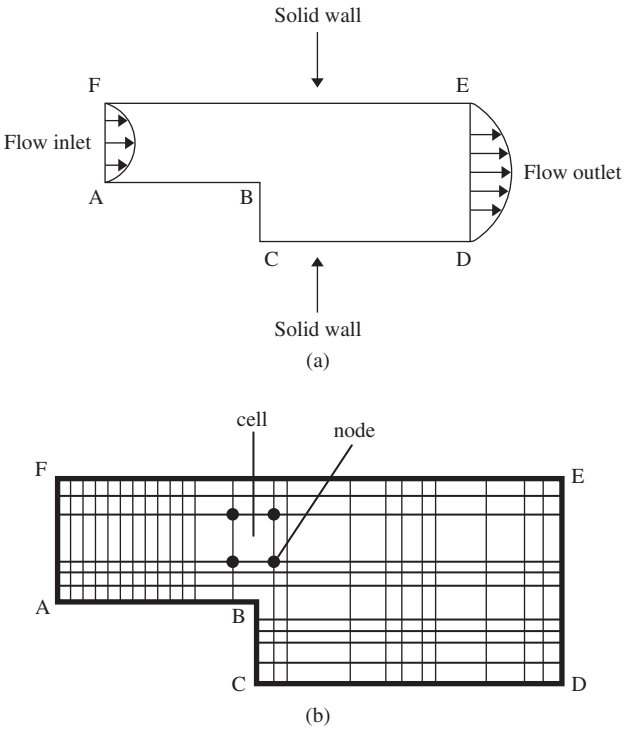

FIGURE 18.28 Calculation of airflow using CFD in a duct section: (*a*) a duct section with sudden expansion; (*b*) finite element model for airflow in a duct section. (*Source: ASHRAE Journal, January 1997. Authors Ladeinde and Nearon. Reprinted by permission*.)

is discretized into nodes and cells as shown in Fig. 18.28*b*. The integration of the equations using the finite element method gives the dependent variables at each node of the CFD model. The graphic of the vectors representing the velocities using the finite element method is shown in Fig. 18.29.

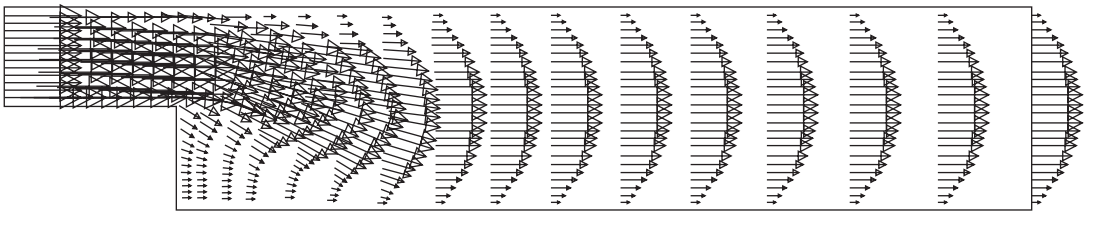

FIGURE 18.29 Velocity vectors of the airflow in a duct section. (*Source: ASHRAE Journal, January 1997. Authors Ladeinde and Nearon. Reprinted by permission*.)

Conducting CFD Experiments

CFD will not replace the physical experiments for airflow and heat transfer in the HVAC&R industry completely; however, it will significantly reduce the physical experiments to study the airflow patterns and the space diffusion effectiveness factor of various space air diffusion systems, the removal of contaminants in clean rooms, and the effectiveness of the introduction of outdoor air to the occupied zone.

Baker et al. (1997) asked, How accurate can CFD be? This depends on the form of Reynolds-averaged Navier-Stokes partial differential (ReNSPD) equations selected as well as the boundary conditions. Therefore, a range of CFD simulations or CFD experiments must be conducted to verify solution sensitivity to model parameters and boundary conditions embedded in the selected ReNSPD equations and their numerical method calculation.

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CHAPTER 19
SOUND CONTROL

19.1 SOUND CONTROL AND SOUND PATHS 19.1
- Sound Control 19.1
- Sound Paths 19.2
- Control at Design Stage 19.3
- Recommended Procedure for Noise Control 19.3

19.2 FAN, COMPRESSOR, PUMP, AND AIRFLOW NOISE 19.4
- Fan Noise 19.4
- Noise from Chillers and Pumps 19.4
- Airflow Noise 19.5

19.3 SOUND ATTENUATION ALONG DUCT-BORNE PATH 19.6
- Sound Attenuation in Ducts 19.6
- Sound Attenuation at Elbows and Branch Takeoffs 19.9
- End Reflection Loss 19.10
- Duct-Borne Crosstalk 19.11
- Attenuation along Duct-Borne Path 19.11

19.4 SILENCERS 19.12
- Types of Silencers 19.13
- Characteristics of Silencers 19.14
- Location of Silencers 19.15
- Active Silencers 19.16
- Selection of Silencers 19.17

19.5 FIBERGLASS IN HVAC&R SYSTEMS 19.17
- Problems 19.17
- Recommendations 19.18

19.6 RADIATED NOISE AND TRANSMISSION LOSSES 19.18
- Breakout and Break-in 19.18
- Duct Rumble 19.19
- Transmission Losses 19.19
- Breakout and Break-in Sound Power Level 19.19
- Transmission Loss for Selected Building Structures 19.23

19.7 RELATIONSHIP BETWEEN ROOM SOUND POWER LEVEL AND ROOM SOUND PRESSURE LEVEL 19.23
- Single or Multiple Sound Sources 19.23
- Array of Ceiling Diffusers 19.24

19.8 NOISE CONTROL FOR A TYPICAL AIR SYSTEM 19.25
- Combination of Supply Fan Noise and Terminal Noise 19.25
- Estimate Sound Pressure Level for Spaces Served by Terminal Units 19.25
- Environmental Adjustment Factor 19.26
- Plenum Ceiling Effect 19.26

19.9 ROOFTOP PACKAGED UNITS 19.29
- Basics 19.29
- Sound Sources and Paths 19.30
- Discharge Side Duct Breakout 19.31
- Sound Path on Return Side 19.31
- Structure-Borne Noise 19.32

REFERENCES 19.32

19.1 SOUND CONTROL AND SOUND PATHS

Sound Control

Sound control for an air conditioned space or, more accurately, an occupied zone is provided to attenuate the HVAC&R equipment sound to an acceptable background level and to provide a suitable acoustic environment for people in the occupied zone. A suitable acoustic environment is as important as a comfortable thermal environment to the occupants. Noise, or any unwanted sound, is always annoying.

The objective of sound control should also be to provide an unobtrusive background sound at a level low enough that it does not interfere with human activities. Sound quality depends on the relative intensities of the sound levels in various octave bands of the audible spectrum. Unobtrusive sound quality means the following: a balanced distribution over a wide frequency range; no tonal characteristics such as hiss, whistle, or rumble; and a steady sound level.

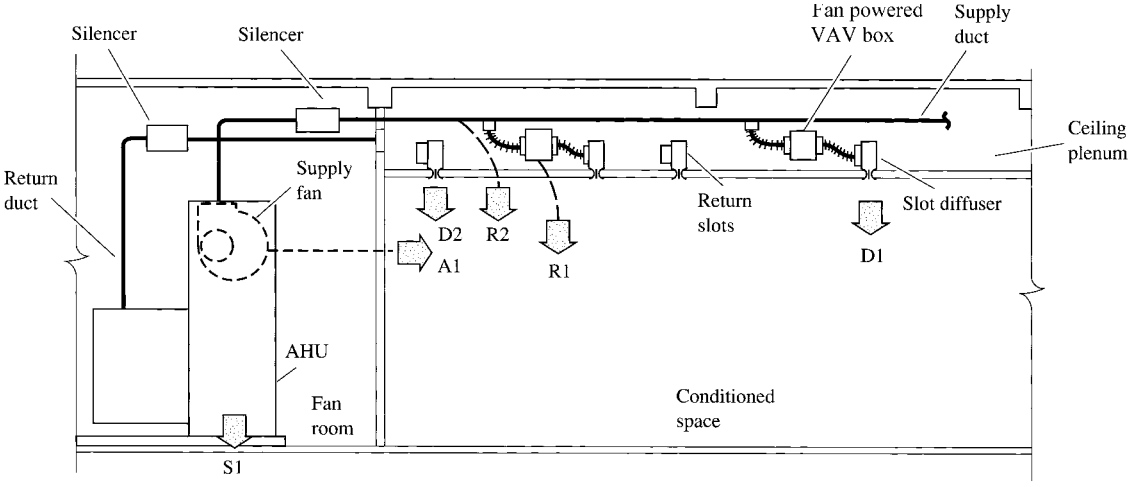
The background sound level to be provided must be optimized. An NC or RC design criterion that is too low always means an unnecessary higher cost. The recommended indoor design RC or NC criteria range is listed in Table 4.8.

Sound Paths

When HVAC&R equipment or a component generates sound, it is often called a sound generator or sound source. Sound created at the source is received by the occupant through various sound transmission paths. HVAC&R equipment that acts as noise generators includes fans, compressors, pumps, and dampers; this noise is in addition to other airflow noise in ducts. Of these, the fan is the major noise source in HVAC&R equipment to be controlled and attenuated because fans are widely used and have more sound transmission paths than compressors and pumps.

Sound transmits from the source to the receiver via the following paths:

- *Duct-borne paths*. These are noise transmission paths through ducts, duct fittings, and other air system components, at either the supply air side or the return air side, as shown in Fig. 19.1. The noise thus transmitted is often called system noise.
- *Radiated sound path*. This is the path along which noise radiates through the duct walls or casings into the ceiling plenum or into an adjacent area and then transmits to the receiver.
- *Airborne path*. This path includes the equipment noise transmitted to adjacent areas through the air.
- *Structure-borne path*. These are the paths along which equipment vibration and noise are transmitted through the building structure.



Duct–borne	D1	Fan → Silencer → Supply duct → Elbows → Branch power division → Flexible duct → Space effect
	D2	Fan → Silencer → Return duct → Elbow → Ceiling plenum → Space effect
Radiated sound	R1	VAV box → Ceiling plenum → Space effect
	R2	Supply duct breakout → Ceiling plenum → Space effect
Airborne	A1	Fan → Fan room → Wall → Space effect
Structure-borne	S1	Fan → Building structure

FIGURE 19.1 Fan room and sound paths.

Structure-borne transmission is actually a combination of sound and vibration effects. This problem is usually solved through the combined effort of the structural and HVAC&R designers.

Control at Design Stage

The evaluation and control of all noise transmission paths should take place at the design stage, and it should be a cooperative effort of the architect and the structural and mechanical engineers. If an excessive noise level occurs due to improper design—the designer fails to analyze the potential noise problem, there are mistakes in analysis, or data are used improperly—remedial measures to reduce the noise level are usually expensive and are often less effective.

One effective measure to reduce noise in buildings is to locate the fan room or other sound source away from critical areas, such as conference rooms and executive offices. For purposes of noise control, the fan room may be located near rest rooms, stairwells, or copy rooms.

For noise control, theoretical estimation and prediction schemes are mainly guidelines. Previous experience and field performance are important. Especially for projects with strict acoustic requirements, field/laboratory tests and checkouts are often necessary.

Recommended Procedure for Noise Control

Before the analysis and evaluation of sound levels, the indoor design NC or RC criteria are usually determined, and the necessary data to evaluate sound levels at various points of interest are collected and investigated. The recommended procedure for noise control uses the following approach:

$$\text{Source} \rightarrow \text{Path (attenuation)} \rightarrow \text{Receiver}$$

The basic procedure is as follows:

1. Determine the sources of the noise from the fan, compressor, and pump, and the sound power level generated. Use certified manufacturer's data.

2. Carefully analyze all the possible sound paths that can transmit noise from the source to the occupied zone. An overlooked sound path may affect the final results.

3. Calculate all the sound attenuation and transmission losses in each sound path during transmission.

4. Determine the airflow noise L_{af}, in duct-borne path due to dampers, elbow, or junctions and the attenuated fan noise near the damper $L_{at,\ fan}$, both in dB. If $L_{at,\ fan} - L_{af} > 8$ dB, L_{af} can be ignored. Only when $L_{at,\ fan} - L_{af} \leq 8$ dB should L_{af} be added to $L_{at,\ fan}$ for duct-borne transmission calculations.

5. For duct-borne paths, determine the attenuated sound power level of fan noise at the supply outlet (system noise or room source L_{wr}), in dB re 10^{-12} W, that affects the receiver. Convert L_{wr} to a sound pressure level received by the receiver at a chosen point in a room L_{pr} or at a plane 5 ft (1.5 m) from the floor L_{pt}, both in dB re 20 μPa.

6. For radiated sound transmissions or airborne transmissions, the sound power level of the sound radiated into the receiving room should be converted to a room sound pressure level.

7. The resultant sound pressure level $L_{\Sigma p}$ at the center frequency of each octave band, in dB, that is perceived by the receiver at a chosen point or at a plane 5 ft (1.5 m) from the floor level is the sum of sound pressure levels from all sound paths. Determine the NC or RC $L_{\Sigma NC}$, in dB, from the calculated $L_{\Sigma p}$ in each octave band.

8. Compare $L_{\Sigma NC}$ with the design criteria NC or RC $L_{p,\ NC}$. If $L_{\Sigma NC} > L_{p,\ NC}$, then one should add a silencer (or an active noise control device, which is discussed in later sections) to the duct-borne sound path, change the configuration of the duct system, or build structures to bring $L_{\Sigma NC}$ down to or slightly below $L_{p,\ NC}$.

9. Check this noise control prediction scheme with the actual field results of projects of similar noise control arrangements, if possible.

19.2 FAN, COMPRESSOR, PUMP, AND AIRFLOW NOISE

Fan Noise

Use the fan noise data provided by the manufacturer. If the manufacturer's data are not available, estimate the sound power level for the fan by Eq. (15.21). The specific sound power levels at full-load operation are listed in Table 15.1.

Fans should be selected for maximum efficiency so as to produce less fan noise. When a fan is operated at off-peak efficiency conditions, according to *ASHRAE Handbook 1999*, *HVAC Applications*, an additional amount up to 6 dB may be added to a plenum fan sound power level operating at the right-hand side of the peak efficiency along the fan curve. Because the calculated system pressure loss is often less than the actual system pressure loss, a centrifugal fan is often selected to operate just at the right-hand side of its rated peak efficiency. When a smaller fan is operated at higher volume flow rates, a noise up to 5 dB will often be produced. Fans should not be operated in surge or stall region at any time because fan noise and vibrations are substantially increased.

According to *ASHRAE Handbook 1999*, *HVAC Applications*, backward-inclined (BI) or air foil (AF) centrifugal fans often have higher sound power level in the mid- and upper-frequency range especially at the blade passing frequency. They are more sensitive to inlet flow distortion than forward-curved (FC) centrifugal fans. When inlet vanes are installed inside the fan inlet, the sound level at the blade passing frequency increases 2 to 8 dB depending on the percentage of the volume flow rate reduced. Externally mounted inlet vanes increase about 2 to 3 dB. BI and AF fans show a narrower operating range and a clearly defined surging region than FC fans. Inside the operating range, BI and AF fans are usually perceived to have lower sound power level at lower frequency (below 100 Hz) than FC fans. For centrifugal fans, air volume flow control by varying fan speed with PWM ac inverter is the acoustically preferred method.

Forward-curved centrifugal fans have a wide operating range without a clearly defined surge region. They are comparatively insensitive to flow distortion at their inlet. In FC fans, modulation of inlet vanes in a VAV system does not cause an increase in sound level at a frequency of 63 Hz and above. FC fans are considered having 31.5- and 63-Hz rumbles.

Plug and plenum fans need a slightly greater energy use, however, they can have a significantly lower sound power level if the fan plenum is properly sized and acoustically treated.

Axial fans are considered to have lower sound power level in lower frequency than centrifugal fans. They are often used in applications where the high-frequency noise can be attenuated by silencers. Axial fans are most sensitive to inlet flow obstructions. Varying blade angle to control volume flow rate is predominately used in vane-axial fans for a VAV system when the system load is reduced. When the air volume flow reduces from 80 to 40 percent, there is a corresponding noise reduction of 2 to 5 dB between 125 and 4000 Hz.

Noise from Chillers and Pumps

If manufacturer's data are not available, according to *ASHRAE Handbook 1991*, *HVAC Applications*, the sound pressure level L_{pA}, in dBA, for centrifugal chillers at a distance 3.3 ft (1 m) from the chiller can be calculated as

$$L_{pA} = 60 + 11 \log \text{TR} \tag{19.1}$$

where TR = refrigeration capacity, in tons. For reciprocating chillers, L_{pA} in dBA at a distance of

3.3 ft (1 m) is

$$L_{pA} = 71 + 9 \log \text{TR} \tag{19.2}$$

The sound pressure level L_p, in dB, at the center frequency of various octave bands can be obtained by adding the following values at each octave band to the calculated L_{pA}:

	Center frequency of octave bands, Hz						
	63	125	250	500	1k	2k	4k
Centrifugal chiller	− 8	− 5	− 6	− 7	− 8	− 5	− 8
Reciprocating chiller	− 19	− 11	− 7	− 1	− 4	− 9	− 14

The centrifugal chiller previously mentioned is constructed hermetically, has internal gears, and operates at medium or full load. At light load operation, L_p may increase 10 to 13 dB for the shaft frequency in the band, 8 to 10 dB in the blade pass frequency band, and 5 dB for the remaining octave bands. For circulating pumps, L_{pA} at a distance of 3.3 ft (1 m) is

$$L_{pA} = 77 + 10 \log \text{hp} \tag{19.3}$$

where hp = power input to the pump, hp.

Airflow Noise

Airflow noise is an aerodynamic noise generated in flow passages or at duct fittings. Airflow noise is the result of vortices passing around obstacles, such as elbows, dampers, branch takeoffs, terminals, and diffusers, that cause local acceleration and deformation and thereby produce local dynamic compression of the air. Airflow noise can also be generated when vortices pass through solid discontinuities.

Airflow noise is mainly determined by the velocity of air flowing through the air passage (duct velocities) and the constrictions due to duct fittings. The geometry of the duct fittings also affects the airflow turbulence and vortices and, therefore, the airflow noise.

Maximum Duct Velocities. Airflow noise can be reduced by using low duct velocities, preventing abrupt changes, and providing sufficient sound attenuation. According to *ASHRAE Handbook 1999, HVAC Applications*, the maximum recommended airflow velocities in ducts or in free openings at various specified RC (NC) ratings are as follows:

		Maximum airflow velocity, fpm			
	RC (NC)	Rectangular duct	Round duct	Supply outlet	Return inlet
In shaft or above drywall ceiling	45	3500	5000		
	35	2500	3500		
	25	1700	2500		
Above suspended acoustic ceiling	45	2500	4500		
	35	1750	3000		
	25	1200	2000		
Within occupied space	45	2000	3900		
	35	1450	2600		
	25	950	1700		
Free openings	45			625	750
	35			500	600
	25			350	425

Branch ducts should have air velocities about 80 percent of the listed value. Airflow velocity of the final runout to outlets should be 50 percent of the listed value. The presence of the elbow, diffuser, and other fittings can increase airflow noise substantially, and therefore the duct velocity should reduce accordingly.

Diffusers and Grilles. The manufacturer's sound rating of a diffuser is obtained with a uniform velocity distribution throughout the neck. If a volume damper is installed immediately before the diffuser, the turbulent airflow at the neck will be significantly increased compared to the manufacturer's laboratory-tested data. This turbulence can be considerably reduced by adding an equalizing grid in the neck of the diffuser. Volume dampers should not be located closer than 5 ft (1.5 m) from a supply outlet.

The sound power level added to the diffuser sound rating is proportional to the damper pressure ratio (DPR). The DPR is equal to the throttled pressure loss across the damper, divided by the minimum pressure loss across the damper when its blades are fully opened. According to the data provided by *ASHRAE Handbook 1999*, *HVAC Applications*, if a volume damper is installed in the neck of a linear diffuser, when the DPR varies from 1.5 to 6, a corresponding 5 to 24 dB should be added to the diffuser sound rating. If the volume damper is installed in the inlet of the plenum of a linear diffuser, when DPR varies from 1.5 to 6, a corresponding of 2 to 9 dB should be added. If the volume damper is installed at least 5 ft (1.5 m) from the inlet plenum of a linear diffuser, only a corresponding of 0 to 5 dB should be added.

Airflow noise generated at the diffusers or grilles at the end of duct-borne paths is difficult to attenuate except by reducing their neck or face velocities. Imbalance of volume flow between various branch runouts may create a greater branch duct velocity and grille face velocity in one of the branch runouts which has the lowest flow resistance.

Poor Fan Entry and Discharge Conditions. Noise is generated because of the abrupt air passage constrictions and sudden changes in airflow direction. Both cause flow turbulence and flow separation as well as energy losses. Such low-frequency noise results in a duct rumble and is very difficult to attenuate. The designer should carefully design the fan intake and discharge connections. Doing so is the best way to control this kind of airflow noise as well as to provide an energy-efficient air system.

19.3 SOUND ATTENUATION ALONG DUCT-BORNE PATH

Sound Attenuation in Ducts

Sound attenuation is the reduction in the intensity of sound, expressed in watts per unit area, as it travels along a sound transmission path from a source to a receiver. Sound attenuation is achieved by (1) the absorption of sound energy by the absorptive material, (2) spherical spreading and scattering, and (3) reflection of sound waves incident upon a surface.

Sound attenuation in duct sections, duct fittings, silencers, and other sound-reducing elements can be indicated by insertion loss. Insertion loss (IL) at a specific frequency is the reduction in sound power level, in dB re 10^{-12} W, measured at the receiver when a sound attenuation element is inserted in the transmission path between the sound source and the receiver. Noise reduction is the difference in sound pressure levels between any two points along the sound transmission path.

Sound absorptivity is the ability of a material to absorb sound energy. When a sound wave impinges on the surface of a porous sound-absorbing material, air vibrates within the small pores. The flow resistance of air and its vibration converts a portion of the absorbed sound energy to heat. The fraction of the incident sound power that is absorbed is called the sound absorption coefficient α. Most sound-absorbing materials have a low α at low frequencies and a higher α at high frequencies. For a typical sound absorption material, α might be equal to 0.15 in the octave band whose center frequency is 63 Hz, and $\alpha = 0.9$ at 1000 Hz.

TABLE 19.1 Approximate Natural Attenuation of Sound in Unlined Rectangular Sheet-Metal Ducts

Duct size, in. × in.	P/A, 1/ft	Attenuation, dB/ft			
		Octave band center frequency, Hz			
		63	125	250	>250
6 × 6	8.0	0.30	0.20	0.10	0.10
12 × 12	4.0	0.35	0.20	0.10	0.06
12 × 24	3.0	0.40	0.20	0.10	0.05
24 × 24	2.0	0.25	0.20	0.10	0.03
48 × 48	1.0	0.15	0.10	0.07	0.02
72 × 72	0.7	0.10	0.10	0.05	0.02

P/A ratio indicates perimeter/area ratio.

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted with permission.

Ducts can be unlined or inner-lined with foil-coated encapsulated fiberglass or other sound-absorptive materials, usually at a thickness of 1 in. (25 mm), sometimes 2 in. (50 mm), to absorb fan noise. Inner-lined ducts have better sound attenuation than unlined ducts, and the inner-lined sound-absorbing layer can also serve as thermal insulation.

In unlined round ducts, the sound attenuation, called natural attenuation, is only about one-tenth that in unlined rectangular ducts as follows:

Diameter, in.	Round duct attenuation, dB/ft			
	Octave band center frequency, Hz			
	<125	250	500	>1000
$D \leq 7$	0.03	0.05	0.05	0.10
$7 < D \leq 15$	0.03	0.03	0.05	0.07
$15 < D \leq 30$	0.02	0.02	0.03	0.05
$30 < D \leq 60$	0.01	0.01	0.02	0.02

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted with permission.

Natural attenuation of unlined rectangular ducts is listed in Table 19.1.

For inner-lined ducts, the sound attenuation, or insertion loss, of a given length of duct section depends on the cross-sectional area and the properties of the absorptive material. Sound attenuation in inner-lined round duct with a 1-in. (25-mm) thickness of duct liner is listed in Table 19.2, and for 1-in-. (25-mm-) thick inner-lined straight rectangular ducts it is listed in Table 19.3.

TABLE 19.2 Round Duct Attenuation (Insertion Loss)

Duct diameter, in.	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
	Approximate attenuation for 1-in. duct liner, dB/ft							
6	0.38	0.59	0.93	1.53	2.17	2.31	2.04	1.26
12	0.23	0.46	0.81	1.45	2.18	1.91	1.48	1.05
24	0.07	0.25	0.57	1.28	1.71	1.24	0.85	0.80
48	0	0	0.18	0.63	0.26	0.34	0.45	0.44

Source: Abridged with permission from *ASHRAE Handbook 1995*, *HVAC Applications*.

TABLE 19.3 Sound Attenuation (Insertion Loss) for Straight Lined Sheet-Metal Rectangular Ducts, Lining Thickness 1 in.

Internal cross-sectional dimensions, in.	Octave and center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
	Attenuation, dB/ft							
8 × 8	0.77	0.90	1.29	2.57	6.23	5.94	3.49	2.42
8 × 18	0.60	0.67	1.04	2.22	5.10	4.72	3.06	2.26
12 × 12	0.56	0.60	0.96	2.09	4.70	4.29	2.90	2.20
12 × 24	0.50	0.48	0.79	1.81	3.85	3.41	2.54	2.06
18 × 18	0.40	0.43	0.74	1.70	3.54	3.10	2.41	2.00
18 × 36	0.30	0.32	0.59	1.47	2.90	2.46	2.11	1.87
24 × 24	0.30	0.32	0.59	1.47	2.90	2.46	2.11	1.87
24 × 48	0.23	0.24	0.47	1.27	2.37	1.95	1.85	1.76
36 × 36	0.21	0.21	0.43	1.20	2.19	1.78	1.75	1.71
36 × 72	0.16	0.16	0.34	1.03	1.79	1.41	1.54	1.60
48 × 48	0.16	0.16	0.34	1.03	1.79	1.41	1.54	1.60
48 × 72	0.14	0.13	0.29	0.94	1.58	1.22	1.42	1.54

1. Based on measurements of surface-coated duct liners of 1.5 to 3 lb/ft^3 density. For the specific materials tested, liner density had a minor effect over the nominal range of 1.5 to 3 lb/ft^3.
2. Add natural attenuation (Table 19.1) to obtain total attenuation.
Source: Abridged with permission from *ASHRAE Handbook 1995*, *HVAC Applications*.

TABLE 19.4 Lined Flexible Duct Insertion Loss

Diameter, in.	Length, ft	Insertion loss dB						
		Octave band center frequency, Hz						
		63	125	250	500	1000	2000	4000
8	12	8	11	21	33	37	37	24
	9	6	8	16	25	28	28	18
	6	4	6	11	17	19	19	12
	3	2	3	5	8	9	9	6
9	12	8	11	22	33	37	36	22
	9	6	8	17	25	28	27	17
	6	4	6	11	17	19	18	11
	3	2	3	6	8	9	9	6
10	12	8	10	22	32	36	34	21
	9	6	8	17	24	27	26	16
	6	4	5	11	16	18	17	11
	3	2	3	6	8	9	9	5
12	12	7	9	20	30	34	31	18
	9	5	7	15	23	26	23	14
	6	3	5	10	15	17	16	9
	3	2	2	5	8	9	8	5
14	12	5	7	16	27	31	27	14
	9	4	5	12	20	23	20	11
	6	3	4	8	14	16	14	7
	3	1	2	4	7	8	7	4

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted with permission.

Lined flexible ducts reduce duct-borne noise significantly. Insertion losses for lined flexible ducts at various diameters and duct lengths are listed in Table 19.4. The length of the flexible duct should normally be from 3 to 6 ft (0.9 to 1.8 m). An abrupt bend may create unacceptably high noise.

The insertion loss (sound attenuation) of lined round, flat oval, rectangular, and flexible ducts is a function of their inside diameter (or the smaller inside dimension of rectangular and flat oval ducts). Generally, as the inside diameter increases, the insertion loss decreases. Surprisingly, the 3 : 1 aspect ratio flat oval duct has approximately the same IL as a round duct with the same diameter as the minor axis.

The thickness of the duct liner has a definite effect on the IL of lined ducts. For duct liner of rectangular duct with a thickness of 2 in. (50 mm), the sound attenuation is nearly double the value of a 1-in. (25-mm) duct liner for octave band center frequencies between 250 and 500 Hz. The ILs are nearly the same for both round and rectangular duct liners with a thickness of 1 in. (25 mm) for octave band center frequencies of 1000 Hz and greater.

According to Bodley (1981), the density of the duct liner material has a minor effect on the IL of inner-lined ducts. The IL is affected by the absorption coefficient of the lining material and its thickness.

Sound Attenuation at Elbows and Branch Takeoffs

Sound is reflected by elbows. Lined or unlined elbows provide sound attenuation to reduce the duct-borne noise. Sound attenuation depends on the size of duct, whether the elbow has a duct lining before and/or after the elbow, and whether turning vanes are provided. The estimated sound attenuation for unlined round elbows is indicated in Table 19.5.

A mitered 90° elbow, often called a square elbow, provides effective noise attenuation by means of reflection loss in the octave bands whose center frequencies equal or exceed 125 Hz. Rumble noise is difficult to attenuate even for a square elbow. In Table 19.6 are listed the insertion losses for elbows that are followed or proceeded by duct lining. Square elbows with turning vanes are slightly less effective in sound attenuation than elbows without turning vanes.

For any branch takeoff from a diverging wye in a main duct, the dominating sound attenuation is the acoustic energy distribution taking place between the main duct and the branch ducts. That part of sound attenuation because of the branch power division $\Delta L_{w,b}$, in dB, is proportional to the ratio of the area of branch A_{br}, in ft^2 (m^2), to the sum of the area of branches ΣA_{br}, in ft^2 (m^2), and can be calculated as

$$\Delta L_{w,b} = -10 \log \frac{A_{br}}{\Sigma A_{br}} \tag{19.4}$$

TABLE 19.5 Insertion Loss of Unlined Round Elbows

	Insertion loss, dB
$fw < 1.9$	0
$1.9 < fw < 3.8$	1
$3.8 < fw < 7.5$	2
$fw > 7.5$	3

$fw = f \times w$
f = frequency, kHz
w = diameter for round duct, in.
Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted with permission.

TABLE 19.6 Insertion Loss of Square Elbows

	Insertion loss, dB	
	Unlined	Lined
Without turning vanes		
$fw < 1.9$	0	0
$1.9 < fw < 3.8$	1	1
$3.8 < fw < 7.5$	5	6
$7.5 < fw < 15$	8	11
$15 < fw < 30$	4	10
$fw > 30$	3	10
With turning vanes		
$fw < 1.9$	0	0
$1.9 < fw < 3.8$	1	1
$3.8 < fw < 7.5$	4	4
$7.5 < fw < 15$	6	7
$fw > 15$	4	7

$fw = f \times w$
f = frequency, kHz
w = duct width or depth of square duct or diameter for round duct, in.

Source: *ASHRAE Handbook 1995, HVAC Applications.* Reprinted with permission.

The sound attenuation of branch power division $\Delta L_{w,b}$ should be subtracted from the sound power level of duct-borne noise at the common end of the diverging wye in the main duct just before the branch takeoff $L_{w,c}$, in dB, to obtain the sound power level of the branch takeoff $L_{w,b}$, in dB,

$$L_{w,b} = L_{w,c} - \Delta L_{w,b} \tag{19.5}$$

If the minor sound attenuations in the diverging wye are ignored, based on the principle of energy conservation, the sum of the sound power levels of the duct-borne noise at the straight-through end $L_{w,s}$, in dB, and the branch takeoff $L_{w,b}$ just after the diverging wye $L_{w,s} + L_{w,b} = L_{w,c}$, from Eq. (19.5); therefore

$$L_{w,s} = \Delta L_{w,b} \tag{19.6}$$

End Reflection Loss

When a sound wave propagates through the end opening of a duct section into a room, a significant amount of sound is reflected back to the duct, and thus substantially reduces the low-frequency sound that reaches the receiver in the occupied zone. According to *ASHRAE Handbook 1995*, *HVAC Applications*, the sound attenuation of end reflection loss for ducts terminated in free space and for ducts terminated flush with a wall is listed in Table 19.7. Diffusers that terminate in a suspended acoustic ceiling can be considered as terminating in free space. In Table 19.7, for a diffuser with a rectangular connecting duct, its diameter $D = \sqrt{4A/\pi}$, in in. (mm), where A = area of the rectangular duct, in.^{2} (mm^2).

Values listed in Table 19.7 were tested based on straight round connecting ducts leading to the diffusers. In actual installation, there may be dampers and guide vanes. The connecting duct between a difffuser and a main duct or terminal may not be straight and smooth. All these factors may affect the listed values in Table 19.7.

TABLE 19.7 Duct End Reflection Loss

Duct diameter, in.	End reflection loss, dB					
	Octave band center frequency, Hz					
	63	125	250	500	1000	2000
Duct terminated in free space						
6	20	14	9	5	2	1
8	18	12	7	3	1	0
10	16	11	6	2	1	0
12	14	9	5	2	1	0
16	12	7	3	1	0	0
20	10	6	2	1	0	0
24	9	5	2	1	0	0
28	8	4	1	0	0	0
32	7	3	1	0	0	0
Duct terminated flush with wall						
6	18	13	8	4	1	
8	16	11	6	2	1	
10	14	9	5	2	1	
12	13	8	4	1	0	
16	10	6	2	1	0	
20	9	5	2	1	0	
24	8	4	1	0	0	
28	7	3	1	0	0	
32	6	2	1	0	0	

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted by permission.

Duct-Borne Crosstalk

Duct-borne crosstalk or plenum crosstalk is the sound transmitted between two rooms through the duct systems or ceiling plenum. Usually, the sound attenuation in the duct system or in the ceiling plenum should be 5 dB greater than the transmission loss of the architectural partition between two rooms. Transmission loss is discussed in detail in the following sections.

Duct lining, devious duct runs, flexible ducts, inner lining in the slot diffusers, and return slots are often used to attenuate the sound transmitted through duct-borne or plenum-connected paths to avoid crosstalk. A well-sealed architectural partition in the ceiling plenum with an inner-lined return air passage effectively prevents crosstalk through the ceiling plenum for applications with strict NC or RC criteria in the occupied zone.

Attenuation along Duct-Borne Path

For a typical VAV system with a supply fan installed in the AHU inside a fan room, as shown in Fig. 19.1, the sound attenuation along the duct-borne paths — the supply duct side and the return duct side — are as follows:

Supply duct-borne path D1	Return duct-borne path D2
Supply side silencer	Return side silencer
Supply duct attenuation	Return duct attenuation
Elbows attenuation	Elbows attenuation

Supply duct-borne path D1	Return duct-borne path D2
Branch power division	Ceiling plenum attenuation or branch power division
Flexible duct attenuation	Return slot attenuation
Room effect	Room effect

On the return air side, an air filter made of a fibrous material may have a certain degree of sound attenuation. This could be considered an added safe factor in the return duct-borne path. Ceiling plenum attenuation is discussed in the following section.

All sound attenuation should be subtracted from the sound power level of the supply fan $L_{w,f}$, in dB re 10^{-12} W, and converted to the sound pressure level at the receiver $L_{p,r}$, in dB re 20 μPa, through the room effect.

19.4 SILENCERS

The purpose of a silencer is to reduce the sound power level of a fan, an airflow noise, or other sound source transmitted along a duct-borne path or airborne path to a required level.

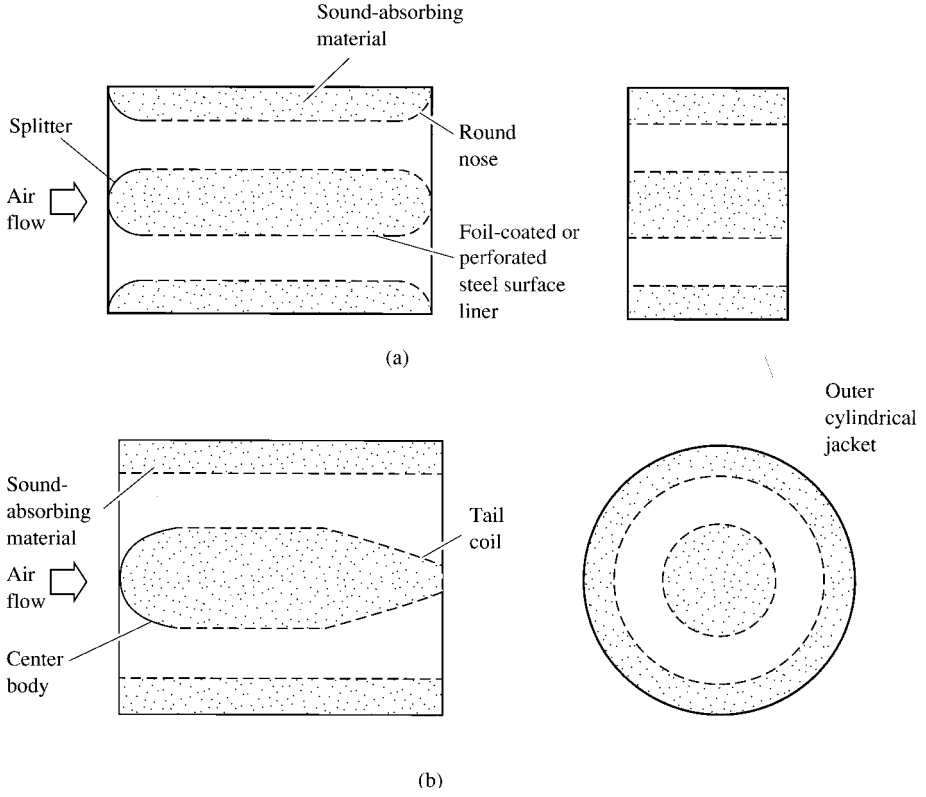


FIGURE 19.2 Rectangular and cylindrical silencers: (a) rectangular silencer and (b) cylindrical silencer.

Types of Silencer

Silencers can be classified into rectanguar, cylindrical, and sound-attenuating plenum according to their configuration. A typical rectangular splitter silencer is shown in Fig. 19.2*a*. Inside the retangular casing are a number of flat splitters, depending on the width of the silencer. These splitters direct the airflow into small sound-attenuating passages. The splitter is made from an envelope containing sound-attenuating material, such as fiberglass or mineral wool, with protected noneroding facing. The thickness of a splitter is often between 1 and 4 in. (25 and 100 mm). Splitters often have a round instead of a flat nose, to reduce their airflow resistance. A rectangular silencer is often connected with retangular ducts or sometimes with rectangular fan intakes and discharges.

A cylindrical silencer has an outer cylindrical jacket and an inner concentric center body, as shown in Fig. 19.2*b*. Both the cylindrical jacket and the center body contain sound-attenuating material and noneroding facing. A cylindrical silencer is often used in conjunction with vane-axial fans and in round duct systems.

A sound-attenuating plenum consists of several splitters in the form of two, three, or four successive square elbows, shown in Fig. 19.3, filled with sound-absorbing material with noneroding

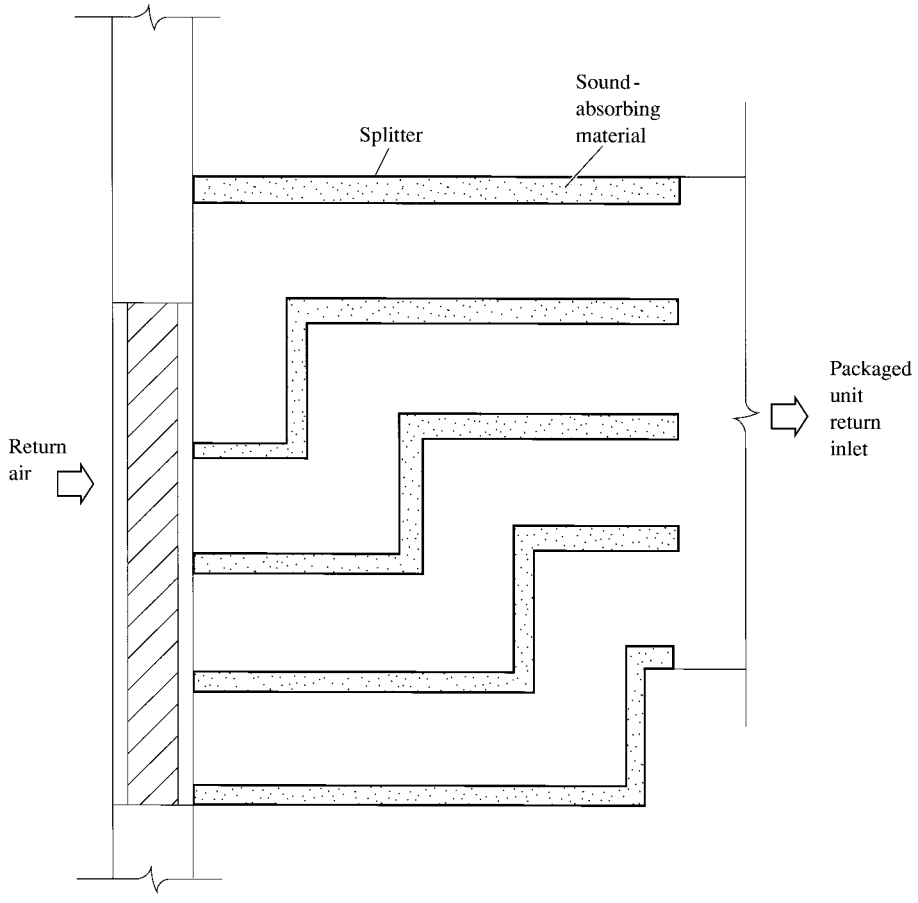


FIGURE 19.3 Sound-attenuating plenum.

facing. A sound plenum effectively attenuates low-frequency noise and is often located between the collective return grille and the return air intake of a packaged unit. A low velocity is important in a sound-attenuating plenum because of the high local loss coefficient of square elbows.

Silencers used in HVAC&R systems can again be classified according to the means of sound attenuation into following types:

1. *Dissipative silencers*. These silencers often use face-covered or encapsulated acoustic material, such as fiberglass, mineral wool, and acrylic polymers, to attenuate noise over a broad range of frequencies. The facing material can be made of
   - Galvanized or aluminum sheet with a perforated area not exceeding 22 percent of the face area
   - Acrylic polymers or polymer sheets
2. *Packless silencers*. There is no fibrous fill. Noise is attenuated by means of acoustically resistive perforations in the splitters. They are often made of sintered aluminum or acrylic plastics.
3. *Reflection-dissipative silencers*. These silencers use the combined effect of sound reflection and dissipation in airflow passages of successive square elbows.
4. *Active silencers*. These silencers produce low-frequency inverse sound waves to cancel the unwanted noise.

Characteristics of Silencers

The acoustic and aerodynamic characteristics of a silencer are mainly indicated by four parameters.

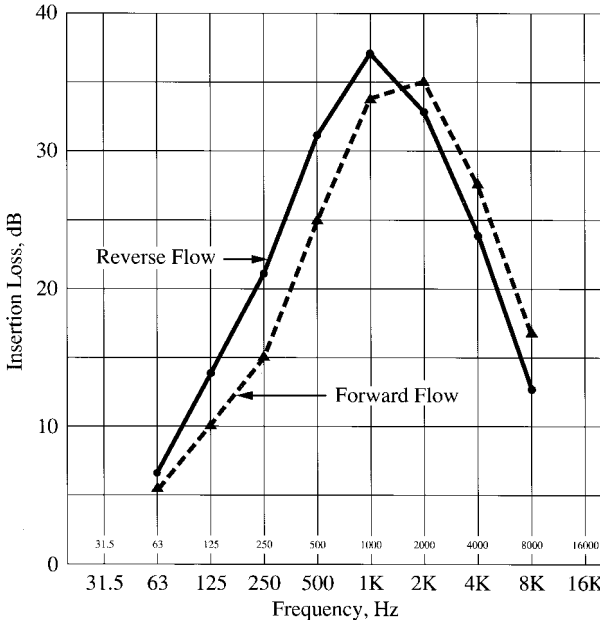


FIGURE 19.4 Difference in insertion loss of a rectangular silencer between forward and reverse flow (at 2000 fpm or 10 m/s face velocity). (***Source**: Handbook of HVAC Design, 1990. Reprinted by permission.*)

Insertion Loss. This is the capacity of a silencer to reduce the sound power level of fan noise or other noise at various octave band center frequencies. The IL of a silencer is affected by the airflow direction, especially when air velocity is greater than 200 fpm (1 m/s). A sound wave that propagates in the same direction as the airflow is said to be in forward flow, and one that propagates opposite to the airflow is said to be in reverse flow. At low frequencies, reverse flow has a longer contact time and, therefore, a higher IL. At high frequencies, sound waves tend to refract toward the absorptive surface in a silencer under reverse flow. Therefore, these effects increase the high-frequency attenuation in the forward flow and decrease the attenuation in reverse flow. The difference in IL between forward and reverse flow of a rectangular silencer at 2000 fpm (10 m/s) face velocity is presented in Fig. 19.4.

Operating Parameters. The operating parameters for a silencer are as follows:

- Volume flow rate of air through the silencer $\dot{V}_{sil}$, in cfm (m^3/s)
- Cross-sectional area of the silencer, including the free area A_{free}, ft^2 (m^2), i,e., area of the passages through which air flows, and the cross-sectional area of the silencer A_{sil}, ft^2 (m^2)
- Free area ratio R_{free}, which is defined as $R_{free} = A_{free}/A_{sil}$
- Face velocity v_{sil}, in fpm (m/s), defined as $v_{sil} = \dot{V}_{sil}/A_{sil}$

Many silencers have a free air ratio between 0.3 and 0.8. The actual mean air velocity inside the airflow passages in a silencer v_{free}, in fpm (m/s), may be 1.25 to 3.3 times the face velocity v_{sil}. The face velocity of a silencer and sound-attenuating plenum is 500 to 2000 fpm (2.5 to 10 m/s).

Self-Noise. This is the lower limit of sound power level, in dB, that a specific silencer can approach at various octave band center frequencies.

Pressure Drop Δp_{sil}. This is the total pressure drop of airstream when it flows through a silencer. Pressure drop Δp_{sil} is a function of its face velocity, free area ratio, length, and the configuration of the splitter or the center body. In general, *ASHRAE Handbook 1995*, *HVAC Applications*, recommends that if $\Delta p_{sil} > 0.35$ in. WC (87 Pa), both v_{free} and airflow noise should be investigated.

Location of Silencers

The location of a silencer to attenuate fan noise should meet the following two requirements:

1. There must be a minimum distance L_{sil} between the upstream fan discharge outlet or other duct fittings and the silencer, to ensure a uniform approach velocity at silencer inlet or an undisturbed discharging velocity at the silencer exit. From the fan discharge, L_{sil} must be equal to or greater than the distance of one duct diameter for every 1000 fpm (5 m/s) average duct velocity. From the fan intake, L_{sil} should be equal to or greater than 0.75 duct diameter for every 1000 fpm (5 m/s) average duct velocity.
2. The silencer should straddle or be adjacent to the solid wall of the fan room (or mechanical room), as shown in Figs. 16.7*a* and 16.8*a*. Such a setup can prevent or reduce the following fan noise transmissions:
   - The fan noise that has not been attenuated by the silencer will not break out into the room adjacent to the fan room.
   - The noise in the fan room may have the chance to break in to the supply main duct after the silencer.

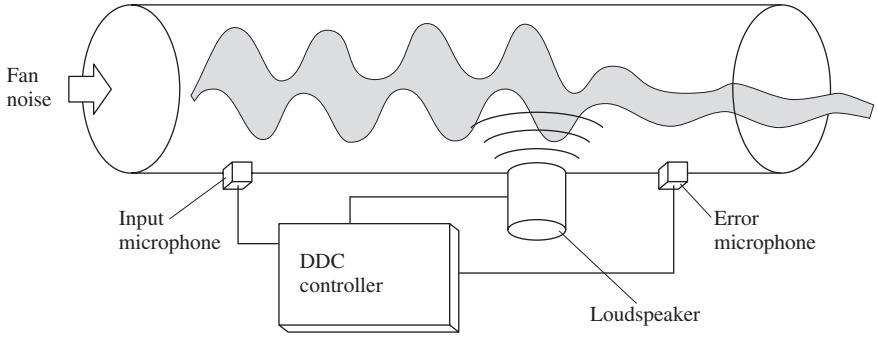


FIGURE 19.5 An active silencer.

Active Silencers

Active silencers use ducted enclosure to cancel duct-borne, low-frequency fan noise (including rumbles) by producing sound waves of equal amplitude and opposite phase. The primary sound source is the unwanted fan noise. The secondary sound source which cancels the unwanted source comprises the inverse sound waves from a loudspeaker.

Operating-Characteristics. An active silencer consists of two or more microphones, a microprocessor-based controller, a loudspeaker, and a ducted enclosure. When a fan noise is propagated down along a duct, as shown in Fig. 19.5, an input microphone located in the airstream measures the noise and sends an electric signal proportional to the sound wave to the controller. The microprocessor-based controller calculates the amplitude, frequency, and phase of the propagating sound and sends a cancel signal to the loudspeaker. The loudspeaker broadcasts sound waves of the same amplitude and frequency as the unwanted noise, but 180° out of phase. The destructive interference between these two sound sources results in the cancellation of the incident fan noise by the secondary source broadcasted by the loudspeaker. The amplitude of the fan noise is reduced downstream of the loudspeaker. An error microphone measures the residual noise. This feedback is used to optimize the performance of the active silencer.

Frequency Limits. Gelin (1997) noted that in an active silencer, the ducted enclosure acts as a waveguide for the sound waves. At lower frequencies, when the wavelength of the sound is at least 2 times longer than the greatest dimension of the waveguide, sound waves may propagate in plain waves in which sound pressure and the phase of the sound wave are uniform across any duct cross section. At intermediate and high frequencies, the wavelength is smaller than a waveguide dimension, and the sound waves may propagate as plain waves as well as more complicated sound waves. A low-frequency fan noise is easily measured, defined, and reproduced 180° out of phase, whereas for a high-frequency noise it is very difficult to do these things. Therefore, active silencers work best at an octave band center frequency range of 31 to 250 Hz.

System Characteristics. The following are the system characteristics of active silencers:

- In an active silencer, sound energy is added to the fan noise to cancel it, whereas in a traditional passive silencer (such as dissipative and packless silencers), sound energy is converted to heat and removed from the system.
- Because the microphone in an active silencer cannot distinguish the fan noise and the turbulent airflow noise, it is acceptable practice that the air velocity in the ducted enclosure of an active silencer

not exceed 1500 fpm (7.5 m/s). When the duct velocity exceeds 2500 fpm (12.5 m/s), the turbulent noise may completely mask fan noise, and the effect of an active silencer is reduced to nil.

- Pressure drop of a passive silencer is often 0.2 to 0.35 in. WC (50 to 63 Pa). The electric energy required to produce the canceling sound is only 40 W, a substantial saving compared to a passive silencer.
- Due to the development of the microprocessor-based technologies, in 1997, the price of the controller used in an active silencer dropped substantially compared to that in 1987.

Performance. Gelin (1997) compared the insertion loss, in dB, of an active silencer, a passive silencer, and a duct liner for a large fan of 5000 cfm (2360 L/s) volume flow rate as follows:

Octave bands, Hz	31	63	125	250	500	1000	2000	4000
Active silencer	5–8	10–15	10–15	2–4	—	—	—	—
1-in. Lined, 84-in.- (2.1-m-) long duct		2	2	4	10	20	17	15
84-in. (2.1-m) Prefabricated silencer		8	17	26	43	43	26	19

The pressure drop of the 84-in. (2.1-m) prefabricated silencer is 0.24 in. WC (60 Pa).

The active silencer has a good sound attenuation in frequencies between 31 and 125 Hz, the duct liner provides effective sound attenuation in frequencies of 500 Hz and more, and the 84-in. (2.1-m) prefabricated silencer is effective in frequencies between 63 and 4000 Hz.

Selection of Silencers

During the selection of silencers, the following points should be considered:

- *Dominating frequencies of noise*. For a low-frequency fan "rumble" noise, an active silencer or a reflective dissipative silencer should be selected. For a broadband noise, a combination of active and dissipative silencers or a dissipative silencer with thick splitters, of 2- to 4-in. (50- to 100-mm) thickness, is preferable.
- *Design requirements*. For clean rooms, fiber-free all-metal silencers are often selected. For health care facilities, fiber-free packless silencers should be selected.
- *Wet surfaces*. If there are wet surfaces in the air system, fiber-free all-metal silencers should be selected.

For a specific type of silencer, variations in its length and free area ratios often meet most of the requirements for IL and pressure drops. Silencers are usually made in lengths of 3, 5, 7, and 10 ft (0.9, 1.5, 2.1, and 3 m). The IL and Δp_{sil} of a silencer are proportional to its length.

A lower face velocity and greater free air ratio for a specific silencer at a given length always result in a lower pressure drop and therefore lower energy consumption. If space is allowed, a lower v_{sil} is beneficial for both IL and Δp_{sil}.

19.5 FIBERGLASS IN HVAC&R SYSTEMS

Problems

Fiberglass-made duct liners, duct boards, silencers, and insulation layers have been widely used in HVAC&R systems for decades. On June 24, 1994, the U.S. Department of Health and Human Services (DHHS) announced that fiberglass will be listed as a material "reasonably anticipated to be a

carcinogen." According to ASHRAE, recently the use of fiberglass in several institutional, eduational, and medical projects has been banned or severely limited because of the concerns that the fibers may be carcinogenic and fiberglass may promote microbial growth.

The International Agency for Research on Cancer (IARC) has performed extensive research concerning the carcinogenicity of fiberglass materials and found inadequate evidence to link insulation of fiberglass with cancer in humans. IARC does indicate a possible link to cancer from glass wool based on heavy dosage in animals. The Canadian Environment Protection Act indicated that studies have failed to show any evidence that fiberglass is carcinogenic and classified it as "unlikely to be carcinogenic to humans."

Concerning the fiber erosion, more recent studies reported in the 6th International Conference on Indoor Air Quality and Climate in Helsinki, Finland, 1993 that airborne fiber levels in buildings with these products range from nondetectable up to only 0.006 fiber per cm^3 (1 $in.^3$ = 16.4 cm^3).

Both liquid water and dirt are required for mold and microbial growth, and microbial growth can occur on any surface in an HVAC&R system. Tinsley (1998) concludes that ". . . the presence of high humidity air, by itself, is not sufficient to cause fungal growth. . . . fungal growth occurred only where both standing water and nutrients (in the form of dirt, pollen, or other organics) could be found." Control of the wetted surface and dirt in the duct system is the most effective method to reduce potential microbial growth.

Recommendations

1. For health care facilities, schools, semiconductor fabricating, food processing, pharmaceutical manufacturing, clean rooms, and many more demanding projects, fiberglass duct liners and fiberglass silencers should be replaced by any one or a combination of the following alternatives:

- Foil-coated products such as acrylic polymer-coated fiberglass duct liners or silencers
- Fibrous acoustic infill (fiberglass or mineral wool) encapsulated inside the polymer sheets
- Fiber-free sound absorber using sintered aluminum or acrylic polymer with microperforations
- Active silencers

2. For air systems that have wetted surfaces (standing water) from outdoor air intakes, cooling coils, coil condensate droplet carryover, drain pans, and humidification processes, especially atomizing devices, direct evaporative coolers, and air washers, all-metal fiber-free silencers or active silencers should be used to attenuate duct-borne fan and airflow noise, if required. Interior surface of ducts that contact with supply air must be galvanized metal sheet. Duct insulation should be wrapped externally.

3. For air systems that employed draw-through fans and medium- or higher-efficiency air filters and had no wetted surfaces (standing water) in contact with the air in the air system (except in the coil section and condensate pan) having a face velocity of the cooling or DX coil of 500 fpm (2.5 m/s) or less, and properly designed, installed, operated, and maintained, fiberglass duct liners and fiberglass silencers with perforated metal facing sheet can be used.

Periodic monitoring of the moisture and dirt conditions of the fiberglass duct liner and silencers must be provided.

19.6 RADIATED NOISE AND TRANSMISSION LOSSES

Break-out and Break-in

When a duct section enters the ceiling plenum or directly passes over occupied spaces carrying duct-borne fan noise that is not well attenuated, the noise radiates through the duct wall, raises the

sound power level in the ceiling plenum or the occupied zone, and is transmitted to the receiver. Noise that radiates through the duct wall and causes the duct wall to vibrate, caused either by internal sound waves or by airflow turbulence, is called breakout. Noise can also be transmitted into a duct and then travel along the duct-borne path, either discharging into a space through the duct opening or breaking out into ambient air, where it is transmitted to the receiver. The transmission of external noise into a duct section through the duct wall is called break-in.

Breakout fan noise from the discharge duct directly under a rooftop packaged unit into the ceiling plenum may cause a serious acoustic problem in the underlying conditioned space.

Duct Rumble

Duct rumble is the result of air pressure fluctuations caused by variations in the speed of the fan, fan motor, or fan belt or due to the airflow instabilities transmitted to the fan housing or nearby ductwork. When the air pressure fluctuations are exerted on large, flat, and unreenforced duct surfaces that have resonance frequencies near or equal to the fluctuating frequencies, the duct surfaces vibrate. Poor fan outlet connections often generate duct rumbles. Duct rumble can produce sound pressure levels of 65 to 95 dB at frequencies from 10 to 100 Hz.

Changing the fan speed as well as the frequencies of the air pressure fluctuations is one method to reduce duct rumble. Another method is to increase the rigidity of the duct wall and thus change the resonance frequencies. Low-frequency duct rumble noise is very difficult to attenuate by duct liners.

Transmission Loss

Transmission loss (TL) is the reduction of sound power when sound is transmitted through a wall, a partition, or a barrier. The relationship between transmission loss and mass of the material indicates that the sound transmission loss of a homogeneous solid partition is a function of its mass and the frequency of sound f transmitted through it. This relationship can be expressed in the following form:

$$\mathrm{TL} = F(\log m_s, \log f) \tag{19.7}$$

where m_s = surface density of homogeneous partition, $\mathrm{lb/ft^2}$ ($\mathrm{kg/m^2}$). Concrete walls and galvanized sheet ducts show that for each doubling of the mass partition, TL increases about 2 to 3 dB for low-frequency noise (less than 800 Hz), and TL increases about 5 to 6 dB for high-frequency sound (800 Hz and over).

The TL-mass relationship is valid only when sound is incident on the surface of the partition in a normal direction and is an approximation. Actual measured data may show considerable deviation from the predicted values. Factors that cause deviations are nonhomogeneity, cracks, stiffness, and resonance. Cracks and gaps around doors, windows, duct and piping sleeves, or other openings on the partition or wall may considerably reduce TL. The TL of a well-sealed partition of 100 $\mathrm{ft^2}$ (9.3 $\mathrm{m^2}$) surface area may drop from 40 to 20 dB because of a total of 1 $\mathrm{ft^2}$ (0.093 $\mathrm{m^2}$) of openings on that partition.

Breakout and Break-in Sound Power Level

According to *ASHRAE Handbook 1999*, *HVAC Applications*, the breakout sound power level radiated from the outer surface of duct walls $L_{w,\mathrm{out}}$, in dB, can be calculated as

$$L_{w,\mathrm{out}} = L_{w,\mathrm{in}} + 10\log\frac{S}{A_i} - \mathrm{TL}_{\mathrm{out}} \tag{19.8}$$

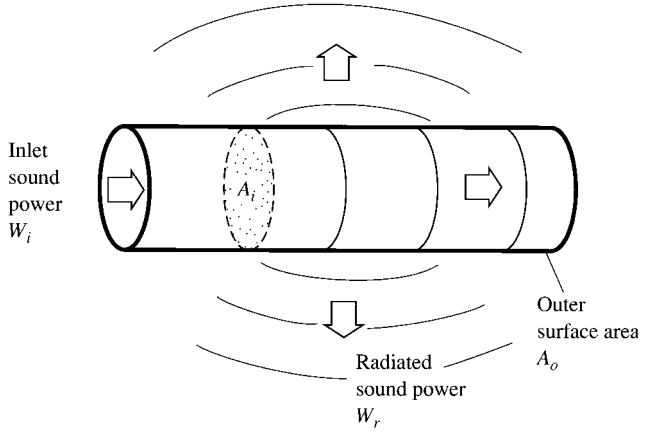


FIGURE 19.6 Breakout transmission loss TL_{out}.

where $L_{w,in}$ = sound power level of sound inside duct, dB
S = surface area of outer sound-radiating duct surface, in.^{2} (mm^2)
A_i = inner cross-sectional area of duct, in.^{2} (mm^2)
TL_{out} = duct breakout transmission loss, dB

Equation (19.8) assumes no sound attenuation along unlined interior duct surface. Such a breakout transmission model is shown in Fig. 19.6.

TABLE 19.8 Duct Breakout Transmission Loss TL_{out} for Round Duct at Various Octave Band Center Frequencies

Diameter, in.	Length, ft	Gauge	TL_{out}, dB						
			Octave band center frequency, Hz						
			63	125	250	500	1000	2000	4000
Long seam ducts									
8	15	26	>45	(53)	55	52	44	35	34
14	15	24	>50	60	54	36	34	31	25
22	15	22	>47	53	37	33	33	27	25
32	15	22	(51)	46	26	26	24	22	38
Spiral-wound ducts									
8	10	26	>48	>64	>75	72	56	56	46
14	10	26	>43	>53	55	33	34	35	25
26	10	24	>45	50	26	26	25	22	36
26	10	16	>48	53	36	32	32	28	41
32	10	22	>43	42	28	25	26	24	40

Note: In cases where background sound swamped the sound radiated from the duct walls, a lower limit on TL_{out} is indicated by a > sign. Parentheses indicate measurements in which background sound has produced a greater uncertainty than usual.
Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted by permission.

TABLE 19.9 TL_{out} for Rectangular Ducts at Various Octave Bands, dB

Duct size		Octave band center frequency, Hz							
in.	Gauge	63	125	250	500	1000	2000	4000	8000
12 × 12	24	21	24	27	30	33	36	41	45
12 × 24	24	19	22	25	28	31	35	41	45
12 × 48	22	19	22	25	28	31	37	43	45
24 × 24	22	20	23	26	29	32	37	43	45
24 × 48	20	20	23	26	29	31	39	45	45
48 × 48	18	21	24	27	30	35	41	45	45
48 × 96	18	19	22	25	29	35	41	45	45

The data are tests on 20-ft-long ducts, but the TL values are for ducts of the cross section shown regardless of length.
Source: Adapted with permission from *ASHRAE Handbook 1991, HVAC Applications*.

As a result of duct sound breakout, the sound pressure level in an occupied space L_p, in dB, can be calculated as

$$L_p = L_{w,\text{out}} - 10\log \pi rL + 10 \tag{19.9}$$

where r = distance between duct and receiver's position where L_p is calculated, ft (m)
L = length of duct sound-radiating and breakout surface, ft (m)

The break-in sound power level transmitted into duct and then transmitted upstream or downstream $L_{w,\text{in}}$, in dB, can be calculated as

$$L_{w,\text{in}} = L_{w,\text{out}} - \text{TL}_{\text{in}} - 3 \tag{19.10}$$

where TL_{in} = duct break-in transmission loss, dB.

Table 19.8 lists the duct breakout transmission losses TL_{out} for round ducts at various octave band center frequencies, Table 19.9 lists TL_{out} for rectangular ducts, and Table 19.10 lists TL_{out} for flat oval ducts. Also Table 19.11 lists the duct break-in transmission losses TL_{in} for round ducts at various octave band center frequencies, Table 19.12 lists TL_{in} for rectangular ducts, and Table 19.13 lists TL_{in} for flat oval ducts.

In lower frequencies, round ducts have the highest TL_{out} and TL_{in}, and rectangular ducts have the lowest TL_{out} and TL_{in} among these three kinds of ducts.

TABLE 19.10 TL_{out} for Flat Oval Ducts at Various Octave Bands, dB

Duct size (a × b)		TL_{out}, dB							
		Octave band center frequency, Hz							
in.	Gauge	63	125	250	500	1000	2000	4000	8000
12 × 6	24	31	34	37	40	43	—	—	—
24 × 6	24	24	27	30	33	36	—	—	—
24 × 12	24	28	31	34	37	—	—	—	—
48 × 12	22	23	26	29	32	—	—	—	—
48 × 24	22	27	30	33	—	—	—	—	—
96 × 24	20	22	25	28	—	—	—	—	—
96 × 48	18	28	31	—	—	—	—	—	—

The data are tests on 20-ft-long ducts, but the TL values are for ducts of the cross section shown regardless of length.
Source: Adapted with permission from *ASHRAE Handbook 1991, HVAC Applications*.

TABLE 19.11 TL_{in} for Round Ducts at Various Octave Bands, dB

Diameter, in.	Length, ft	Gauge	TL_{in}, dB: Octave band center frequency, Hz 63	125	250	500	1000	2000	4000
			Long seam ducts						
8	15	26	>17	(31)	39	42	41	32	31
14	15	24	>27	43	43	31	31	28	22
22	15	22	>28	40	30	30	30	24	22
32	15	22	(35)	36	23	23	21	19	35
			Spiral-wound ducts						
8	10	26	>20	>42	>59	>62	53	43	26
14	10	26	>20	>36	44	28	31	32	22
26	10	24	>27	38	20	23	22	19	33
26	10	16	>30	>41	30	29	29	25	38
32	10	22	>27	32	25	22	23	21	37

Note: In cases where background sound swamped the sound radiated from the duct walls, a lower limit on TL_{in} is indicated by a > sign. Parentheses indicate measurements in which background sound has produced a greater uncertainty than usual.

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted by permission.

TABLE 19.12 TL_{in} for Rectangular Ducts at Various Octave Bands, dB

Duct size		Octave band center frequency, Hz							
in.	Gauge	63	125	250	500	1000	2000	4000	8000
12 × 12	24	16	16	16	25	30	33	38	42
12 × 24	24	15	15	17	25	28	32	38	42
12 × 48	22	14	14	22	25	28	34	40	42
24 × 24	22	13	13	21	26	29	34	40	42
24 × 48	20	12	15	23	26	28	36	42	42
48 × 48	18	10	19	24	27	32	38	42	42
48 × 96	18	11	19	22	26	32	38	42	42

The data are tests on 20-ft-long ducts, but the TL values are for ducts of the cross section shown regardless of length.

Source: Adapted with permission from *ASHRAE Handbook 1991, HVAC Applications*.

TABLE 19.13 TL_{in} for Flat Oval Ducts at Various Octave Bands, dB

Duct size in. × in.	Gauge	TL_{in}, dB: Octave band center frequency, Hz 63	125	250	500	1000	2000	4000
12 × 6	24	18	18	22	31	40	—	—
24 × 6	24	17	17	18	30	33	—	—
24 × 12	24	15	16	25	34	—	—	—
48 × 12	22	14	14	26	29	—	—	—
48 × 24	22	12	21	30	—	—	—	—
96 × 24	20	11	22	25	—	—	—	—
96 × 48	18	19	28	—	—	—	—	—

Note: The data are for duct lengths of 20 ft, but the values may be used for the cross section shown regardless of length.

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted by permission.

TABLE 19.14 Transmission Loss of Some Building Structures

Building structures	Octave band center frequency, Hz						
	63	125	250	500	1000	2000	4000
4-in. dense concrete or solid concrete block, 48 lb/ft^2	32	34	35	37	42	49	55
4-in. hollow-core dense aggregate concrete block, 28 lb/ft^2	29	32	33	34	37	42	49
8-in. hollow-core dense aggregate concrete block	31	33	35	36	41	48	54
Standard drywall partition, $^5/_8$-in. gypsum board on both sides of 2-in. × 4-in. wood studs	12	17	34	35	42	38	44
Standard drywall partition, two layers of $^5/_8$-in. gypsum board on each side of 3 $^5/_8$-in. metal studs	25	36	43	50	50	44	55
$^1/_2$-in. plate glass	11	16	23	27	32	28	32
Double glazing, two $^1/_2$-in. panes, $^1/_2$-in. airspace	12	16	23	27	32	30	35
$^5/_8$-in. Gypsum board ceiling	9	15	20	25	31	33	27
Roof construction, 6 in. thick, 20 gauge (0.0396-in). steel deck with 4-in. lightweight concrete topping, $^5/_8$-in. gypsum board ceiling on resilient hangers	25	41	47	56	65	68	69
*Plenum/ceiling							
*Plenum/ceiling cavity effect: lay-in mineral fiber tile $^5/_8$ in., 35 lb/ft^3		5	9	10	12	14	15
*Plenum/ceiling cavity effect: finished sheetrock, $^5/_8$ in.	10	15	21	25	27	26	27
Acoustic equipment housing, 20-gauge steel outer shell, 2-in.-thick acoustic insulation, 22-gauge (0.0336-in.) perforated inner shell	15	18	21	39	38	49	55
Solid-core wood door, normally closed†		23	27	29	27	26	29

* *ASHRAE Transactions*, 1989, Part I.
† *Handbook of HVAC Design*, 1990.
Source: *ASHRAE Handbook 1991, HVAC Applications*.

Transmission Loss for Selected Building Structures

The transmission losses for selected building structures — including walls, partitions, window glass, ceiling and plenum, and acoustic equipment housing — are listed in Table 19.14. These are mainly abridged from *ASHRAE Handbook 1991*, *HVAC Applications*. In Table 19.14, the combined plenum/ceiling cavity effect indicates the combined effect of the plenum sound absorption and the transmission through the ceiling material. These values are based on data from several manufacturers' laboratory and mock-up spaces.

19.7 RELATIONSHIP BETWEEN ROOM SOUND POWER LEVEL AND ROOM SOUND PRESSURE LEVEL

The sound pressure level at a given location in a room corresponding with a particular point source or an array of multiple sound sources is a function of the sound power level and transmission characteristic of the sound source, the room's acoustical properties, and the distance between the sound source and the receiver.

Single or Multiple Sound Sources

Based on field measurements, Schultz (1985) recommended the following empirical formula to estimate the room sound pressure level L_{pr}, in dB re 20 μPa, from the room sound power level of

single or multiple sound sources at a chosen point in a room:

$$L_{pr} = L_{wr} - 5 \log V - 3 \log f - 10 \log r + 25 \tag{19.11}$$

where L_{wr} = sound power level of room source, dB re 10^{-12} W
V = volume of room, ft^3 (m^3)
f = octave band center frequency, Hz
r = distance from source to receiver or reference point, ft (m)

For a single sound source in the room, Eq. (19.11) can be applied directly. If there are multiple sound sources, estimate L_{wr} for each sound source and sum the contributions to obtain the total sound pressure level at the receiver due to multiple sound sources.

The difference between the room sound pressure level and the room sound power level $L_{pr} - L_{wr}$ is called the *room effect*.

Array of Ceiling Diffusers

Offices and large rooms in commercial buildings often have several ceiling diffusers mounted in a distributed array on the ceiling. Although the sound pressure levels decrease as the vertical distance from the ceiling increases, the sound pressure level along any selected horizontal plane, such as a reference plane of 5 ft (1.5 m) from the floor level, is nearly constant. According to *ASHRAE Handbook 1995*, *HVAC Applications*, the calculation of sound pressure level for a distributed array of ceiling diffusers at a reference plane of 5 ft (1.5 m) from the floor level can be greatly simplified as

$$L_{p5} = L_{w,s} - D \tag{19.12}$$

where $L_{w,s}$ = sound power level of single outlet, dB re 10^{-12} W. Values of D are listed in Table 19.15.

In Eqs. (19.11) and (19.12), the calculation of sound pressure level L_p is based on the assumption that the acoustic characteristics of a room range from average to medium dead, which is generally true for most rooms. For rooms that are acoustically medium-live to live (have little sound absorption), Eqs. (19.11) and (19.12) can overestimate the decrease in L_p.

TABLE 19.15 Values for D in Eq. (19.12)

Floor area per diffuser, ft^2	Value for D, dB						
	Octave band center frequency, Hz						
	63	125	250	500	1000	2000	4000
Ceiling height 8 to 9 ft							
100 to 150	2	3	4	5	6	7	8
200 to 250	3	4	5	6	7	8	9
Ceiling height 10 to 12 ft							
150 to 200	4	5	6	7	8	9	10
250 to 300	5	6	7	8	9	10	11
Ceiling height 14 to 16 ft							
250 to 300	7	8	9	10	11	12	13
350 to 400	8	9	10	11	12	13	14

Source: *ASHRAE Handbook 1995, HVAC Applications*. Reprinted by permission.

19.8 NOISE CONTROL FOR A TYPICAL AIR SYSTEM

In many commercial buildings using a variable-air-volume air system employing VAV boxes, there are two noise sources, as shown in Fig. 19.1:

- Fan noise due to the supply fan of the air-handling unit usually installed in a fan room
- Airflow noise due to the VAV box with a single-blade damper installed inside the ceiling plenum

Sound is transmitted from these two sources to the receiver in the conditioned space via the following paths:

- Duct-borne path, supply side
- Duct-borne path, return side
- Radiated noise breakout from the supply duct in the ceiling plenum
- Radiated sound breakout from the VAV box casing in the ceiling plenum
- Airborne path through fan room walls to adjacent area

Combination of Supply Fan Noise and Terminal Noise

Variable-air-volume boxes are tested for radiated sound power level and discharge-side sound power level in accordance with ADC/ARI Industry Standard 880-83, and the results are listed in the manufacturers' catalogs. The attenuated duct-borne sound power level of the supply fan from the air-handling unit at terminal unit L_{AHUi} should be added to the discharge sound power level from the VAV box L_{ter} to form a resultant sound power level if their difference $L_{ter} - L_{AHUi} \leq 8$ dB. If their difference is greater than 8 dB, the smaller of these two sound power levels can be ignored.

Estimate Sound Pressure Level for Spaces Served by Terminal Units

Recently, the ADC and ARI jointly developed Industry Standard 885 to estimate the space sound pressure levels when terminal units are installed inside the ceiling plenum. Consider a terminal unit connected to two slot diffusers through flexible ducts to serve a conditioned space, as shown in Fig. 19.7, where C represents the terminal casing, D the duct-borne or discharge side, D_1 and D_2 the discharge outlets 1 and 2, and O_1 the supply outlet 1. Another outlet, O_2, is not shown in Fig. 19.7. Along with the arrows, B represents the transmission loss of the breakout from duct and casing, P the sound attenuation due to the plenum ceiling, and S the room effect.

There are altogether seven sound paths by which the receiver can hear fan, terminal, and slot diffuser noises.

Path 1: Radiated sound power level of the terminal unit → plenum ceiling → room effect → receiver

Path 2: Discharge sound power level → breakout → plenum ceiling → room effect → receiver

Path 4: Duct-borne sound power level at the outlet of slot diffuser → room effect → receiver

Path 5: Sound power level of the slot diffuser → room effect → receiver

Sound path 3 transmits radiated sound breakout from another flexible duct after the terminal unit, and it is similar to path 2. The duct-borne noise is transmited along sound path 6 at another slot diffuser and is similar to sound path 4. The airflow noise of another slot diffuser transmits along sound path 7 and is similar to sound path 5.

The radiated sound power level that breaks out at the inlet of the terminal unit has the same kind of sound path as paths 2 and 3. The sound power level of duct-borne noise at D_{o1} equals the sound power level at D_1 or D_2 minus the sound attenuation of the flexible duct and the slot diffuser.

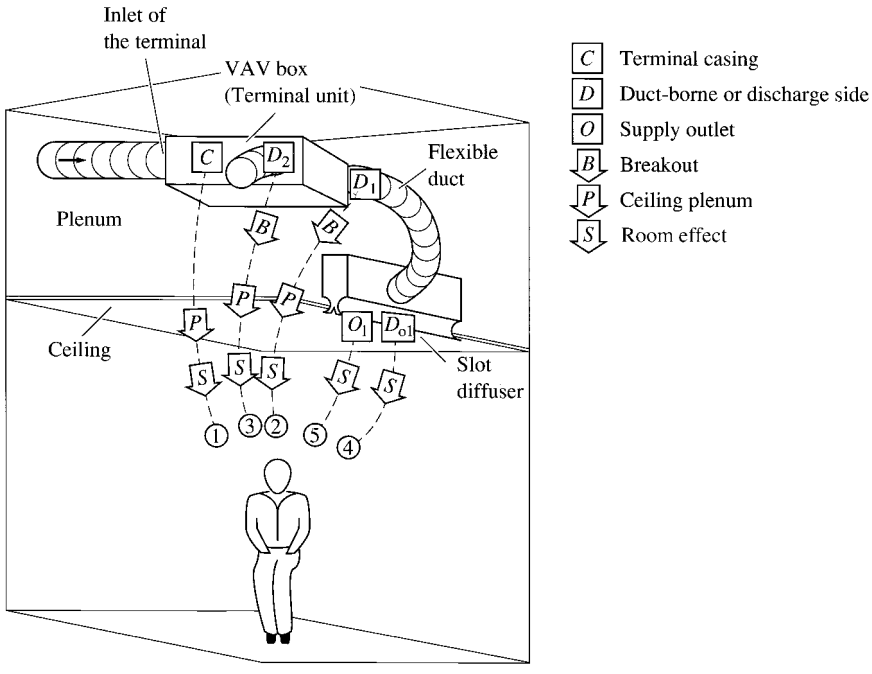


FIGURE 19.7 Sound path of a typical terminal unit installed in the ceiling plenum.

Environmental Adjustment Factor

According to Industry Standard (IS) 880, sound power levels of the terminal unit as listed in the manufacturers' catalogs are based on free field (outdoor) calibration of the reference sound source. At low frequencies, actual rooms are highly reverberant. The adjustment factor that takes into account the difference in sound power levels of a commonly used reference sound source measured in a free field and a reverberant field is called the *environmental adjustment factor* E_f, in dB. IS 885 recommends the following E_f values:

Band frequency	63	125	250	500	1000	2000	4000	8000
E_f, dB, re 10^{-12} W	+7	+3	+2	+1	+1	+1	+1	+1

Plenum Ceiling Effect

Sound attenuation of the plenum ceiling combination not only includes the transmission loss of the ceiling, but also considers the effect of the plenum. For a $\frac{5}{8}$-in.- (16-mm-) thick mineral fiber acoustic tile ceiling with a density of 35 lb/ft^3 (560 kg/m^3), IS 885 recommends the following values, which are the same as those in Table 19.14:

Octave band center frequency, Hz	125	250	500	1000	2000	4000
Plenum ceiling effect, dB	−5	−9	−10	−12	−14	−15

Example 19.1. A variable-air-volume box (terminal unit) receives conditioned air from an air-handling unit through main and branch ducts. The VAV box is installed inside a ceiling plenum and is connected to two slot diffusers by 10-in.- (250-mm-) diameter flexible ducts, each of which is 5 ft (1.5 m) long. Information is available from manufacturers' catalogs on the duct-borne fan noise at the inlet of the VAV box, the radiated sound power level $L_{w,\text{rad}}$, the discharge sound power level $L_{w,\text{dis}}$ of the VAV box, and the sound power level of the slot diffuser $L_{w,s}$. From measurements, the transmission loss of the breakout noise through flexible duct wall and the insertion loss of the flexible duct are known. These data are summarized as follows:

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
Fan noise at inlet, dB	65	63	61	52	46	45
$L_{w,\text{rad}}$, dB	60.5	55	50.5	46.5	41.5	37.5
$L_{w,\text{dis}}$, dB	63	60.5	47	47.5	54.5	52
$L_{w,s}$, dB	40	39	35	33	31	24
Flexible, TL_{out}, dB	− 8	− 11	− 14	− 17	− 20	− 25
Flexible, IL, dB/ft	0.50	0.85	1.48	2.2	2.04	1.64

If the volume of the room is 5040 ft^3 (143 m^3), estimate the total sound pressure level in the occupied zone with an average distance from the receiver of 8 ft (2.4 m). Ignore the fan noise transmitted through the return duct system.

Solution

1. From Eq. (19.11), the room effect at the octave band center frequency of 125 Hz of this occupied zone is

$$L_{pr} - L_{wr} = -5 \log V - 3 \log f - 10 \log r + 25$$
$$= -5 \log 5040 - 3 \log 125 - 10 \log 8 + 25 = -8.8 \text{ dB}$$

The room effect of other octave bands can be estimated similarly as follows:

Octave band center frequency, Hz	125	250	500	1000	2000	4000
Room effect, dB	− 9	− 9.5	− 11	− 11.5	− 12.5	− 13.5

2. For path 1, after considering the influence of the environmental adjustment factor E_f, one finds that the radiated sound power levels of the VAV box $L_{\text{rad},r} = L_{w,\text{rad}}\text{-}E_f$, in dB, are as follows:

Octave band center frequency, Hz	125	250	500	1000	2000	4000
Reverberant field, dB	57.5	53	49.5	45.5	40.5	36.5

Because the transmission loss of the casing of the VAV box is rather high, the breakout of duct-borne fan noise through the casing is far smaller than $L_{\text{rad},r}$, and is ignored.

The sound pressure level in the occupied zone due to the radiated sound power level from the VAV box can be calculated by subtracting the plenum ceiling effect from $L_{\text{rad},r}$ and then converting to the sound pressure level through the room effect.

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
$L_{\text{rad},r}$, dB	57.5	53	49.5	45.5	40.5	36.5
Ceiling plenum effect, dB	− 5	− 9	− 10	− 12	− 14	− 15
Room effect, dB	− 9	− 9.5	− 11	− 11.5	− 12.5	− 13.5
Total sound pressure level, dB	43.5	34.5	28.5	22	14	8

3. For path 2, the discharge sound power level of the VAV box changes from the free field $L_{w,\text{dis}}$ to the reverberant field $L_{\text{dis},r}$ and the duct-borne fan noise at the terminal unit $L_{w,f}$. Their combination is as follows:

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
$L_{\text{dis},r}$, dB	60	58.5	46	46.5	53.5	51
Fan noise $L_{w,f}$, dB	65	63	61	52	46	45
$L_{\text{dis},r} + L_{w,f}$, dB	66.5	64.5	61.5	53.5	54.5	52

The sound pressure level in the occupied zone due to the combination $L_{\text{dis},r} + L_{w,f}$ transmitted along sound path 2 can be estimated by subtracting the transmission loss of the flexible duct wall, the plenum ceiling effect, and the room effect from $L_{\text{dis},r} + L_{w,f}$.

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
$L_{\text{dis},r} + L_{w,f}$, dB	66.5	64.5	61.5	53.5	54.5	52
Flexible, TL_{out}, dB	− 8	− 11	− 14	− 17	− 20	− 25
Ceiling-plenum, dB	− 5	− 9	− 10	− 12	− 14	− 15
Room effect, dB	− 9	− 9.5	− 11	− 11.5	− 12.5	− 13.5
Sound pressure level, dB	44.5	35	26.5	13	8	< 0

4. For path 4, there are two flexible ducts and slot diffusers connected to the terminal unit; therefore, the branch power division is 3 dB. The sound pressure level in the occupied zone due to duct-borne noise transmitted along sound path 4 can be evaluated by subtracting the branch power division $L_{w,b}$, the insertion loss of the flexible duct IL_{flex}, and the room effect from the combination $L_{\text{dis},r} + L_{w,f}$.

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
$L_{\text{dis},r} + L_{w,f}$, dB	66.5	64.5	61.5	53.5	54.5	52
$L_{w,b}$, dB	− 3	− 3	− 3	− 3	− 3	− 3
IL_{flex}, dB	− 2.5	− 4	− 7	− 11	− 10	− 8
Room effect, dB	− 9	− 9.5	− 11	− 11.5	− 12.5	− 13.5
Sound pressure level, dB	52	48	40.5	28	29	27.5

5. For path 5, change the sound power level of the slot diffuser from free field $L_{w,s}$ to reverberant field $L_{s,r}$.

Then the sound pressure level in the occupied zone due to the slot diffuser can be estimated by subtracting the room effect from $L_{s,r}$.

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
$L_{s,r}$, dB	37	37	34	32	30	23
Room effect, dB	− 9	− 9.5	− 11	− 11.5	− 12.5	− 13.5
Sound pressure level, dB	28	27.5	23	20.5	17.5	9.5

6. The resultant estimated sound pressure level in the occupied space $L_{p,t}$, in dB re 20 μPa, is the sum of the contributions of all sound paths and can be calculated as follows:

	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
Path 1	43.5	34.5	28.5	22	14	8
Path 2	44.5	35	26.5	13	8	0
Path 3	44.5	35	26.5	13	8	0
Path 4	52	48	40.5	28	29	27.5
Path 5	52	48	40.5	28	29	27.5
Path 6	28	27.5	23	20.5	17.5	9.5
Path 7	28	27.5	23	20.5	17.5	9.5
$L_{p,t}$, dB	56	51.5	44	32.5	32.5	31

These resultant sound pressure levels at the center frequencies of various octave bands are all equal to or below the NC 40 curve.

19.9 ROOFTOP PACKAGED UNITS

Basics

Rooftop packaged units are widely used in low-rise commercial buildings. Because of the use of large rooftop packaged units, many rooftop packaged units are curb-mounted on lightweight roof deck construction. The distances between the fan and ceiling plenum, and between the fan and conditioned space, are sometimes shorter than the distance between the fan in the AHU and space. They may cause unique noise control problems. Objectionable noise from a rooftop packaged unit usually includes direct transmission of fan and compressor noise through the building roof and ceiling, structure-borne vibration, duct-borne fan noise, duct breakout noise, and duct rumble.

Locating the rooftop packaged unit at least 25 ft (7.6 m) horizontally away from areas with critical sound control requirements, such as conference rooms and executive offices whose NC curve should be 35 dB or less, is the most effective and economical method of noise control. A rooftop packaged unit can be located over buffer spaces, low-use storage areas, high-noise loading areas, and acoustically nonsensitive spaces such as rest rooms and stairwell buffer spaces.

Using a number of small packaged units instead of one or two large units divides the impact over a greater roof area and is sometimes beneficial. A rooftop packaged unit with a lower sound power level is often one size larger than the smallest unit for that capacity. The fan in such a unit is properly sized, with less restricted inlet and outlet conditions inside the unit.

Sound Sources and Paths

In a typical rooftop packaged unit, the paths by which fan and compressor noise is transmitted from the rooftop packaged units to the receiver in the occupied space are shown in Fig. 19.8*a*:

1. Radiated fan noise breaks out from the supply duct in the ceiling plenum → plenum ceiling effect → room effect → receiver.
2. Duct-borne fan noise on the discharge side → attenuation of ducts, elbows, and silencer → end reflection → room effect → receiver.
3. Radiated fan noise breaks out from the return air duct in the ceiling plenum → plenum ceiling effect → room effect → receiver.
4. Duct-borne fan noise on the return side → attenuation of ducts, elbows, and silencer → end reflection → room effect → receiver.

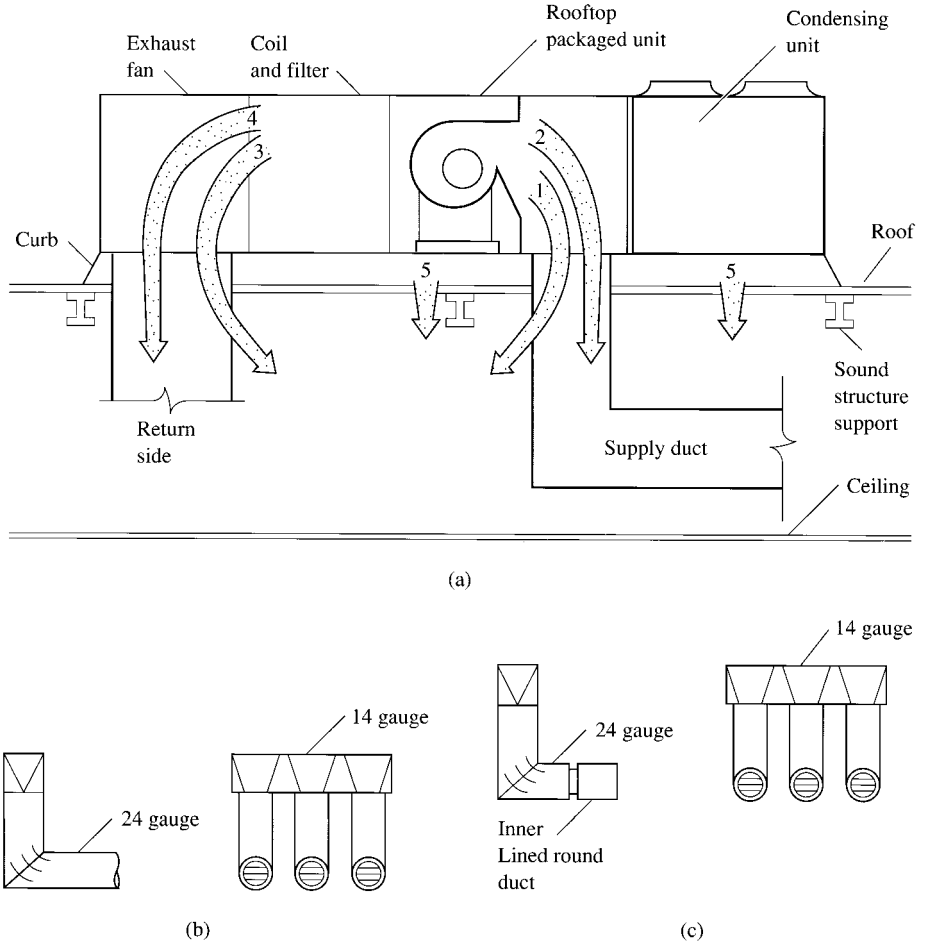


FIGURE 19.8 Sound path and sound control for a typical rooftop packaged unit: (*a*) installation of rooftop unit with rectangular supply duct; (*b*) rectangular to multiple drop: round mitered elbows with turning vanes; (*c*) rectangular to multiple drop: round mitered elbows with turning vanes, inner-lined round duct after mitered elbow.

5. Structure-borne fan or compressor noises → roof construction → plenum ceiling effect → receiver.

Discharge Side Duct Breakout

Experience has shown that the low-frequency noise breakout from the supply duct inside the ceiling plenum (path 1) and subsequent transmissions to the receiver through the ceiling are the critical noise problem in most rooftop packaged units. A supply fan that has a down shot outlet and that penetrates the roof directly produces many noise problems, such as very loud breakout sound inside the ceiling plenum. A supply fan with horizontal outlet discharge to an inner-lined plenum inside the rooftop packaged unit is better. Use 2-in. (50-mm) acoustic linnings for fan, compressor, and condenser sections to reduce sound transmission directly through the roof. An even better sound attenuation arrangement is to run the supply duct horizontally from the rooftop unit 10 to 20 ft (3 to 6 m), combined with silencers along the roof, and then penetrate into the building, directly above acoustically sensitive spaces. Duct exposed to outside atmosphere must be inner-lined, externally insulated, weatherproof enclosures that are covered with noise barriers.

Round ducts have a far greater TL_{out} than rectangular ducts. Based on laboratory tests, Beatty (1987) recommended using rectangular to multiple round duct drops with round mitered elbows and turning vanes, as shown in Fig. 19.8*b* and *c* to contain low-frequency rumble noise. Compared with rectangular ducts without turning vanes, these configurations have an additional noise reduction of 17.5 dB in octave band 1, 11.5 dB in octave band 2, and 13 dB in octave band 3 when the configuration is the same as in Fig. 19.8*b*. If the multiple drops of round duct have the configuration shown in Fig. 19.8*c*, there will be an additional noise reduction of 18 dB in octave band 1, 13 dB in octave band 2, and 16 dB in octave band 3 compared to rectangular ducts without turning vanes.

Use heavier-gauge sheet metal to fabricate the rectangular plenum before the round duct drops to reduce breakout noise.

If the first run of duct from the fan outlet must be rectangular and passes a noise sensitive area, Harold (1993) recommended the following:

- Use 16-gauge sheet metal for duct wall for the first 25-ft (7.5-m) run of duct.
- Use 1500 fpm (7.5 m/s) as the maximum duct velocity for the rectangular duct.
- Enclose the supply duct in a drywall enclosure of two layers of $^5/_8$-in. (16-mm) gypsum board with freestanding walls that have a 4-in. (100-mm) airspace around the duct and contain a 3-in.-(75-mm-) thick fiberglass blanket that touches only the duct or drywall.
- Employ an active silencer.

Careful design of the fan outlet configurations will prevent sharp turning of airstreams. Inner lining of the ductwork and sometimes an additional silencer are necessary at the downstream side for more effective noise control.

The duct system should not be designed for an excessive velocity, i.e., normally not greater than 3000 fpm (15 m/s). Design the duct fittings at minimum dynamic losses.

VAV boxes and reheating VAV boxes in a VAV reheat system should not be operated at a completely shut-off condition because a high fan total pressure at a low volume flow rate is generally a noisy operating point for centrifugal fans.

Sound Path on Return Side

On the return side, there are three types of return air systems:

- A rooftop unit whose return air system connects directly the return ceiling plenum to the return intake of the rooftop unit through a short return duct system. This is the type of return air system

widely used for rooftop units. A rectangular return duct may be used if its duct wall is made of 16-gauge (1.6-mm thickness) galvanized sheet metal and breakout noise is not critical. Adopt a duct velocity of 1500 fpm (7.5 m/s) or less. Use 2-in. (50-mm) thick inner lining in the return side (including ductwork) of the rooftop unit. Use tees with a minimum leg length of 3 to 5 equivalent diameters to increase end reflection loss.

- For ducted return air systems, the sound power level at the supply or return fan intake should be attenuated by the return duct liners, return side silencer, and room effect. For a rooftop unit with a collective return grille, a sound-attenuating plenum, as shown in Fig. 19.3, is often cost-effective if space is available.
- For an open fan room (Fig. 16.7), return air is often transferred to the open fan room from the return ceiling plenum through a transfer duct or a transfer wall passage. Both require an inner lining 2 in. (50 mm) thick with a perforated metal sheet facing. Air velocity in the return transfer duct or passage should be 1000 fpm (5 m/s) or less.

Structure-Borne Noise

Structure-borne noise can be controlled by using suitable vibration isolators. Special curb-mounting bases should be used to provide better sound and vibration control. Sound structural support should be provided for rooftop packaged units. Borzym (1991) says that the best place for the rooftop packaged unit is directly over a column or straddling a major beam close to a column. Good locations are those near the intersection of beam with a heavy end close to a column, straddling a beam, or close to a beam. The rooftop packaged unit should be mounted on a spring-isolated roof curb and should float on the springs. The roof below the rooftop packaged unit should not vibrate.

The roof directly under the rooftop packaged unit within the roof curb should contain a good acoustic barrier to prevent sound transmission through the bottom panel of the packaged unit, the roof, and the ceiling plenum. A typical sound barrier is made of semirigid fiberglass insulation, at least 4 in. (100 mm) thick, covered by two layers of $^5/_8$-in. (16-mm) gypsum board. The staggered gypsum boards are used to seal the joints and seams airtight.

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CHAPTER 20
AIR SYSTEMS: BASICS AND CONSTANT-VOLUME SYSTEMS

20.1 AIR SYSTEM BASICS 20.2
- Air Systems 20.2
- Air Distribution Systems 20.3
- Ventilation Systems 20.3
- Mechanical Ventilation Systems 20.3
- Regenerative Systems 20.3
- Smoke Control Systems 20.4
- Terminals 20.4
- Primary, Secondary, and Transfer Air 20.4

20.2 BUILDING LEAKAGE AREA AND BUILDING TIGHTNESS 20.5
- Effective Leakage Area 20.5
- Building Air Leakage 20.5
- Air Change Per Hour at 50 Pa (ACH50) 20.6

20.3 SPACE PRESSURIZATION 20.7
- Space Pressure Characteristics 20.7
- Stack Effect 20.7
- Stack Effect for High-Rise Buildings 20.9
- Wind Effect 20.10
- Air Systems and Mechanical Ventilation Systems 20.11
- Airflow Balance and Space Pressurization by Differential Flow 20.11

20.4 INFILTRATION AND EXFILTRATION 20.14
- Volume Flow Rate of Infiltration 20.14

20.5 FAN-DUCT SYSTEMS 20.14
- System Operating Point 20.15
- Buckingham π Method and Fan Laws 20.15

20.6 SYSTEM EFFECT 20.17
- Mechanism of System Effect 20.17
- Inlet System Effect 20.18
- Inlet System Effect Loss 20.19
- Inlet System Effect Loss Coefficient 20.19
- Outlet System Effect 20.20
- Outlet System Effect Loss Coefficient 20.22
- Selecting Fan Considering System Effect Losses 20.23

20.7 COMBINATION OF FAN-DUCT SYSTEMS 20.24
- Two Fan-Duct Systems Connected in Series 20.25
- Fans Combined in Parallel and Connected in Series with a Duct System 20.26
- Two Parallel Fan-Duct Systems Connected with Another Duct System 20.28

20.8 MODULATION OF THE FAN-DUCT SYSTEM 20.31
- Modulation Curve 20.31
- Modulation of Fan-Duct Systems 20.33

20.9 CLASSIFICATION OF AIR SYSTEMS 20.39

20.10 CONSTANT-VOLUME SYSTEMS 20.40
- System Characteristics 20.40
- Energy per Unit Volume Flow 20.41

20.11 AIR CONDITIONING PROCESSES 20.41
- Sensible Heat Ratio 20.41

20.12 SPACE CONDITIONING AND SENSIBLE COOLING AND HEATING PROCESSES 20.43
- Space Conditioning Process 20.43
- Sensible Heating and Cooling Processes 20.44

20.13 HUMIDIFYING PROCESSES 20.45
- Steam Injection and Heating Element Humidifier 20.45
- Air Washer 20.46
- Oversaturation 20.46

20.14 COOLING AND DEHUMIDIFYING PROCESS 20.46

20.15 ADIABATIC MIXING AND BYPASS MIXING PROCESSES 20.50
- Two-Stream Adiabatic Mixing Process 20.50
- Bypass Mixing Process 20.52

20.16 SINGLE-ZONE, CONSTANT-VOLUME SYSTEMS—COOLING MODE OPERATION 20.53
- Air Conditioning Cycle 20.53
- Cooling Mode Operation in Summer 20.53
- Cooling Mode Operation in Winter without Space Humidity Control 20.56
- Cooling Mode Operation in Winter with Space Humidity Control 20.57
- Part-Load Operation and Controls 20.58
- Outdoor Ventilation Air and Exhaust Fans 20.58

20.17 SUPPLY VOLUME FLOW RATE 20.59
- Based on Space Cooling and Heating Load 20.59
- Based on Requirements Other Than Cooling Load 20.60
- Rated Volume Flow of Supply and Return Fans 20.61

20.18 DETERMINATION OF THE SUPPLY AIR CONDITION 20.62
Air Conditioning Rules 20.63
Graphical Method 20.63
Influence of Sensible Heat Ratio 20.64
20.19 CONSTANT-VOLUME SINGLE-ZONE SYSTEMS—HEATING MODE OPERATION 20.69
Heating Mode without Space Humidity Control 20.69
Heating Mode with Space Humidity Control 20.71
Part-Load Operation 20.73
Dual-Thermostat Year-Round Zone Temperature Control 20.73
20.20 CONSTANT-VOLUME MULTIZONE SYSTEM WITH REHEAT 20.74
Reheating, Recooling, and Mixing 20.74
Constant-Volume Multizone System with Reheat 20.75
Control Systems 20.75
Operating Parameters and Calculations 20.77
System Characteristics 20.78
REFERENCES 20.79

20.1 AIR SYSTEM BASICS

Air Systems

Air systems and their controls directly affect the indoor environment and indoor air quality (IAQ). In a broad sense, air systems are a group of subsystems in an air conditioning system. Air systems inclued supply and return air systems (space recirculating systems), mechanical ventilation systems, air distribution systems, regenerative systems, smoke control systems, and terminals. The main functions of air systems are as follows:

- Conditioning the supply air including heating or cooling, humidification or dehumidification, cleaning and purifying, and attenuation of objectional noise produced by fans, compressors, and pumps
- Distributing the conditioned supply air with adequate outdoor air to the conditioned space, extracting space air for recirculating, and exhausting or relieving unwanted space air to the outdoors
- Providing space pressurization, toxic gas exhaust, and smoke control for occupants' safety and fire protection
- Controlling and maintaining required space temperature, humidity, cleanliness, air movement, sound level, and pressure differential within predetermined limits at optimum energy consumption and cost
- Using high-temperature airstream to reactivate the desiccant, if any

In actual practice or in a narrower sense, an air system or an air-handling system often denotes a more specific system that conditions, transports, distributes the supply air, recirculates the space air, and controls the indoor environment according to requirements. Such an air system often consists of fans, heat exchangers (including coils, evaporative coolers, and dehumidifiers), a direct-fired gas heater, filters, a mixing box, dampers, ductwork, terminals, space diffusion devices, and controls.

In a central hydronic air conditioning system, an air system consists of an air-handling unit (air is cooled and dehumidified by a water cooling coil), ducts, terminals, space diffusion devices, and controls. In a unitary packaged air conditioning system, the air system consists of an air handler (in which air is cooled and dehumidified by a DX coil or rotary dehumidifier), ducts, terminals, space diffusion devices, and controls. In a space air conditioning system, its air system is a combination of an outdoor air ventilation system and a space air recirculating system, which consists of a fan, a coil, a filter, ductwork, and controls. A separate outdoor ventilation air system provides conditioned

outdoor air for occupants. A space recirculating system conditions the space recirculating air to offset the space cooling or heating load. In an individual air conditioning system, the air system, heating system, and refrigeration systems are integrated in a single packaged air conditioner. It is not necessary to distinguish them separately. The air system in a thermal storage, desiccant cooling, or an evaporative-cooling air conditioning system is similar to that of a central hydronic system.

The air system is the foundation in an air conditioning system. It directly contacts the occupants, manufacturing products, and conditioned space. Air systems play a key role in maintaining a healthy, safe, and comfortable indoor environment with an acceptable indoor air quality at optimum energy use.

It is more specific, convenient, and clear that the systems which condition, transport, distribute, and control the supply air and recirculate the space air are called air systems; and all the other air systems in a broad sense that do not cool and dehumidify the air are called mechanical ventilation systems, exhaust systems, and smoke control systems.

Air Distribution Systems

An air distribution system is a subsystem of an air system which consists of ductwork, terminals, space diffusion devices, and controls to transport and distribute the conditioned air to the occupied zone as well as to control the indoor environment within required limits.

Ventilation Systems

As discussed in Sec. 4.10, ventilation air comprises outdoor air plus treated recirculated air with acceptable indoor air quality. Ventilation air is also the air used for the dilution of the contaminant concentrations in the conditioned space. Ventilation air may or may not be conditioned. Ventilation indicates the process to supply or to remove ventilation air. Because of the importance of the ventilation air supply, a ventilation system is usually considered as part of air systems that use fans to supply and to remove ventilation air to and from any space. Natural ventilation (including infiltration) is a process that provides outdoor ventilation air by stack or wind effect, or both.

Mechanical Ventilation Systems

A mechanical ventilation system uses mechanical means (fans) to transport the air without cooling and dehumidifying it. The following are the types of mechanical ventilation systems currently used:

- Mechanical ventilation systems for cooling and contaminant dilution, which supply outdoor air to a room to cool equipment or to dilute air contamination to acceptable level, as in a transformer room or a garage
- Exhaust systems, which discharge indoor air, or exhaust toxic gas and contaminated air from localized sources such as exhaust systems from bathrooms or from chemical hoods in laboratories
- Makeup air systems as a part of mechanical ventilation systems in which outdoor air is often heated and filtered to compensate for the exhausted air

Regenerative Systems

In a regenerative or drying system, outdoor air is heated to a higher temperature such as 150 to 200°F (65 to 93°C) and is then used to remove the absorbed moisture from the desiccant by drying. After drying, the desiccant is reactivated and is ready to dehumidify the process air again.

Smoke Control Systems

Smoke control systems include stairwell pressurizing systems and zone smoke control systems. Stairwell pressurizing systems pressurize stairwells as an escape route in high-rise buildings during building fire. Zone smoke control systems exhaust the smoke in the fired floor and pressurize the floors immediately above and below the fired floor to prevent smoke contamination. Smoke control systems are discussed in Chap. 22.

Terminals

A terminal is a factory-made device installed inside a control zone either in a ceiling plenum above the conditioned space or sometimes directly mounted adjacent to the external wall of the conditioned space or in a floor plenum. A terminal usually has an individual control system to maintain a selected indoor enviromental parameter within predetermined limits for a control zone.

Conditioned air including outdoor air is supplied to a terminal from the supply main duct through branch takeoffs. It may be mixed with recirculating air and is supplied to the control zone served by the terminal in a multizone air system.

A terminal may perform any of the following functions or combinations thereof:

- Heating/cooling or filtration
- Mixing of outdoor ventilation air or conditioned air with recirculating air, or mixing of cold supply air with warm supply air
- Controlling and maintaining a predetermined zone temperature by modulating the volume flow rate of supply air, the temperature of supply air, or the ratio of the volume flow of cold and warm air supplies. VAV boxes, reheating VAV boxes, mixing VAV boxes, fan-powered VAV boxes, and fan-coil units are all terminals, and will be discussed in Chap. 21.

Primary, Secondary, and Transfer Air

Primary air is conditioned makeup outdoor air or a conditioned mixture of outdoor and recirculating air. Primary air is either conditioned and supplied from a separate makeup AHU (PU) or from an AHU (PU) using a mixture of outdoor and recirculating air to the terminals. Sometimes the primary air is supplied directly to the space.

Primary air is usually mixed with the recirculating air, which is often called secondary air. The mixing most often takes place in the terminals, or another AHU or PU, or sometimes directly in the conditioned space. The mixture of primary air and recirculating air (secondary air) may be conditioned again in the terminals. It is then supplied to the conditioned space to offset the space load. The ratio of the volume flow rate of primary air to the mixture (zone supply air) varies from 0.15 to 0.5.

Recirculating air is the air recirculated from the conditioned space, through an AHU, a PU, or a terminal, and supplied again to the conditioned space. Recirculating air is different from the return air which returns from the conditioned space to an AHU, or PU, and exhausts outdoors. The purpose of mixing primary air with recirculating air in the terminal is to save energy including fan energy, as well as to save investment by using smaller supply and return ducts, which occupy less space.

Transfer air is indoor air that moves or supplies to a conditioned space from an adjacent area. Transfer air is entirely composed of recirculating air (secondary air). Transfer air often has less unused outdoor air than a mixture of outdoor and recirculating air. Unused outdoor air is outdoor air whose capability of dilution of air contaminants has not been expended. Recirculating air still may contain a certain amount of unused outdoor air. Restrooms are normally supplied by transfer air.

20.2 BUILDING LEAKAGE AREA AND BUILDING TIGHTNESS

Currently, the building tightness or the building air leakage can be indicated by the following indices: (1) an effective leakage area, (2) air leakage per unit area of building shell, and (3) air change per hour (ACH) at a specific indoor-outdoor pressure differential.

Effective Leakage Area

The effective leakage area of a building $A_{e,l}$, in ft^2 (m^2), is the amount of open wall area that allows the same volume flow rate of air flowing through it as flowing through the actual building when the pressure difference across the open wall area and the shell of the actual building is the same. The $A_{e,l}$ value is usually determined by a fan pressurization method, in which a blower installed at the door of a building is used to maintain a pressure difference across the building shell. The airflow rate $\dot{V}$, in cfm (m^3), can then be measured , and the $A_{e,l}$ value is calculated as

$$\dot{V} = C_d A_{e,l} \sqrt{\frac{2\,\Delta p}{\rho}} = C_d A_{e,l} 4005 \sqrt{\Delta p} \tag{20.1}$$

where C_d = discharge coefficient; its value is 1 for effective leakage area
ρ = air density, lb/ft^3 (kg/m^3)
Δp = pressure differential between outdoor and indoor air at same level, in. WC (Pa)

According to *ASHRAE Handbook 1997*, *Fundamentals*, effective air leakage areas for representative residential building components at an outdoor-indoor pressure difference of 0.016 in. WC (4 Pa) are as follows:

	Best estimate, in.^{2}/ft^2
External walls	
Cast-in-place concrete	0.007
Clay brick cavity wall, finished	0.0098
Lightweight concrete block, painted	0.016
Window	
Wood frame	
Uncaulked	0.025
Caulked	0.004
Double-hung, not weatherstripped	0.12
Double-hung, weatherstripped	0.031
Door, average	0.015 in.^{2}/lftc (linear ft. crack)
Ceiling	0.026

Building Air Leakage

When air flows through a crack or gap, the flow never becomes fully developed. The relationship between overall air leakage rate $\dot{V}_{\text{leak}}$, in cfm (L/s), and the indoor-outdoor pressure difference Δp, in in. WC (Pa), can often be represented by a widely used air leakage equation as

$$\dot{V}_{\text{leak}} = C_{\text{flow}} A_w (\Delta p)^n \tag{20.2}$$

where C_{flow} = flow coefficient, cfm/ft^2 (m^3/h · m^2) at Δp = in. WC (Pa)
A_w = area of exterior wall or roof, ft^2 (m^2)
Δp = pressure difference across building envelope, in. WC (Pa)

Because the cracks have large flow resistance, in Eq. (20.2), the exponent n may have a value closer to 1 than openings with less flow resistance. Shaw et al. (1993) recommended $n = 0.65$ based on field-measured results. Persily (1999) found significant levels of air leakage in commercial and institutional buildings. Among the data measured from 139 buildings around the world, flow coefficients C_{flow}, in cfm/ft^2 (m^3/h · m^2), varies from 0.15 to 6.85 cfm/ft^2 (2.7 to 124.5 m^3/h · m^2) at an indoor-outdoor pressure difference Δp of 0.3 in. WC (75 Pa) using the fan pressurization test method with a mean $C_{flow} \approx 1.5$ cfm/ft^2 (27.1 m^3/h · m^2) of exterior wall area at $\Delta p = 0.3$ in. WC (75 Pa). There is no correlation between air leakage and the age of the building tested. Building height does show an impact on airtightness with high-rise buildings being tighter, especially one-story buildings are leakier.

Based on Persily (1999) who analyzed building envelope airtightness data measured from 139 commercial and institutional buildings, flow coefficients C_{flow}, in cfm/ft^2 (m^3/h · m^2), per unit exterior wall area at 0.3 in. WC (75 Pa) for tight, average, and leaky buildings can be estimated as follows:

	C_{flow}, cfm/ft^2 (m^3/h · m^2) of exterior wall area at 0.3 in. WC (75 Pa)
Tight building	0.30
Average	1.00
Leaky building	2.00

Air Change per Hour at 50 Pa (ACH50)

The air leakage rate of a building can also be expressed by the air changes per hour at a specific oudoor-indoor pressure difference of which 0.2 in. (50 Pa) is most widely used. As in the effective leakage area, a fan is installed at the door of a building and is used to maintain a specific pressure difference across the building shell. The air leakage $\dot{V}_{leak}$, in cfm (L/min), supplied by the fan is squeezed outdoors through the cracks and gaps on the building shell and is then measured. The air change per hour of air leakage, denoted by ACH, can be calculated as

$$\text{ACH} = \frac{60\ \dot{V}_{leak}}{V_r} \tag{20.3}$$

where V_r = the space volume of room or enclosure, ft^3 (m^3). For residential buildings, in *ASHRAE Handbook 1997*, *Fundamentals*, ACH50 $\leq$ 3 is considered exceptionally tight, and for residential buildings 5 $<$ ACH50 $\leq$ 20 is considered conventional. There are 10 percent of the 204 low-income houses whose ACH50 $>$ 47.5.

Withers et al. (1996) measured the ACH50 values for 18 small commercial buildings after repair of duct leaks, the sealing of the building shell, and other repairs, four of them with ACH50 $\leq$ 5.2, seven of them with 10.7 $\leq$ ACH50 $\leq$ 19.7, five of them with 21.2 $\leq$ ACH50 $\leq$ 28.8, one with ACH50 = 46, and the highest one with ACH50 = 52.8, a court office.

Because of the desire for energy conservation, the airtightness of buildings has been improved during the past decades. Since the ratio of the exterior wall area to the floor area is quite different for different types of buildings, it is difficult to compare the airtighness of different types of buildings, especially using different air leakage indices.

20.3 SPACE PRESSURIZATION

Space Pressure Characteristics

Space pressurization or building pressurization is a process that maintains a static pressure difference between space air and the air in the surrounding area, either outdoor air at atmospheric pressure or air in adjacent rooms, for the purpose of providing a required indoor environmental control.

As mentioned in Chap. 4, normally, slightly positive space pressure should be maintained if the required level of indoor air quality and environmental control of the conditioned space is higher than that of the surrounding area. In laboratories with toxic gas exhaust systems, negative space pressure should be maintained. As covered in Chap. 4, for comfort air conditioning systems for low-rise buildings, the pressure differential between space air p_r and outdoor air p_o is usually between + 0.005 and + 0.05 in. WC (+ 1.25 and + 12.5 Pa), and + 0.01 to + 0.03 in. WC (5 to 7.5 Pa) is often adopted. For high-rise buildings, the pressure differences between p_r and p_o are far higher because of the stack effect on various floors. For clean rooms, the minimum pressure differential between a clean room and any adjacent area of less strict cleanliness requirements should be 0.05 in. WC (12.5 Pa) with all entryways closed. A space positve pressure of 0.05 in. WC (12.5 Pa) exerts a force of 4 lb_f (18 N) on a door with an area of 16 ft^2 (1.5 m^2). An excessive pressure differential may make opening and closing doors difficult.

A positive pressure means that $p_r > p_o$ or $p_r > p_{sur}$. Here p_{sur} denotes static pressure in the surrounding area, in in. WG (Pa). When a positive pressure is maintained in a conditioned space, in addition to the air returned and exhausted from the space, a certain amount of air is squeezed out or exfiltrated from the space through openings and cracks in the building shell. Unconditioned outdoor air or contaminated air from the surrounding area is prevented from infiltrating the conditioned space because of this positive space pressure. A negative pressure means that $p_r < p_o$ or $p_r < p_{sur}$, which induces infiltration.

Except in air jets, the velocity pressure in the conditioned space is small and is usually ignored in space pressurization analysis.

The space pressure difference between the conditioned space and the surrounding area is affected by the stack effect, wind effect, and operation of the air systems and exhaust systems as well as the airtightness (amount of leakage area) of the building shell.

Stack Effect

The temperature difference between the cold outdoor and warm indoor air columns causes a density difference between these air columns that, in turn, creates a pressure difference between the cold and the warm air columns. This phenomenon is called the stack effect.

In any building under the stack effect alone, there is a level at which the indoor and outdoor pressures are equal. This level is called the neutral pressure level (NPL), as shown in Fig. 20.1*b*. During the heating season, outdoor cold air pressure is greater than indoor warm air pressure below the NPL. This pressure difference causes the outdoor cold air to enter the building through the lower inlets, and the indoor warm air to flow upward and discharge from the top openings. During the cooling season, the reverse effect occurs, but the stack effect is smaller because of the smaller temperature difference between outdoor and indoor air.

The pressure difference due to the stack effect between outdoor and indoor air Δp_{st}, in in. WC (Pa, 1 in. WC = 248.6 Pa) at height H (in ft or m), from a reference level, is given as

$$\Delta p_{st} = p_{so} - p_{si} = 0.19256(\rho_o - \rho_i)(H - H_{NPL})$$
$$= 0.19256\,\rho_i \frac{(H - H_{NPL})(T_{Ri} - T_{Ro})}{T_{Ro}} \tag{20.4}$$

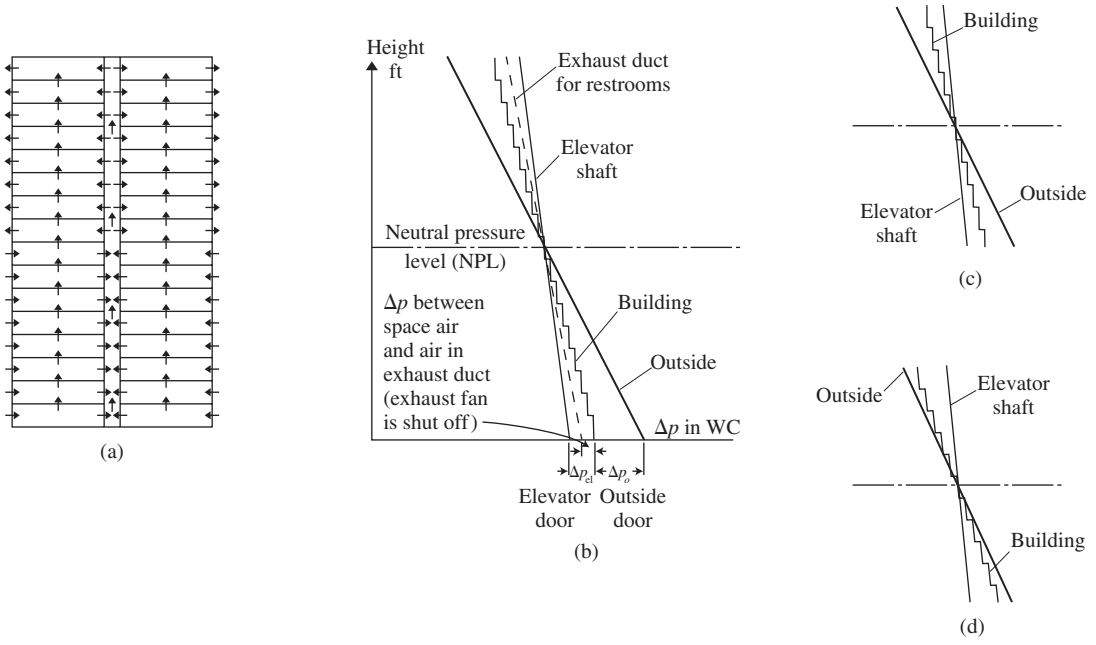


FIGURE 20.1 Stack effect in a high-rise building (*a*) schematic diagram of building airflow; (*b*) pressure profiles; (*c*) pressure profiles for tight building shell; (*d*) pressure profiles for leaky building shell.

where p_{so}, p_{si} = outdoor and indoor absolute static pressure, in. WG (Pa)
ρ_o, ρ_i = outdoor and indoor air density, lb/ft^3 (kg/m^3)
H_{NPL} = vertical distance between NPL and reference level, ft (m)
T_{Ro},T_{Ri} = outdoor and indoor absolute temperature, °R

For a high-rise building with two openings on the external wall, N_{NPL}, in ft (m), measured from lower opening can be calculated as

$$H_{NPL} = \frac{H_o}{1 + (A_1/A_2)^2(T_{Ri}/T_{Ro})} \tag{20.5}$$

where H_o = vertical distance between two openings, ft (m)
A_1, A_2 = area of lower and higher openings, ft^2 (m^2)
T_{Ri}, T_{Ro} = absolute temperature of indoor and outdoor air, °R (K)

In Eq. (20.5), $T_{Ri} > T_{Ro}$. If the indoor air is cooler, that is, $T_{Ro} > T_{Ri}$, then T_{Ri}/T_{Ro} should be changed to T_{Ro}/T_{Ri}. For conditioned spaces where $T_{Ri} > T_{Ro}$, the chimney and exhaust systems raise the NPL level and the cold outdoor air supply system lowers the NPL level.

In Fig. 20.1*b*, it can be seen that the absolute static pressure of outdoor air p_{so} is greater at the low level than at the high level. This is because the air column is taller at the lower level.

In an ideal single-cell building with no inside partitions, with a vertical distance of 100 ft (30 m) between the lower air inlet and upper air outlets and an indoor and outdoor air temperature difference of 50°F (10°C), the static pressure difference Δp_{st} is about 0.15 in. WC (38 Pa).

Most buildings have inside partitions; i.e., they are multicell buildings. They have doors at the lower entering inlet, windows at the upper discharge outlets, and doors to the stairwells. Usually

doors and windows are closed. When outdoor cold air flows through the cracks and gaps around these doors, windows, and partitions, its velocity and flow rate are significantly reduced because of these dynamic losses.

Stack effect depends on the magnitude of outdoor-indoor temperature difference and therefore varies from maximum to zero as the outdoor climate changes.

Stack Effect for High-Rise Buildings

Lovatt and Wilson (1994) measured the pressure differences of the various elements of the building shell and internal partitions of a 44-story high-rise office building of 577 ft (176 m) above grade located in Montreal, Quebec, Canada. The air leakage rate for this building was found by Tamura and Wilson (1967) to be 0.55 cfm/ft^2 (2.8 L/s · m^2). The stairwells in this building are not directly connected to a stairwell pressurization system. The results were as follows:

Condition	Normal occupied
Outdoor temperature	20.3°F (− 6.5°C)
Lobby temperature	73.0°F (22.8°C)
Pressure difference Δp, in. WC (Pa)	
Outside–lobby vestibule	0.29 (72)
Lobby vestibule–lobby	0.02 (5)
Low-rise elevators–lobby	− 0.10 (− 24)
Mid-rise elevators–lobby	− 0.12 (− 30)
High-rise elevators–lobby	− 0.16 (− 40)
M2 stairwell–lobby	− 0.008 (− 2)
Roof elevator–41st floor	− 0.23 (− 56)
Roof elevator–roof vestibule	− 0.32 (80)
Observation deck–roof vestibule	− 0.07 (− 18)
Roof–stairwell	− 0.78 (− 195)
43d Floor–stairwell	− 0.19 (− 47)
42d Floor vestibule–stairwell	− 0.07 (− 17)
Exhaust fan room–42d floor vestibule	0.06 (16)
42d Floor–42d floor vestibule	− 0.17(− 43)
High-rise elevator mech. room–43d floor	0.21 (53)
Mid-rise elevator mech. room–32d floor	0.02 (5)
Low-rise elevator mech. room–16th floor	0 (0)

The stack effect for high-rise buildings in cold climates may cause the following problems:

- Pressure differences across elevator doors $\Delta p_{\text{elev},d}$ generally do not exceed the 0.1-in. WG (25-Pa) limit. This is the limit recommended by the manufacturer for proper door operation. Sticky elevator doors may occur when $\Delta p_{\text{elev},d} > 0.1$ in. WC (25 Pa).
- Pressure difference across the stairwell doors $\Delta p_{\text{sta},w}$ is generally less than 0.04 in. WC (10 Pa). These stairwell doors to occupied floors are easy to open and provide an escape route to outdoors during a building fire.
- During cold weather, the pressure difference across the roof door Δp_{roof} may well be above the NFPA limit (1988) between 0.21 and 0.45 in. WC (52.5 and 112.5 Pa).
- The lobby entrance doors show greater pressure difference across them.
- The stack effect may affect the airflow distribution of a vertical exhaust duct from restrooms.

For a tight high-rise building with less air leakage area on the building shell, the pressure profile of the high-rise building is shown in Fig. 20.1*c*, where the pressure difference across the outside (entrance) door is greater than the pressure difference across the elevator doors. For a leaky high-rise

building, the pressure profile is shown in Fig. 20.1*d*; the pressure difference across the outside door is smaller than the pressure difference across the elevator doors.

Lovatt and Wilson (1994) recommended that if the pressure difference of the elevator doors of a high-rise building during cold climate $\Delta p_{\text{elev},d} > 0.1$ in. WC (25 Pa), then repair of the air leakage area on the building shell is often the first remedy. Reduction of air leakage will reduce the infiltration and the heat energy use as well.

Add airtight vestibules around all entrance doors, and around doors serving elevators and stairwells. These setups divide the pressure difference of the entrance door, elevator doors, and stairwell doors into two parts, and again they reduce infiltration. Add airtight separations in vertical shafts whenever possible, to reduce the pressure difference across the roof door in the stairwell.

To pressurize or depressurize the entrance lobby or any other floor by a mechanical ventilation system is often not an effective and economical means to reduce the pressure difference across doors or other building elements, because one of the pressure differences is reduced at the expense of increasing the other one.

Wind Effect

As wind flows over a building, it creates positive pressure on the windward side of the building and negative pressure on the leeward side, as shown in Fig. 20.2. Wind pressure on the building surface relative to the approaching wind speed p_w, in. WC (Pa), can be calculated as

$$p_w = C_p P_{vw} = C_p \left(\frac{v_{wa}}{4005} \right)^2 \tag{20.6}$$

where C_p = surface pressure coefficient
p_{vw} = velocity pressure at approaching wind speed, in. WC (Pa)
v_{wa} = approaching wind speed at wall height on windward side, fpm (m/s)

If the wind direction is normal to the windward surface of a high-rise building with a width-depth ratio of 4, the average C_p value is about 0.60 on the windward side, about -0.5 on the leeward side and on the flat roof, and about -0.25 on the other two sides.

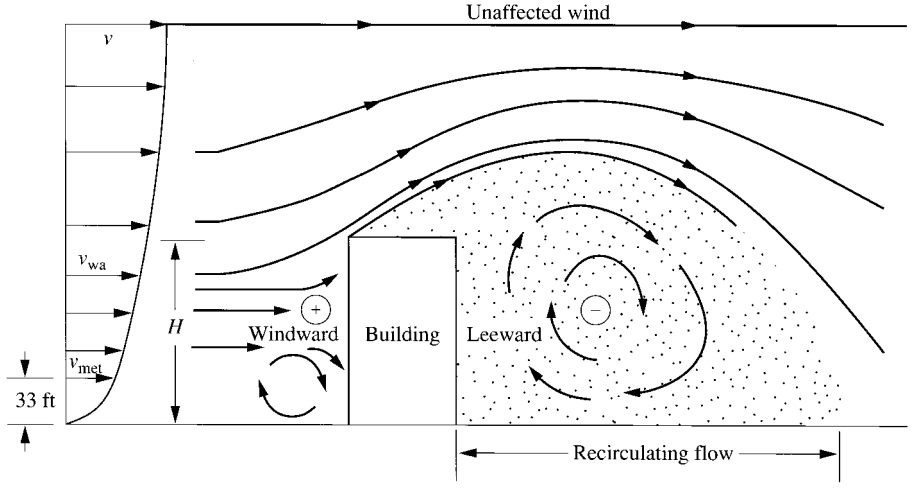


FIGURE 20.2 Airflow around buildings.

Wind speed from a meteorological station v_{met}, in fpm (m/s), is usually measured at a height of 33 ft (10 m). The approaching wind speed at wall height v_{wa} should be corrected for height and terrain roughness as follows:

$$v_{wa} = A_o v_{met}\left(\frac{H_w}{H_{met}}\right)^n \tag{20.7}$$

where H_w = height of wall where wind pressure is exerted, ft (m)
H_{met} = height at which wind speed is measured at meteorological station, ft (m)

In Eq. (20.7), A_o is a correction factor for terrain roughness, and n is the velocity profile exponent. For suburban areas, $A_o = 0.6$ and $n = 0.28$; for urban areas, $A_o = 0.35$ and $n = 0.4$; for airports, $A_o = 1.0$ and $n = 0.15$.

The mean wind speeds measured at meteorological station v_{met} in most U.S. cities in winter are below 1200 fpm (6 m/s). Wind pressure on a wall at a height of 33 ft (10 m) in suburban areas with $v_{met} = 1200$ fpm (6 m/s) is about 0.054 in. WC (13 Pa).

The influence of the wind effect on space and building pressure characteristics is as follows:

1. Rooms on the windward side of the building are usually at a positive pressure and on the leeward side at a negative pressure relative to the corridor pressure. It is best to build clean spaces, such as conference rooms, on the windward side of the prevailing wind, and laboratories with toxic gas exhaust systems on the leeward side of the building.
2. Outdoor air intake should be located on the side with a positive surface pressure coefficient C_p in the prevailing wind. Exhaust outlets should be located where C_p is negative, preferably on the rooftop.
3. Sufficient total pressure must be provided by the supply fan to overcome the negative pressure at the outdoor intake and by the exhaust fan to overcome the positive pressure at the exhaust outlet. Alternatives should be provided to allow outdoor air intake and outlet when the wind direction is different from that of the prevailing wind.

Wind speed may vary from zero to its maximum speed. Wind velocity is a vector quantity.

Air Systems and Mechanical Ventilation Systems

If a space is served by air systems including supply, return, and exhaust systems, the space pressure characteristics are closely related to the characteristics of air systems during their operating time. When a hurricane comes, the windward pressure may be as high as several inches WC (several hundred Pa), and the leeward negative pressure has a similar negative pressure. Hurricanes and storms last a comparatively short time. Hurricanes affect entrance doors, windows, as well as the performance of outdoor intake and exhaust outlets in the air systems and HVAC&R equipment outdoors that are not well shielded and protected.

For high-rise buildings in cold winter, the stack effect has a significant influence on the indoor-outdoor pressure difference on various floors. For low-rise buildings, air systems (including mechanical ventilation systems) have a dominant influence on space pressure characteristics except on stormy days.

Airflow Balance and Space Pressurization by Differential Flow

Consider a single-cell room model of an enclosed space surrounded by an area of uniform pressure. Based on the principle of continuity of mass, for an incompressible steady flow process, the total mass flow rate of air entering an enclosed space must equal the total mass flow rate of air leaving

the space (see Fig. 20.3), or

$$\dot{m}_s + \dot{m}_{\text{inf}} = \dot{m}_{\text{rt}} + \dot{m}_{\text{ex}} \tag{20.8}$$

where $\dot{m}_s$ = mass flow rate of supply air, lb/min (kg/s)
$\dot{m}_{\text{rt}}$ = mass flow rate of return air, lb/min (kg/s)
$\dot{m}_{\text{ex}}$ = mass flow rate of exhaust air, lb/min (kg/s)

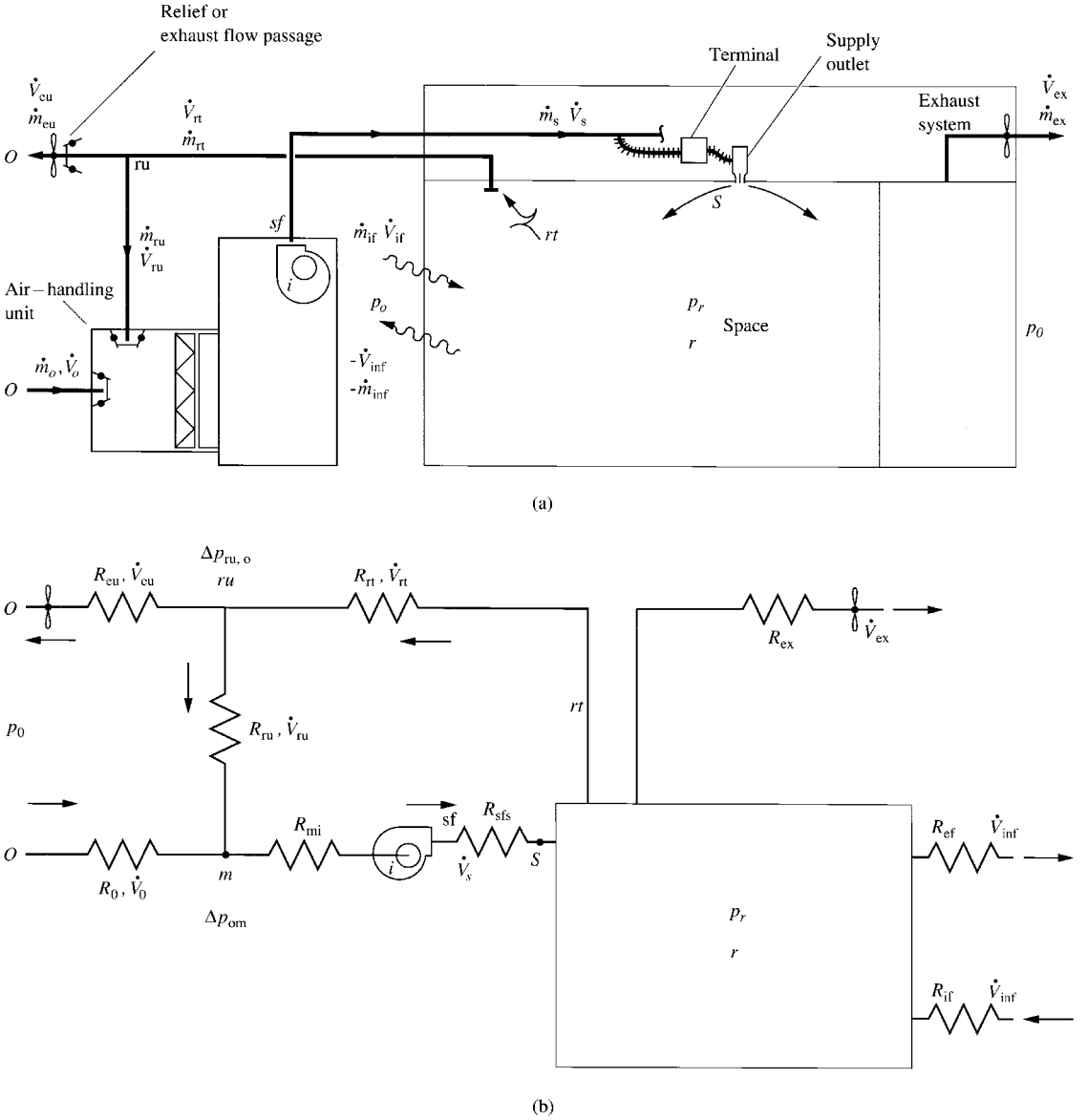


FIGURE 20.3 Mass flow balance of an enclosed space: (*a*) mass flow balance; (*b*) flow circuit.

In Eq. (20.8), $\dot{m}_{\text{inf}}$ indicates the mass flow rate of infiltration air, lb/min (kg/s). The term $-\dot{m}_{\text{inf}}$ means the mass flow rate of exfiltration. Because $\dot{m} = \dot{V}\rho$, for simplicity, if the difference in air density is ignored, then

$$\dot{V}_s + \dot{V}_{\text{inf}} = \dot{V}_{\text{rt}} + \dot{V}_{\text{ex}} \tag{20.9}$$

where $\dot{V}_s$, $\dot{V}_{\text{inf}}$, $\dot{V}_{\text{rt}}$, $\dot{V}_{\text{ex}}$ = volume flow rate of supply, infiltration or exfiltration, return, and exhaust air, cfm (L/s).

For convenience, if the stack effect and wind effect are zero, and the pressure of the surrounded area or outside pressure p_o is equal to the atmospheric pressure and is assumed to be zero, that is, $p_{\text{o}} = p_{\text{at}} = 0$, then the pressure of the enclosed space is represented by p_i, in in. WG (Pa).

Let $\Delta p_{o,\text{st}} = p_{o,\text{st}} - p_{\text{at}}$ be the increase of the outside pressure due to the stack effect, and let $\Delta p_{ow} = p_{ow} - p_{\text{at}}$ be the increase of the outside pressure due to the wind effect; both are in in. WC (Pa).

Space pressurization by using differential flow control is the most widely used method today. When $\Delta p_{o,\text{st}}$ and Δp_{ow} both are zero and the difference in air density is ignored, if an enclosed space is pressurized or depressurized because of differential flow of air (supply and return/exhaust airflow) systems, the following relationships must hold:

$$-\dot{V}_{\text{inf}} = \dot{V}_s - (\dot{V}_{\text{rt}} + \dot{V}_{\text{ex}}) \tag{20.10}$$

From Eq. (20.1)

$$-\dot{V}_{\text{inf}} = 4005 C_d A_{e,l} \sqrt{\Delta(p_{im} - p_o)}$$

and

$$\Delta(p_{im} - p_o) = \left(\frac{\dot{V}_{\text{inf}}}{4005 C_d A_{e,l}} \right) \tag{20.11}$$

From Eq. (20.10), if $\dot{V}_s > (\dot{V}_{\text{rt}} + \dot{V}_{\text{ex}})$, then the enclosed space pressure is pressurized from atmospheric pressure p_o up to p_{im}, in in. WG (Pa), and $\dot{V}_{\text{inf}}$ is negative, that is, $-\dot{V}_{\text{inf}}$, an exfiltration. If $\dot{V}_s < \dot{V}_{\text{rt}} + \dot{V}_{\text{ex}}$, $p_{im} < p_o$, then $\dot{V}_{\text{inf}}$ is positive, an infiltration.

From Eq. (20.11), space pressurization $\Delta(p_{im} - p_o)$ or space depressurization $-\Delta(p_{im} - p_o)$ depends on the amount of differential flow $\dot{V}_s - (\dot{V}_{\text{rt}} + \dot{V}_{\text{ex}}) = \dot{V}_{\text{inf}}$ that has been infiltrated or exfiltrated through the building shell and the air leakage area $A_{e,l}$ (airtightness of the building). The greater the $-\dot{V}_{\text{inf}}$ and the smaller the $A_{e,l}$, the higher will be the pressurization. Similarly, the greater the $\dot{V}_{\text{inf}}$ and the smaller the $A_{e,l}$, the higher the depressurization.

Cummings et al. (1996) made field measurements of uncontrolled airflow and depressurization of eight restaurants. From the measured results, the average building pressures $\Delta(p_{im} - p_o)$ of five restaurants was greater than or equal to -0.032 in. WC (-0.006, -0.003, -0.024, -0.032, and -0.018 in. WC). These are the results of the following two restaurants:

	Floor area, ft	ACH50	$A_{e,l}$, ft^2	OA & MA, cfm	Exhaust, cfm	Building pressure, in. WC (Pa)
Chicken 1	3,161	14.81	3.90	8,410	10,606	-0.032 (7.6)
Chicken 2	3,321	7.70	2.16	6,360	9,272	-0.172 (43)

Here OA represents outdoor air and MA the makeup air. The building pressure of the restaurant had a smaller $A_{e,l} = 2.16$ ft^2 (0.242 m^2), and a greater differential of airflow between exhaust flow and OA and MA flow was depressurized further (-0.172 in. WC or 43 Pa).

Coogan (1996) recommended a space pressurization by differential airflow control using a more complicated room model surrounded by two zones, which is useful when the enclosed space is adjacent to two spaces of different pressures.

20.4 INFILTRATION AND EXFILTRATION

Infiltration is the uncontrolled inward flow of outdoor air through cracks and openings in the building shell when the pressure of the outdoor air is higher than the pressure of the indoor air at the same level. Exfiltration is the uncontrolled outward flow of air through cracks and openings in the building shell when the pressure of indoor air is higher than that of outdoor air. Exfiltration is negative infiltration.

Infiltration and exfiltration can be induced by the stack effect, wind effect, air systems or mechanical ventilating systems, or a combination thereof. If atmospheric pressure is p_{at} zero, the outdoor pressure p_o is the sum of the stack effect $\Delta p_{o,st}$ and wind effect Δp_{ow}, both in in. WC (Pa), and for the building shell at a normal direction to the approaching wind, outdoor pressure p_o, in in. WC (Pa), can be calculated as

$$p_o = \Delta p_{o,st} + \Delta p_{ow} + p_{at} \tag{20.12}$$

If both $p_{o,st}$ and p_{ow} are zero and the space pressure is not pressurized or depressurized, then the indoor pressure $p = p_{at} = 0$. If the space is pressurized by the supply or exhaust system, $p_i = p_{im}$, both in in. WC (Pa).

Volume Flow Rate of Infiltration

The volume flow rate of infiltration $\dot{V}_{inf}$, in cfm (m^3/s), for a residential building can often be calculated by Eq. (20.1) using the effective leakage area $A_{e,l}$. For low-rise commercial buildings in summer, as discussed in Chap. 6, the stack effect is small; if windows are well sealed and the space is pressurized with a positive pressure, the infiltration can be ignored. For low-rise buildings in winter, according to field measurements, the following infiltration rates, in ACH, may be considered in space heating load calculations for rooms with windows on one side of a commercial building that are not well sealed:

Winter mean wind speed v_{met}, fpm (m/s)	Infiltration rate, ACH
1000 (5)	0.2
1200 (6)	0.3

For high-rise buildings in both summer and winter design conditions, the infiltration induced by stack and wind effect should be considered. First, the outdoor pressure due to stack effect $\Delta p_{o,st}$ and wind effect Δp_{ow} for each floor can be calculated by Eqs. (20.4) and (20.6), and the outdoor air pressure p_o can be calculated by Eq. (20.12). As the pressurized indoor pressure is p_{im}, the pressure differece across the building shell $\Delta p = p_o - p_{im}$. Second, from the building construction data, find the area of the exterior wall, in ft^2 (m^2). Third, calculate the volume flow rate of the infiltration $\dot{V}_{inf}$, in cfm (m^3/s), for each floor by the air leakage equation, Eq. (20.2). According to the airtightness of the building, select the flow coefficient C_{flow} as listed previously.

20.5 FAN-DUCT SYSTEMS

A fan-duct system is an idealized air system in which a fan is connected to ductwork or equipment (a flow resistance), or several fans are connected to many sections of ductwork and equipment. More precisely, a fan-duct system is a combination of fan and flow resistance. Fan-duct systems are used to study and analyze the operating characteristics of air systems.

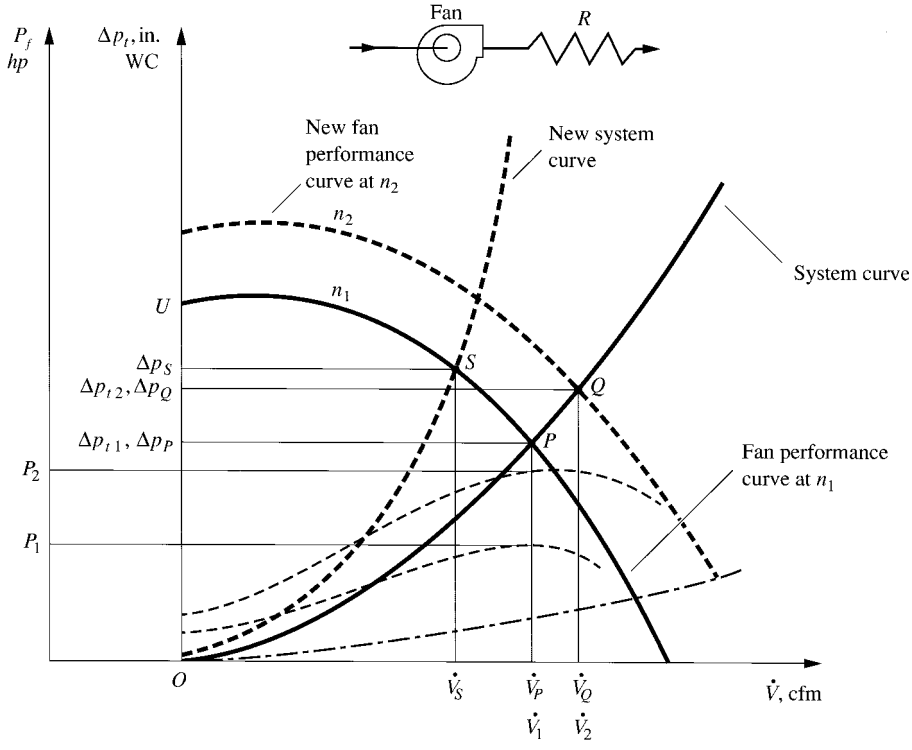


FIGURE 20.4 System operating point.

System Operating Point

A system operating point indicates the operating condition and characteristics of a fan-duct system. Because the operating point of the fan must lie on the fan performance curve and the operating point of the duct system must lie on the system curve, the system operating point of a fan-duct system must be the intersection point P of the fan performance curve and the system curve, as shown in Fig. 20.4. At the operating point of the fan-duct system, its volume flow rate is $\dot{V}_P$, and its total pressure loss is Δp_P.

If the flow resistance of the duct system remains the same while fan performance changes, and its characteristics are represented by a new fan performance curve, the operating point of the fan-duct system moves to point Q in Fig. 20.4. On the other hand, if the fan performance curve remains the same while the flow resistance of the duct system is varied and represented by a new system curve, the operating point of the fan-duct system moves from P to S.

The fan in a constant-volume air system should be selected with a $\dot{V}$ and Δp_t so that the system operating point P is near the point of maximum fan total efficiency η_t. The actual operating point of a constant-volume system is usually within a range of $\pm$ 10 percent of the wide-open volume from the point of maximum η_t.

Buckingham π Method and Fan Laws

The relationship between various interdependent variables that represent fan characteristics for dynamically similar systems can be determined by using the Buckingham π method. For two systems

to be dynamically similar, they must be similar in shape and construction, i.e., geometrically similar. Their velocity distribution or profile of fluid flow should also be similar. The Buckingham π method gives the relationship as

$$\pi = FD^a n^b \rho^c \tag{20.13}$$

where F = dependent variable such as $\dot{V}$, Δp_t, or power
D = diameter of impeller, ft (m)
n = fan speed, rpm
ρ = air density, lb/ft^3 (kg/m^3)

In Eq. (20.13), a, b, c are indices to be determined so that the π group as a whole is dimensionless.

Variables corresponding to fan-duct systems and their dimensions in terms of mass M, length L, and time t are as follows:

		Indices		
Variable	Unit	M	L	t
Volume flow $\dot{V}$	cfm (m^3/s)	0	3	−1
Total pressure Δp_t	lb$_f$/ft^2 (N/m^2)	1	−1	−2
Power P_f	ft·lb$_f$/min (kJ/s)	1	2	−3
Impeller diameter D	ft (m)	0	1	0
Speed n	rpm	0	0	1
Air density ρ	lb$_m$/ft^3 (kg/m^3)	1	−3	0

For the dimensionless group π_1 including dependent variable volume flow $\dot{V}$ and independent variables D, n, and ρ, their relationship can be found by determining the indices as follows:

$$\pi_1 = \dot{V} D^a n^b \rho^c$$
$$[0] = [L^3 t^{-1}][L]^a [t^{-1}]^b [ML^{-3}]^c$$

Setting up equations in M, L, and t, we have

$$M \quad 0 = c$$
$$L \quad 0 = 3 + a - 3c \qquad a = -3$$
$$t \quad 0 = -1 - b \qquad b = -1$$

That is,

$$\pi_1 = \dot{V} D^{-3} n^{-1} = \frac{\dot{V}}{D^3 n}$$

Or, $\dot{V}/(D^3 n)$ = constant, then

$$\dot{V} = K_v D^3 n \tag{20.14}$$

where K_v = volume flow constant. Similarly,

$$\Delta p_t = K_p D^2 n^2$$
$$P_f = K_P D^5 n^3 \rho \tag{20.15}$$

where K_p = pressure constant and K_P = power constant.

Equations (20.14) and (20.15) can be used to determine the performance of a fan-duct system under the following two conditions:

- *Condition 1.* For the same fan-duct system operating at different speeds $D_1 = D_2$, if the difference in air density is negligible, the volume flow rate ratio is proportional to the speed ratio, or

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{K_v D_2^3 n_2}{K_v D_1^3 n_1} = \frac{n_2}{n_1} \tag{20.16}$$

In Eq. (20.16), subscripts 1 and 2 indicate the original and changed operating conditions, as shown in Fig. 20.4. The total pressure rise ratio is equal to the square of the speed ratio

$$\frac{\Delta p_{t2}}{\Delta p_{t1}} = \frac{K_p D_2^2 n_2^2 \rho_2}{K_p D_1^2 n_1^2 \rho_1} = \frac{n_2^2}{n_1^2} \tag{20.17}$$

The fan power ratio is equal to the cube of the speed ratio

$$\frac{P_2}{P_1} = \frac{K_P D_2^5 n_2^3 \rho_2}{K_P D_1^5 n_1^3 \rho_1} = \frac{n_2^3}{n_1^3} \tag{20.18}$$

- *Condition 2.* For fan-duct systems with geometrically and dynamically similar fans and the same duct system, i.e., the same type of fans having the same K_v, K_p, and K_P, operating at high Reynolds numbers and installed in the same duct system, the volume flow ratio, total pressure rise ratio, and power ratio can be expressed as

$$\begin{aligned} \frac{\dot{V}_2}{\dot{V}_1} &= \frac{K_v D_2^3 n_2}{K_v D_1^3 n_1} = \frac{D_2^3 n_2}{D_1^{3+} n_1} \\ \frac{\Delta p_{t2}}{\Delta p_{t1}} &= \frac{K_p D_2^2 n_2^2 \rho_2}{K_p D_1^2 n_1^2 \rho_1} = \frac{D_2^2 n_2^2}{D_1^2 n_1^2} \\ \frac{p_2}{p_1} &= \frac{K_p D_2^5 n_2^3 \rho_2}{K_p D_1^5 n_1^3 \rho_1} = \frac{D_2^5 n_2^3}{D_1^5 n_1^3} \end{aligned} \tag{20.19}$$

Equations (20.16) through (20.19) relate the parameters of the fan-duct system at various operating conditions. They are often called the fan laws.

20.6 SYSTEM EFFECT

Mechanism of System Effect

The system effect of a fan-duct system describes the loss of fan performance caused by uneven and nonuniform velocity profiles at the inlet entering and after the fan outlet using actual operating inlet and outlet connections, compared to the performance of that fan test unit during laboratory ratings.

Individual fans are tested based on ANSI/ASHRAE Standard 51-1985 and ANSI/AMCA Standard 210-85. The configuration of the test unit is shown in Fig. 20.5*b*. There are no inlet connections. There is also a minimum length of 10 duct diameters of straight duct connected to the fan outlet.

The system effect of a fan-duct system consists of inlet system effect and outlet system effect.

Inlet System Effect

Consider a 90° elbow that is connected to the fan inlet with a very short connecting duct between the elbow and the fan inlet. Eddies, large-scale turbulence, and a nonuniform velocity profile form after the elbow and are not smoothed to a uniform velocity distribution because of the short connecting duct. Air enters the fan with a nonuniform velocity profile that is quite different from the

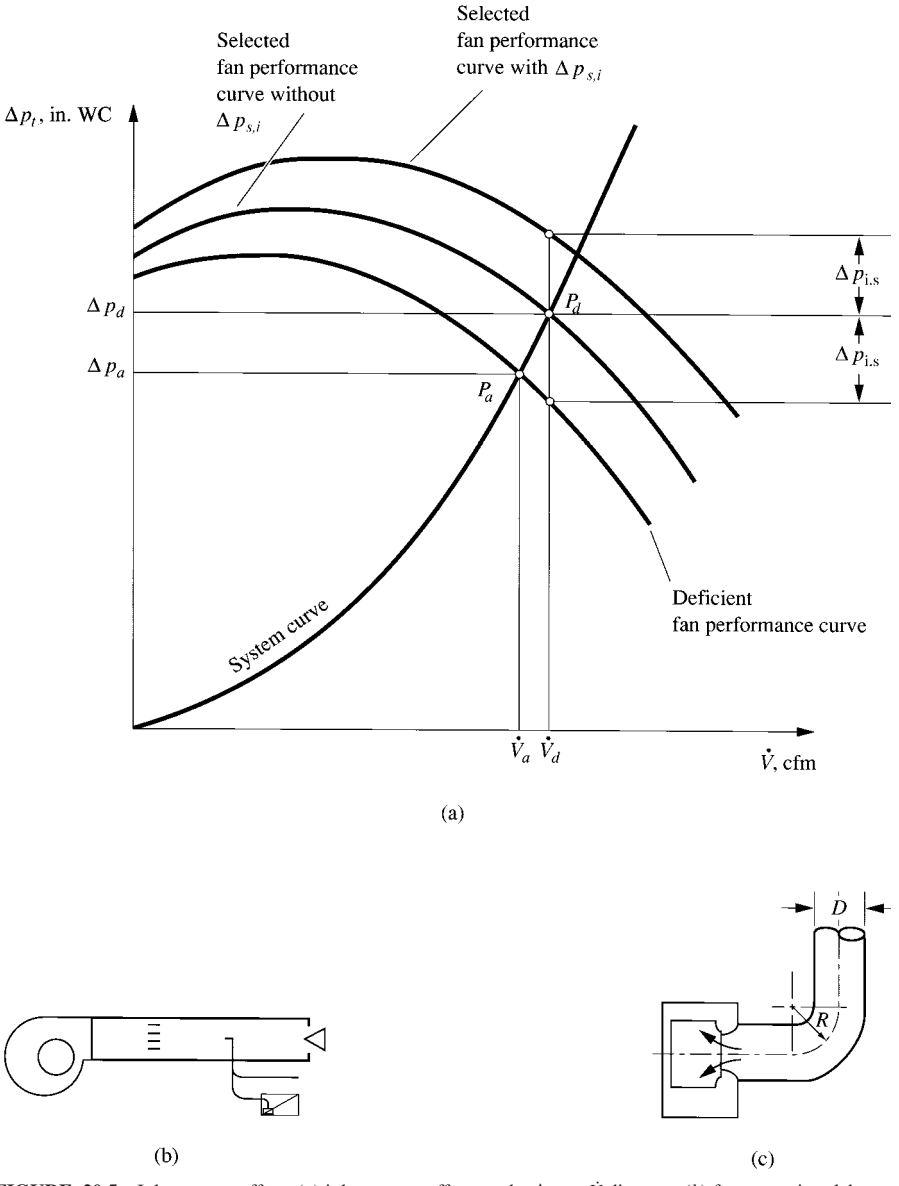


FIGURE 20.5 Inlet system effect: (*a*) inlet system effect on the $\Delta p_t - \dot{V}$ diagram; (*b*) fan test unit at laboratory; (*c*) round elbow before fan inlet.

fan inlet conditions in the standard laboratory test unit. This results in a fan performance curve with lower fan total pressure and volume flow than the catalog fan performance curve, as shown in Fig. 20.5*a*.

The difference between the fan total pressure in the fan-duct system with a nonuniform velocity profile at fan inlet and the fan total pressure during standard rating at the laboratory is called the inlet system effect. Fans must be selected at a fan total pressure that compensates for the inlet system effect, so that the actual fan performance curve can meet the required fan total pressure at designed volume flow at operating point P_d, as shown in Fig. 20.5*a*.

If a fan-duct system does not consider the pressure loss caused by the inlet system effect, it will actually operate at point P_a with a lower volume flow and fan total pressure than the calculated values, even though the calculated design operating point is at P_d.

Inlet System Effect Loss

Inlet system effect loss $\Delta p_{s,i}$, in in. WC (Pa), is the difference in pressure loss between actual and catalog performance curves caused by the inlet system effect, and it can be calculated as dynamic loss, as defined in Eq. (17.52)

$$\Delta p_{s,i} = C_{s,i}\left(\frac{v_{fi}}{4005}\right)^2 \tag{20.20}$$

where $C_{s,i}$ = inlet system effect loss coefficient. In Eq. (20.20), v_{fi} is the air velocity at the fan inlet based on the area of the inlet collar, fpm (m/s).

Inlet system effect loss should be added to the system total pressure loss calculated at the designed volume flow rate, as described in Eq. (17.23). The selected catalog fan performance curve should meet the system total pressure loss, including the inlet system effect loss.

Inlet System Effect Loss Coefficient

According to the Air Movement and Control Association (AMCA) *Fan and Systems*, Publication 201 (1990), if an elbow is connected by a round duct to the inlet of a centrifugal fan and the length of the connecting duct is equal to 2 duct diameters, the inlet system effect loss coefficients are as follows:

R/D of elbow	0.75	1.0	2.0
$C_{s,i}$	0.8	0.66	0.53

Here R is the radius of the elbow, and D is the duct diameter, both in ft (m). If the length of the connecting duct is 5D, then $C_{s,i}$ is approximately one-half the values listed above. If there is no connecting duct, then $C_{s,i}$ is about twice the values listed above.

According to AMCA (1973), for a square elbow (with or without turning vanes) with an inlet transition and a connecting duct of length 2D, as shown in Fig. 20.6, the inlet system effect loss coefficients are as follows:

R/H of square elbow	0.5	0.75	1.0	2.0
$C_{s,i}$ without turning vanes	1.6	1.2	0.66	0.47
$C_{s,i}$ with turning vanes	0.47	—	0.33	0.22

Here H indicates the dimension of either side of the square duct, in ft (m).

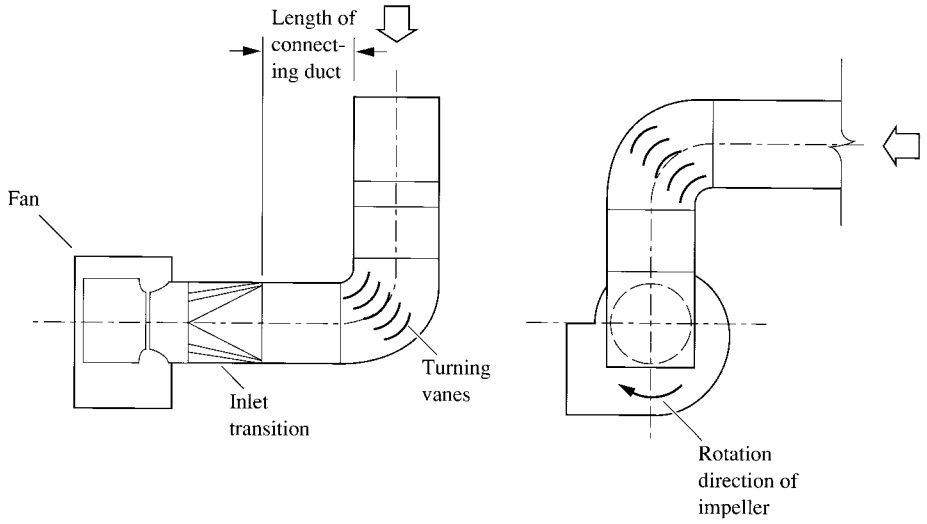


FIGURE 20.6 Square elbow with turning vanes before fan inlet.

For a square elbow before a fan inlet with a connecting duct of 5*D* and with turning vanes and an inlet transition, $C_{s,i}$ is approximately 0.6 of the value listed above.

If a square elbow before a fan inlet has turning vanes and an inlet transition with no connecting duct, for $R/H = 1$, $C_{s,i} = 0.53$; and for $R/H = 0.5$, $C_{s,i} = 0.8$.

For double-width double-inlet fan installed in a plenum or in a cabinet, the distance between the fan inlet and the wall of the plenum or cabinet should not be less than 0.4 diameter of the fan inlet. If it is only 0.2 diameter of the fan inlet, an inlet system effect loss of $C_{s,i} = 0.08$ should be added to the system total pressure loss at the design volume flow rate. For details, refer to AMCA *Fan and Systems*, Publication 201 (1973).

The fans in most air-handling units and packaged units installed according to ASHRAE and ARI standards are rated and tested as complete units. The condition of fan inlet of the actual operating fan-duct system is usually the same as the rated condition. There is no inlet system effect loss in such a unit.

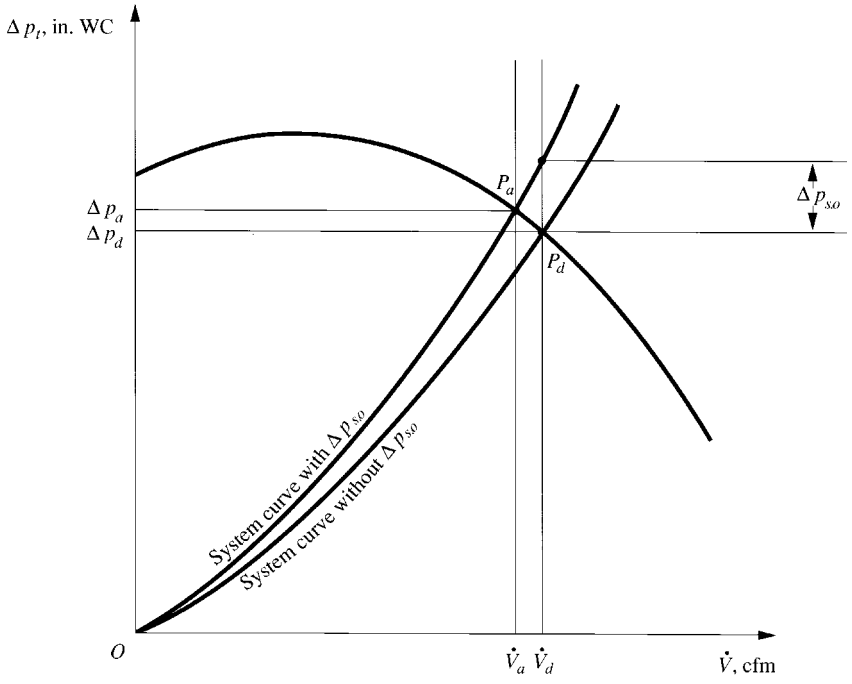
Outlet System Effect

If a duct fitting is connected to a fan outlet with a short connecting duct, its dynamic loss Δp_{dy} and local loss coefficient C_{dy} increase because of the nonuniform velocity profile of the airstream discharged from the fan outlet. The difference between the dynamic losses of a duct fitting connected to a fan outlet with a nonuniform and uniform velocity profile at the duct fitting inlet is known as the outlet system effect loss $\Delta p_{s,o}$, in in. WC (Pa). The outlet system effect results in an increase of flow resistance of the duct system, as shown in Fig. 20.7*a*.

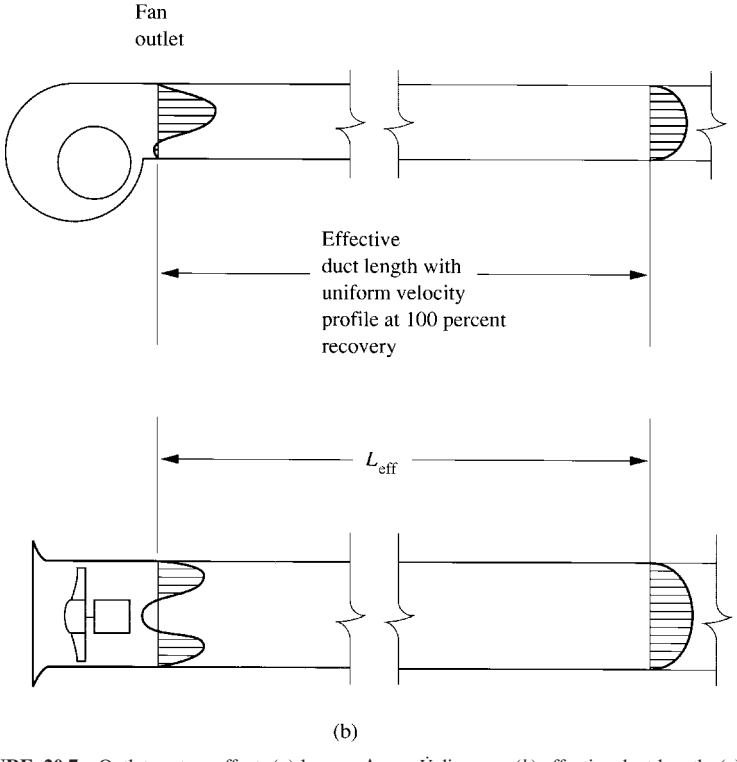
As in Eq. (20.20), outlet system effect loss at design volume flow rate can be calculated as

$$\Delta p_{s,o} = C_{s,o}\left(\frac{v_{fo}}{4005}\right)^2 \tag{20.21}$$

where $C_{s,o}$ = outlet system effect loss coefficient
v_{fo} = fan outlet velocity based on fan outlet area, fpm (m/s)

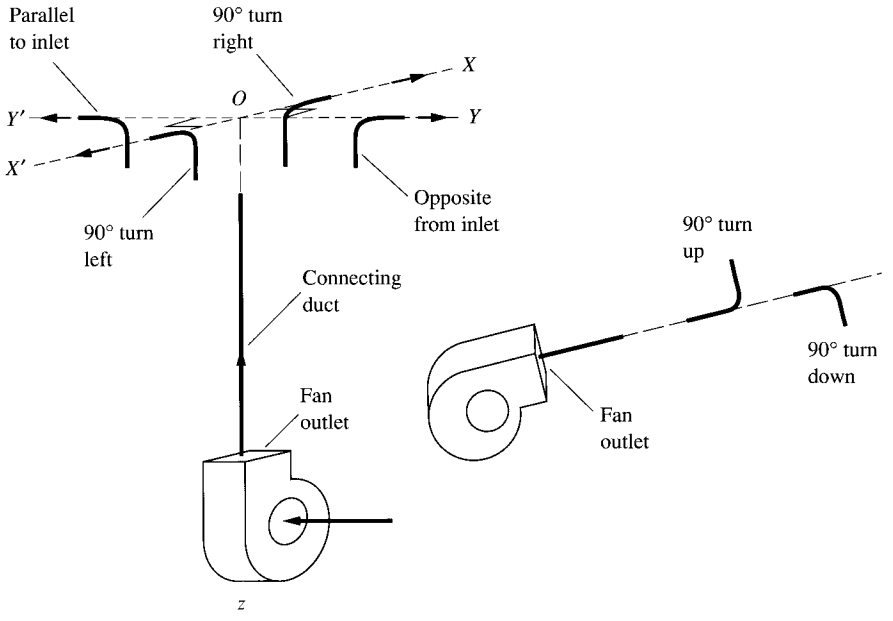


(a)



(b)

FIGURE 20.7 Outlet system effect: (*a*) loss on $\Delta p_t - \dot{V}$ diagram; (*b*) effective duct length; (*c*) orientation of elbow connected to duct.



(c)

FIGURE 20.7 (*Continued*)

Outlet system effect loss at the design volume flow should be added to the calculated total system pressure loss.

Outlet System Effect Loss Coefficient

Outlet system effect loss $\Delta p_{s,o}$ and outlet system effect loss coefficient $C_{s,o}$ both depend on the effective duct length ratio L/L_{eff}, the air velocity at the fan outlet v_{fo}, and the configuration of the duct fitting with respect to the fan outlet. Here L indicates the length of duct section between fan outlet and the duct fitting in duct diameters.

The effective duct length L_{eff} is the length of duct between the fan outlet and a cross-sectional plane at which the uniform velocity profile has been completely recovered, in duct diameters or equivalent diameters D and L and L_{eff} must be expressed in the same units. Effective duct length can be evaluated as

$$L_{\text{eff}} = 2.5 + \frac{v_{fo} - 2500}{1000} \tag{20.22}$$

For a single-width single-inlet (SWSI) centrifugal fan, the configurations of a 90° elbow with respect to the fan outlet at the end of the connecting duct can be classified as 90° turn right or 90° turn down, 90° turn left or 90° turn up, parallel with inlet, and opposite to inlet, as shown in Fig. 20.7*c*.

According to AMCA *Fan and Systems*, Publication 201 (1973), the magnitudes of $C_{s,0.8}$ at various L/L_{eff} values and elbow orientations with a blast area to fan outlet area ratio $A_{\text{blast}}/A_{\text{out}} = 0.8$ are as follows:

	L/L_{eff}			
	0.12	0.25	0.5	1.0
90° turn right or down	0.64	0.44	0.20	0
90° turn left or up, parallel with inlet	1.16	0.76	0.36	0
90° turn left or up, opposite to inlet	0.80	0.52	0.24	0

For a SWSI centrifugal fan with $A_{blast}/A_{out} = 0.7$, a multiplier $K_{blast} = 1.4$ should be applied to the above values. If $A_{blast}/A_{out} = 0.6$, then $K_{blast} = 2$; and if $A_{blast}/A_{out} = 0.9$, then $K_{blast} = 0.75$.

For double-width double-inlet (DWDI) centrifugal fans, an additional multiplier K_{DI} should be applied to the $C_{s,0.8}$ values. For a 90° elbow connected to the duct at horizontal turning orientation, $K_{DI} = 1.25$. The outlet system effect coefficient $C_{s,o}$ can then be determined as

$$C_{s,o} = K_{DI}K_{blast}C_{s,0.8} \tag{20.23}$$

For details, refer to AMCA, *Fan and Systems* (1973).

For fans installed in air-handling units and packaged units and rated as complete units, the ratings are based on a straight duct length of 2 duct diameters from the unit outlet. The outlet system effect loss is not included in the laboratory-rated fan performance curve.

Selecting Fans Considering System Effect Losses

If the calculated volume flow rate of the design operating point P_d (as shown in Fig. 20.8) is $\dot{V}_d$, in cfm (L/s), and the calculated system total pressure loss is Δp_{sy}, in in. WC (Pa), the selected fan that

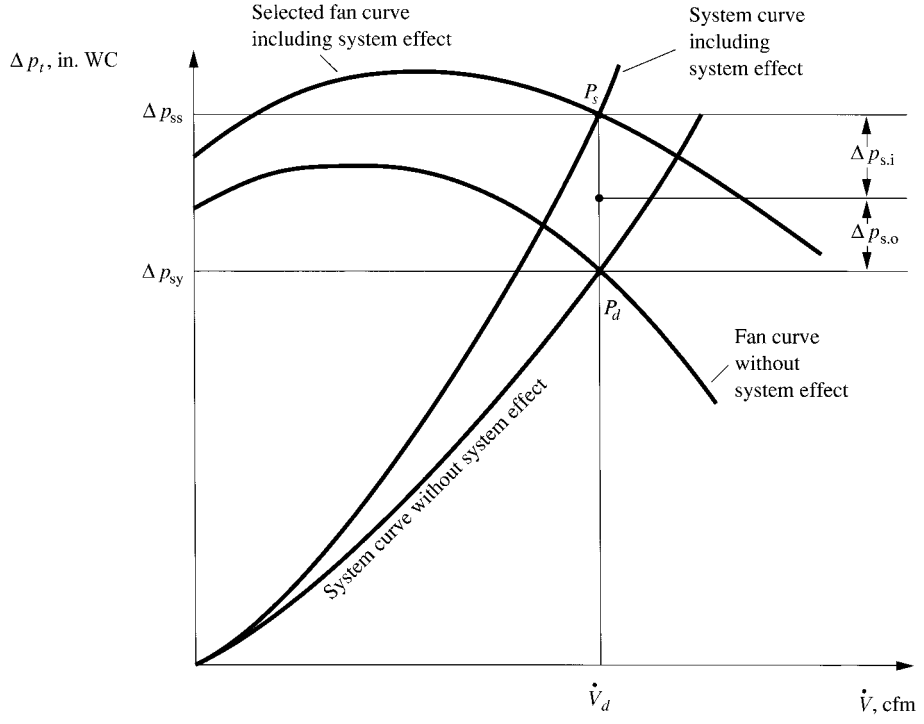


FIGURE 20.8 Selecting fan with performance including system effect loss.

compensates for inlet and outlet system effect losses must operate on a performance curve that intersects with point P_s, which has a volume flow rate of $\dot{V}_d$ and a total pressure rise of Δp_{ss} including system effect losses, in in. WC (Pa). That is,

$$\Delta p_{ss} = \Delta p_{sy} + \Delta p_{s,i} + \Delta p_{s,o} \tag{20.24}$$

System effect directly affects the performance of the air system as well as the function of the air conditioning system. Because the air velocity at the fan inlet and outlet is usually between 1500 and 3000 fpm (7.5 and 15 m/s), system effect losses may amount to 5 to 20 percent of the system pressure loss of the air system.

Inlet and outlet system effect losses must be calculated correctly. Moreover, air systems must be designed to reduce system effect losses to save energy.

20.7 COMBINATION OF FAN-DUCT SYSTEMS

The purposes of combining fan-duct systems are as follows:

- To increase the volume flow rate of the combined fan-duct system
- To provide the required pressure characteristics for the combined fan-duct system
- To balance the volume flow of the combined fan-duct system
- To maintain a required space pressure characteristic

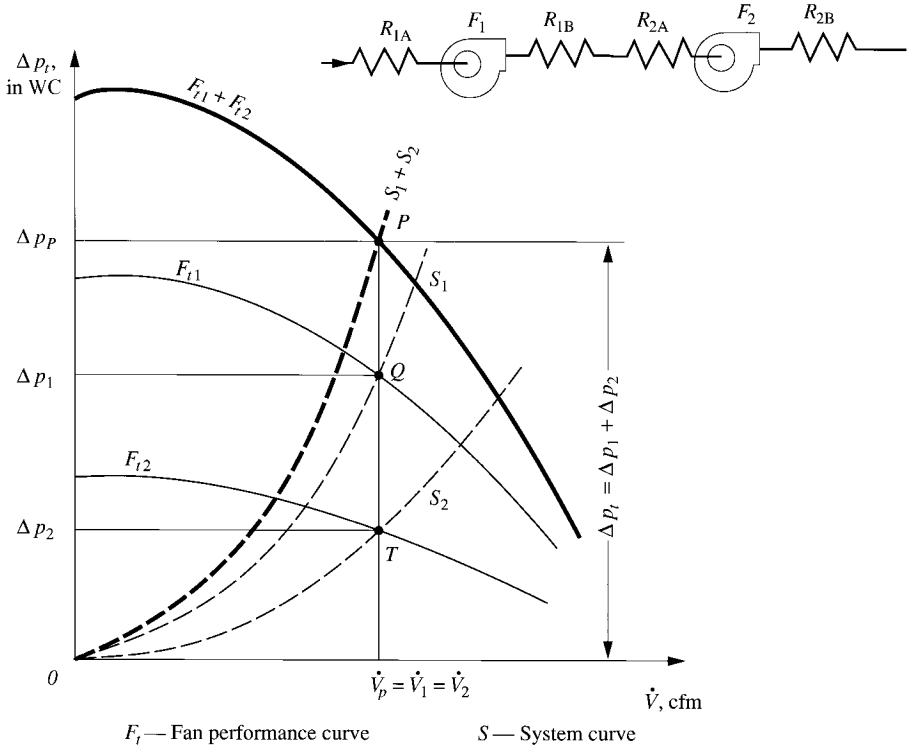


FIGURE 20.9 Two-fan fan-duct system connected in series.

Two Fan-Duct Systems Connected in Series

Figure 20.9 shows two fan-duct systems connected in series equipped with fans F_1 and F_2 and the corresponding two duct systems represented by flow resistances R_1 and R_2.

Flow resistance R_1 may be the combination of two flow resistances R_{1A} and R_{1B} connected in series, and flow resistance R_2 may be the combination of two flow resistances R_{2A} and R_{2B}. The volume flow rate of the combined fan-duct system $\dot{V}_p$, in cfm (L/s), must be the same as the volume flow rate $\dot{V}_1$ or $\dot{V}_2$ of fan-duct system 1 or 2 and their components fan F_1, fan F_2, or

$$\dot{V}_p = \dot{V}_1 = \dot{V}_2 \tag{20.25}$$

The total pressure loss across flow resistance R_1 has a magnitude Δp_1, and the relationship $\Delta p_1 = R_1 \dot{V}_1^2$ holds. This relationship also determines the system curve S_1. Here the total pressure loss in the duct system includes system effect loss. Fan performance curve F_{t1} can be obtained from the data given in the manufacturer's catalog.

Total pressure loss Δp_1 and volume flow $\dot{V}_1$ also determine the operating point of fan-duct system 1 at point Q, which is graphically the intersection of fan performance curve F_{t1} and system curve S_1. For fan-duct system 2, Δp_2, $\dot{V}_2$, R_2, fan performance curve F_{t2}, system curve S_2, and system operating point T can be similarly determined.

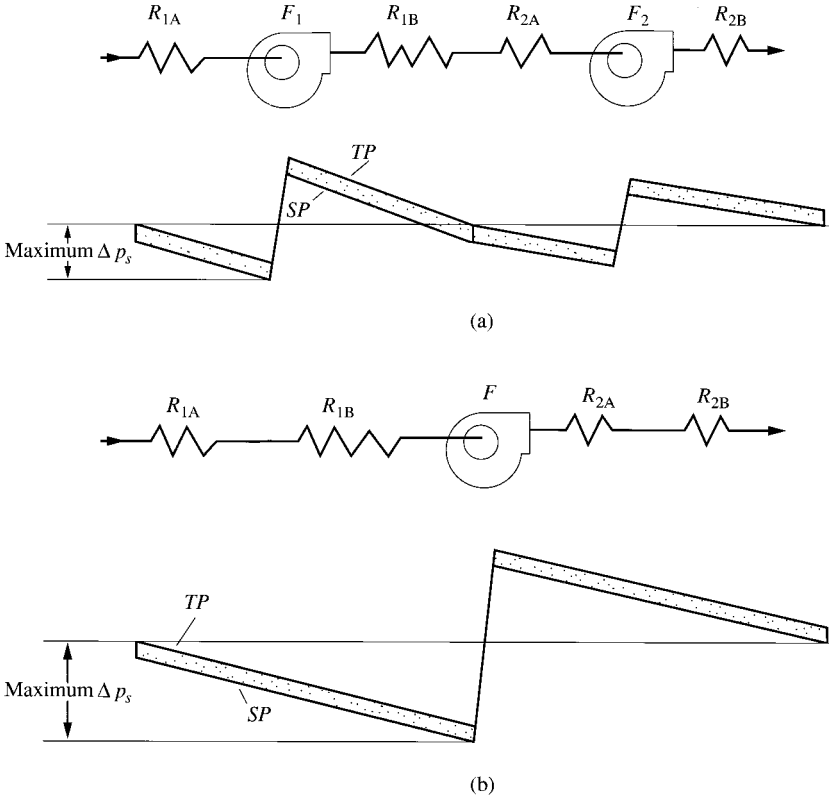


FIGURE 20.10 Comparison of pressure characteristics: (*a*) combined fan-duct system; (*b*) single fan-duct system.

The total pressure loss of the combined fan-duct system Δp_t, in. WC (Pa), is

$$\Delta p_t = \Delta p_1 + \Delta p_2 \tag{20.26}$$

Graphically, the combination of the curves of two fans connected in series on a Δp_t-$\dot{V}$ diagram can be performed by drawing several constant-volume flow rate lines. For each constant-volume line, the fan total pressure of the combined fan curve $F_{t1} + F_{t2}$ always equals the sum of the sections of fan total pressure of fan curve 1, F_{t1} and fan curve 2, F_{t2}, represented by Δp_1 and Δp_2, as shown in Fig. 20.9. The purpose of connecting two fan-duct systems in series is to increase the fan total pressure that can be provided by the combined system.

Figure 20.10 shows the system pressure characteristics of a combined fan-duct system with fans F_1 and F_2 and a single fan-duct system that uses only one fan F. It can be seen that the static pressure across the duct wall is considerably higher in a fan-duct system with a single fan than in the combined fan-duct system with two fans. Of course, a single-fan system has a lower initial cost.

For two fan-duct systems connected in series, the volume flow rates of the two fans should be similar. If a large fan is connected in series with a small fan, at large volume flow rates, the fan total pressure of the combined system may be less than if the large fan were operated alone. This loss of efficiency results from the influence of negative total pressure of the small fan when it is operated at a volume flow rate greater than its wide-open volume flow.

Fans Combined in Parallel and Connected in Series with a Duct System

When two fans F_1 and F_2 combined in parallel are connected to a duct system represented by flow resistance R, as shown in Fig. 20.11, the volume flow rate of the parallel combined fans $\dot{V}_P$ is the sum of the volume flow rates of individual fans $\dot{V}_1$ and $\dot{V}_2$.

The purpose of such a combination is to increase the volume flow rate of the system. Usually, the fans in parallel combination are identical fans, i.e., fans of the same model with the same rated

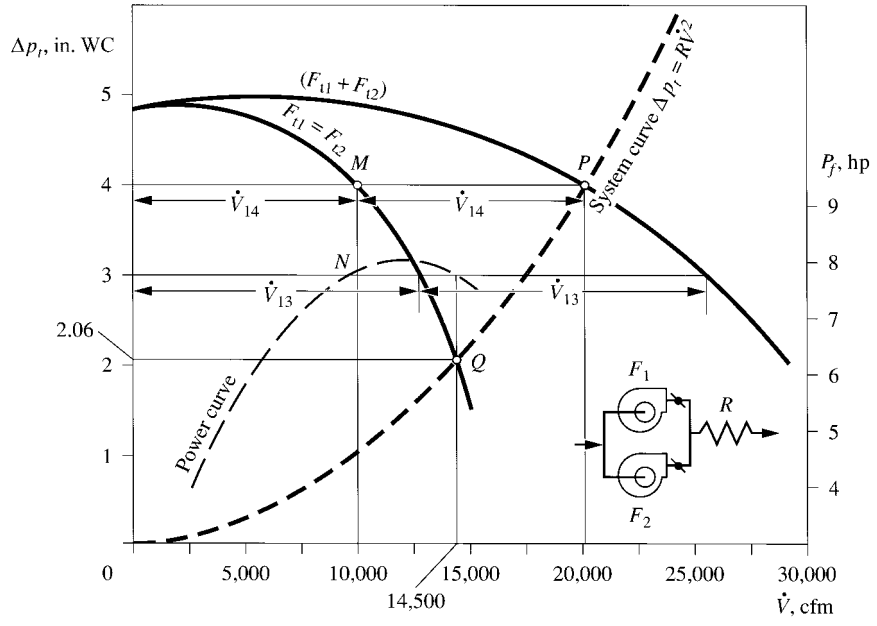


FIGURE 20.11 Two fans combined in parallel and connected in series with a duct system.

volume flow and fan total pressure. As in fan-duct systems connected in series, a large fan should not be combined in parallel with a small fan because of the possible negative effect of the small fan.

The procedure for determining the combined fan performance curve $F_{t1} + F_{t2}$ of the two identical fans connected in parallel is as follows:

1. Plot the fan performance curves F_{t1} and F_{t2} according to the manufacturer's data.
2. Draw the constant total pressure loss Δp_t lines (horizontal lines). Points on the combined curve along the constant Δp_t lines have a volume flow of $2\dot{V}_1$. Here $\dot{V}_1$ indicates the volume flow of each identical fan along the constant Δp_t lines.
3. Plot the combined curve $F_{t1} + F_{t2}$ by connecting points having $\dot{V} = 2\dot{V}_1$ on the constant Δp_t lines.

The system curve can be plotted based on the relationship $\Delta p_t = R\dot{V}^2$. The intersection of the combined curve $F_{t1} + F_{t2}$ and the system curve is the system operating point P. The volume flow at the system operating point $\dot{V}_P$, in cfm (L/s), is the volume flow rate flowing through the duct system at a flow resistance R. The combined system total pressure at the system operating point Δp_P, in in. WC (Pa), is the fan total pressure of the identical fans or the system total pressure loss of the duct system.

Example 20.1. An air system uses two identical fans combined in parallel and connected in series with a duct system, as shown in Fig. 20.11. The pressure-volume and power-volume characteristics of each of the identical fans are listed below:

$\dot{V}$, cfm	2500	5000	7500	10,000	12,500	15,000
Δp_t, in. WC	5.00	4.88	4.54	4.00	3.13	1.75
Power P_f, hp	3.93	5.90	7.14	7.87	8.20	7.60

If the total pressure loss of this air system is known to be 1 in. WC (250 Pa) at an airflow of 10,000 cfm (4719 L/s), determine the following:

1. The system operating point, the corresponding air volume flow rate, and the system total pressure when both fans are operating
2. The operating point, the corresponding airflow rate, and the fan total pressure when only one fan is running, assuming that the air damper after the other fan is closed
3. The fan total efficiency when two fans or only one fan is running

Solution

1. From the given data, the flow resistance of this air system is

$$R = \frac{\Delta p_t}{\dot{V}^2} = \frac{1}{(10{,}000)^2} = 1 \times 10^{-8}\ \text{in. WC/(cfm)}^2$$

The system pressure loss at various airflow rates is as follows:

$\dot{V}$, cfm	5000	10,000	15,000	20,000	25,000
Δp_t, in. WC	0.25	1.0	2.25	4.0	6.25

After the single-fan performance curve is plotted, the combined curve of the two fans in parallel can be drawn. Plot the system curve from the data given above. The volume flow rate and the

system pressure loss at operating point P, which is the intersection of the combined curve and the system curve when two fans are running (as shown in Fig. 20.11), are found to be

$$\dot{V}_p = 20{,}000 \text{ cfm } (9440 \text{ L/s}) \qquad \Delta p_P = 4.0 \text{ in. WC } (995 \text{ Pa})$$

2. When only one fan is operating and the damper after the other fan is closed, the intersection of the single-fan curve and the system curve is at point Q (see Fig. 20.11). At point Q

$$\dot{V}_Q = 14{,}500 \text{ cfm } (6840 \text{ L/s}), \qquad \Delta p_Q = 2.06 \text{ in. WC } (510 \text{ Pa})$$

3. When two fans are running, each fan has a volume flow of $20{,}000/2 = 10{,}000$ cfm and a fan total pressure of 4 in. WC. From the power curve, the corresponding fan power input is 7.75 hp. From Eq. (15.6), the fan total efficiency is then

$$\eta_{\text{two}} = \frac{\Delta p_t \dot{V}}{6356 P_f} = \frac{4 \times 10{,}000}{6356 \times 7.75} = 0.81, \text{ or } 81 \text{ percent}$$

When only one fan is running, the corresponding fan power input is also 7.75 hp. The fan total efficiency is

$$\eta_{\text{one}} = \frac{2.06 \times 14{,}500}{6356 \times 7.75} = 0.61, \text{ or } 61 \text{ percent}$$

Two Parallel Fan-Duct Systems Connected with Another Duct System

When two parallel fan-duct systems are connected in series to a third duct system represented by flow resistance R_3, as shown in Fig. 20.12, the volume flow rate $\dot{V}_3$ flowing through R_3 is the sum of volume flow rates flowing through the parallel combined fan-duct systems $\dot{V}_1$ and $\dot{V}_2$

$$\dot{V}_{BC} = \dot{V}_3 = \dot{V}_1 + \dot{V}_2$$

Although the total pressure loss Δp_{AB} between points A and B of parallel combined fan-duct systems 1 and 2 is the same, flow resistances R_1 and R_2 may be different from each other; therefore, at their own operating points, the fan total pressure of fan F_1, denoted by Δp_{t1}, may also be different from the fan total pressure of fan F_2, Δp_{t2}, even though fans F_1 and F_2 are identical.

Through the concept of residual pressure of a fan-duct system, the performance of two parallel combined fan-duct systems can be determined. When fan F_1 is connected in series with flow resistance R_1, the total pressure loss across R_1 is $\Delta p_{R1} = R_1 \dot{V}^2$. If the volume flow rate of air flowing through R_1 is of a given value $\dot{V}_1$, the residual pressure after flowing through the flow resistance R_1, Δp_{res1}, in in. WC (Pa), is

$$\Delta p_{\text{res 1}} = \Delta p_{F1} - \Delta p_{R1} = \Delta p_{F1} - R_1 \dot{V}_1^2 = F_{t1} - S_1 \tag{20.27}$$

where Δp_{F1}, F_{t1} = fan total pressure of fan F_1, in. WC (Pa)
Δp_{R1}, S_1 = total pressure loss across flow resistance R_1 or total pressure loss of duct system 1, in. WC (Pa)

For fan-duct system 2, residual pressure can be similarly calculated.

For two fan-duct systems connected in parallel, the residual pressure of fan-duct system 1 after R_1, Δp_{res1}, must be equal to the residual pressure of fan-duct system 2 after R_2, Δp_{res2}, because there is only one unique total pressure p_{tB} at point B.

If two parallel combined fan-duct systems are connected in series with another duct system having flow resistance R_3, the residual pressure provided after fan-duct systems 1 and 2 can be used to overcome the total pressure loss across flow resistance R_3 is as follows:

$$\Delta p_{\text{res1}} = \Delta p_{\text{res 2}} = R_3 \dot{V}_3^2 \tag{20.28}$$

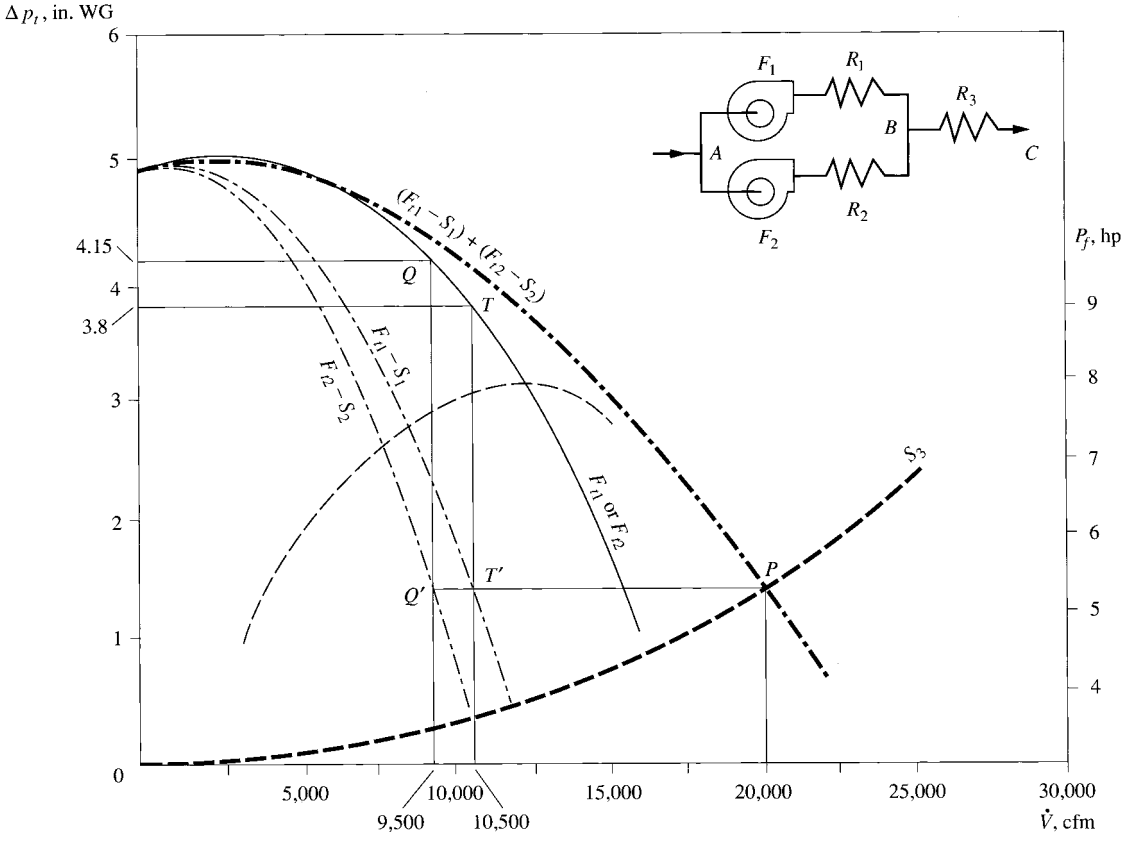


FIGURE 20.12 Two parallel fan-duct systems connected with another duct system.

Because the pressure at points A and C is equal to atmospheric pressure (zero), the performance of two parallel combined fan-duct systems connected in series with another duct system can be determined as follows:

1. Plot two fan curves F_{t1} and F_{t2}, the residual pressure curves $F_{t1} - S_1$ and $F_{t2} - S_2$, and the combined residual pressure curve $F_{t1} - S_1 + F_{t2} - S_2$. The residual pressure $F_t - S$ curve can be plotted as follows:

- At a volume flow of $\dot{V}_A$, find a corresponding fan total pressure $\Delta p_{tfA} = F_{tA}$ from the fan curve given by the manufacturer.
- Calculate the pressure drop across the duct system at a volume flow $\dot{V}_A : S_A = \Delta p_{tRA} = R\dot{V}_A^2$.
- Calculate residual pressure $\Delta p_{\text{res}A} = F_{tA} - S_A$ at a volume flow of $\dot{V}_A$.
- Plot point A on the residual pressure $F_t - S$ curve at a volume flow of $\dot{V}_A$ and a residual pressure of $\Delta p_{\text{res}A}$.
- Similarly, plot points B, C, . . . on the residual pressure curve.
- Residual pressure curve $F_t - S$ can be drawn by joining points A, B, C, . . . etc.

At each point on a residual curve, there is a corresponding residual pressure $\Delta p_{\text{res}} = (F_t - RV^2)$ that can be provided at a corresponding volume flow of $\dot{V}$ to overcome the flow resistance R_3 of a

connected series. Combining of residual pressure curves is performed in the same way as combining of fan curves.

2. Calculate R_3 from the required volume flow rate and total pressure loss in that duct system represented by R_3. From the relationship $\Delta p_t = R_3 \dot{V}_3^2$, draw system curve S_3.

3. Determine the intersection point P of the combined residual pressure curve $F_{t1} - S_1 + F_{t2} - S_2$ and the system curve S_3, which denotes the operating condition at R_3. The residual pressure at point P is the pressure that can be provided by fan-duct systems 1 and 2 to overcome the total pressure loss across R_3 at that volume flow rate $\dot{V}_3 = \dot{V} + \dot{V}_2$. In fan-duct system design, if the volume flow rate through R_3 thus found is greater or smaller than the required volume flow rate, select a larger or smaller fan in duct systems 1 or 2 to make them approximately the same.

4. Draw a horizontal line PQ' from point P, which intersects with residual pressure curves at points T' and Q', and draw vertical lines QQ' and TT' from points Q' and T'. Points Q and T on fan curve F_1 or F_2 indicate the volume flow, fan total pressure of the fan-duct systems 1 and 2, and fan power input to fan F_1 and fan F_2.

Example 20.2. An air system is equipped with two parallel combined fan-duct systems connected in series with another duct system, as shown in Fig. 20.12. The fans in the fan-duct systems are identical and have the same pressure-volume and power-volume characteristics as listed in Example 20.1.

Flow resistances R_1, R_2, and R_3 have the following pressure-volume characteristics:

- R_1 has a Δp_t loss of 2.29 in. WC at a $\dot{V}$ of 10,500 cfm.
- R_2 has a Δp_t loss of 2.82 in. WC at a $\dot{V}$ of 9500 cfm.
- R_3 has a Δp_t loss of 1.50 in. WC at a $\dot{V}$ of 20,000 cfm.

Determine

1. Air volume flow rate flowing through the duct system having flow resistance R_3

2. Operating point of fan-duct systems 1 and 2

3. Fan total efficiency of fans 1 and 2

Solution

1. From the given data, the flow resistances can be calculated as

$$R_1 = \frac{\Delta p_t}{\dot{V}^2} = \frac{2.29}{(10{,}500)^2} = 2.08 \times 10^{-8} \text{ in. WC}/(\text{cfm})^2$$

$$R_2 = \frac{2.82}{(9500)^2} = 3.12 \times 10^{-8} \text{ in. WC}/(\text{cfm})^2$$

$$R_3 = \frac{1.5}{(20{,}000)^2} = 3.75 \times 10^{-9} \text{ in. WC}/(\text{cfm})^2$$

2. From Eq. (20.27), the residual pressure of fan-duct systems 1 and 2 after R_1 and R_2 at a volume flow rate of 2500 cfm can be calculated as

$$\Delta p_{\text{res1}} = F_{t1} - S_1 = \Delta p_{F1} - R_1 \dot{V}_1^2 = 5.0 - 2.08 \times 10^{-8} \times (2500)^2$$
$$= 4.87 \text{ in. WC}$$

$$\Delta p_{\text{res2}} = F_{t2} - S_2 = 5.0 - 3.12 \times 10^{-8} \times (2500)^2 = 4.80 \text{ in. WC}$$

Similarly, the residual pressure of other volume flow rates can be calculated as listed below:

	$\dot{V}$, cfm				
	2500	5000	7500	10,000	12,500
F_t, in. WC	5.0	4.88	4.54	4.0	3.13
$R_1\dot{V}_1^2$, in.WC	0.13	0.52	1.17	2.08	3.25
$R_2\dot{V}_2^2$, in.WC	0.20	0.78	1.76	3.12	4.88
$F_{t1} - S_1$, in. WC	4.87	4.36	3.37	1.92	
$F_{t2} - S_2$, in. WC	4.80	4.10	2.78	0.88	

3. At various volume flow rates flowing through the duct system with flow resistance R_3, the total pressure loss $\Delta p_t = R_3\dot{V}_3^2$ can be calculated as follows:

$\dot{V}$, cfm	5000	10,000	15,000	20,000	25,000
$S_3 = R_3\dot{V}_3^2$, in. WC	0.09	0.38	0.84	1.50	2.34

4. Plot the single-fan curve F_t, the residual pressure curves $F_{t1} - S_1$ and $F_{t2} - S_2$, the combined residual pressure curve $F_{t1} - S_1 + F_{t2} - S_2$, and the system curve S_3 on $\Delta p_t - \dot{V}$ diagram, as shown in Fig. 20.12. The intersection point P of the combined residual pressure curve $F_{t1} - S_1 + F_{t2} - S_2$, and the system curve S_3, gives the volume flow rate in the duct system with flow resistance R_3. From the diagram, $\dot{V}_p = 20{,}000$ cfm (9438 L/s).

5. Draw horizontal line PQ' from point P to intersect the $F_{t2} - S_2$ curve at Q' and $F_{t1} - S_1$ curve at point T'. Again, draw vertical lines $Q'Q$ and $T'T$ from points Q' and T'. The lines intersect fan curve F_t at points Q and T. For fan-duct system 1, operating point T gives

$$\dot{V}_1 = 10{,}500 \text{ cfm}(4955 \text{ L/s}) \qquad \text{and} \qquad \Delta p_{R1} = 3.8 \text{ in. WC } (945 \text{ Pa})$$

For fan-duct system 2, the operating point Q gives

$$\dot{V}_2 = 9500 \text{ cfm}(4480 \text{ L/s}) \qquad \text{and} \qquad \Delta p_{R2} = 4.15 \text{ in. WC } (1032 \text{ Pa})$$

6. From the plotted power-volume curve in Fig. 20.12, for operating point T, the fan power input $P_{f1} = 7.8$ hp, and for operating point Q, the fan power input $P_{f2} = 7.6$ hp. Fan total efficiency of fan 1 is therefore

$$\eta_1 = \frac{\dot{V}_1 \Delta p_{R1}}{6356 P_{f1}} = \frac{10{,}500 \times 3.8}{6356 \times 7.8} = 0.80, \text{ or } 80 \text{ percent}$$

Fan total efficiency for fan 2 is

$$\eta_2 = \frac{9500 \times 4.15}{6356 \times 7.6} = 0.82, \text{ or } 82 \text{ percent}$$

20.8 MODULATION OF THE FAN-DUCT SYSTEM

Modulation Curve

The modulation curve of a fan-duct system or, practically, an air system, is its operating curve when its volume flow rate is modulated at part-load operation. A modulation curve is also the locus of the system operating points at reduced system loads and volume flow rates.

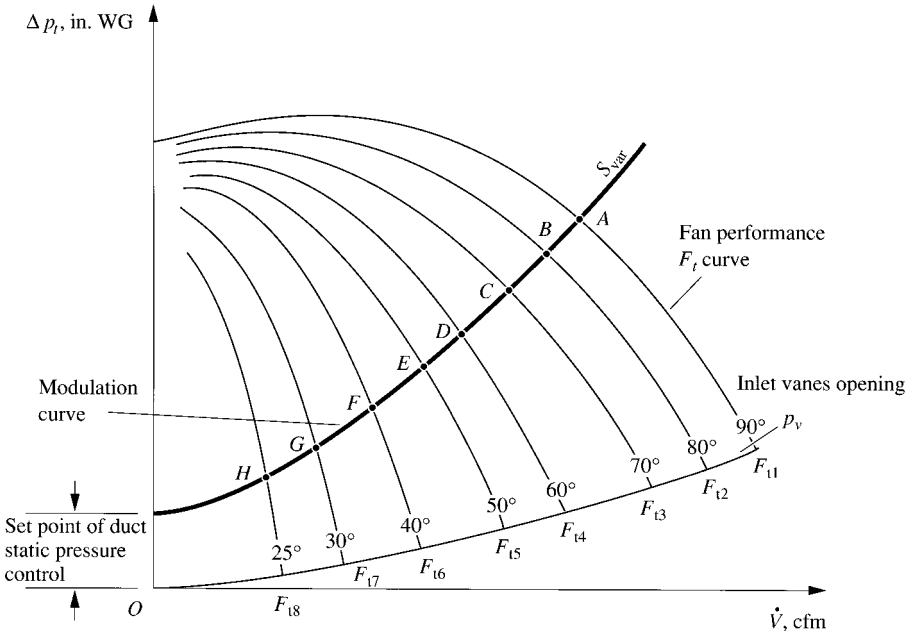


FIGURE 20.13 Modulation curve of a VAV system installed with duct static pressure control.

For a variable-air-volume system with duct static pressure control, the pressure loss of the air system can be divided into fixed part Δp_{fix} and variable part Δp_{var} when the volume flow rate varies. This is described in detail in Chap. 21.

Figure 20.13 shows a typical modulation curve of a VAV system with duct static pressure control using inlet vanes to modulate fan capacity. This VAV system has many fan curves F_t at various volume flow rates and a system curve for the variable pressure loss S_{var}.

During reduction of the volume flow rate of a VAV system, the dampers in the VAV boxes close partially. The variable part of the pressure loss of the air system Δp_{var} from the AHU or PU up to the pressure sensor of duct static pressure control varies as the volume flow is reduced. Its pressure-volume characteristics are indicated by the system curve S_{var}. However, the pressure loss of the duct system, branch takeoffs, and VAV boxes beyond the duct static pressure sensor remains constant, and is equal to the set point of duct static pressure control Δp_{fix}. The modulation curve is actually the system curve of the part of VAV air system with variable pressure loss S_{var}.

At the same time, the inlet vanes at the fan inlet also reduce their opening in response to the sensed higher duct static pressure. Therefore, a new fan curve F_{t2} is formed. The intersection of F_{t2} and system curve S_{var} at the reduced volume flow rate is the system operating point B. Similarly, at a further reduction of volume flow rate, the system operating points are C, D, E . . . .

As the volume flow rate of the air system approaches zero, theoretically, the system total pressure will equal the fixed part of pressure loss Δp_{fix}, which is equal to the set point of duct static pressure control, as shown in Fig. 20.13.

The modulation curve of a fan-duct system should not enter the fan surge area or stall region, or unstable operation will result. However, a forward-curved centrifugal fan may be operated in the surge area if the system total pressure is below 1.5 in. WC. Refer to the manufacturer's data for detailed analysis.

Modulation of Fan-Duct Systems

The performance of a fan-duct system can be modulated either by changing the fan characteristics or by varying the flow resistance of the duct system. This can be achieved by any of the following methods.

Modulation Using Dampers. Use of a damper is the simplest method of modulation. A multiblade damper is usually installed inside the main duct after the centrifugal supply fan. When the damper closes, the system flow resistance increases and the system operating point moves from point A to B on a different system curve. This method is known as "riding on the curve," as shown in Fig. 20.14*a*. If the fan F_t curve is flatter, then the increase in Δp_t during reduction of the volume flow rate of the fan-duct system is small.

Because of the considerable total pressure loss across the damper, power input to the fan is only slightly decreased by this method. Modulation dampers have the lowest installation cost and only small energy savings.

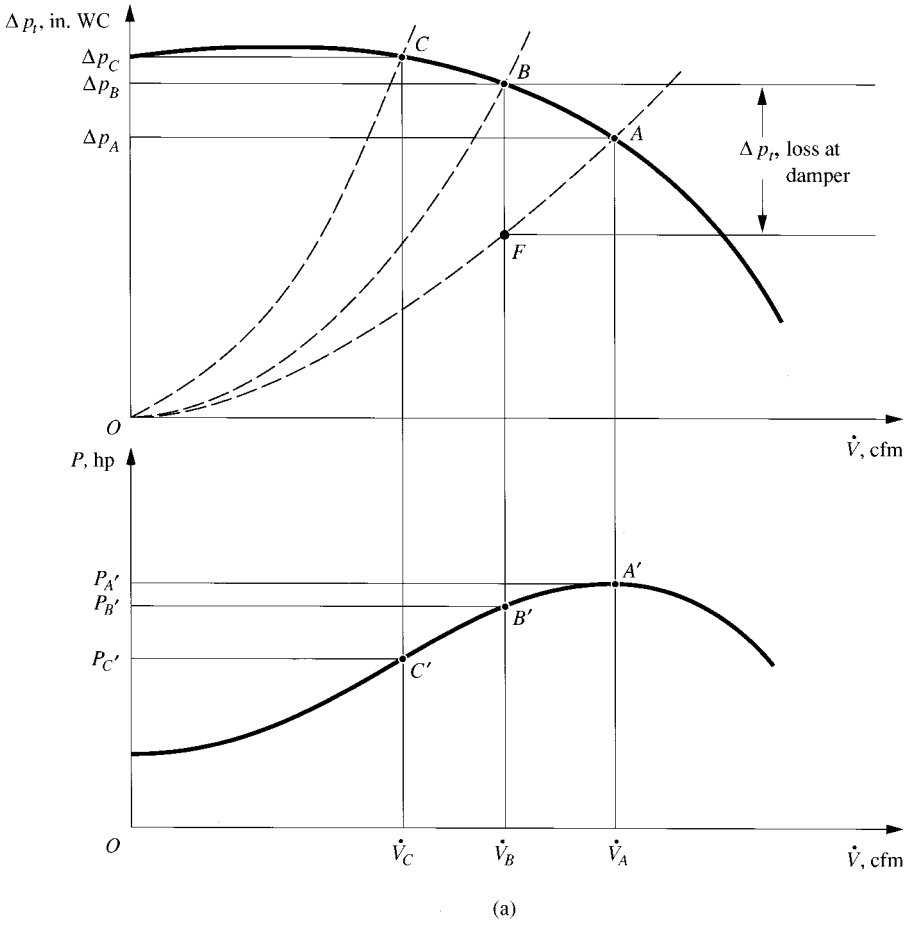


FIGURE 20.14 Modulation of fan-duct systems: (*a*) using dampers; (*b*) using inlet vanes; (*c*) using ac inverter to vary fan speed.

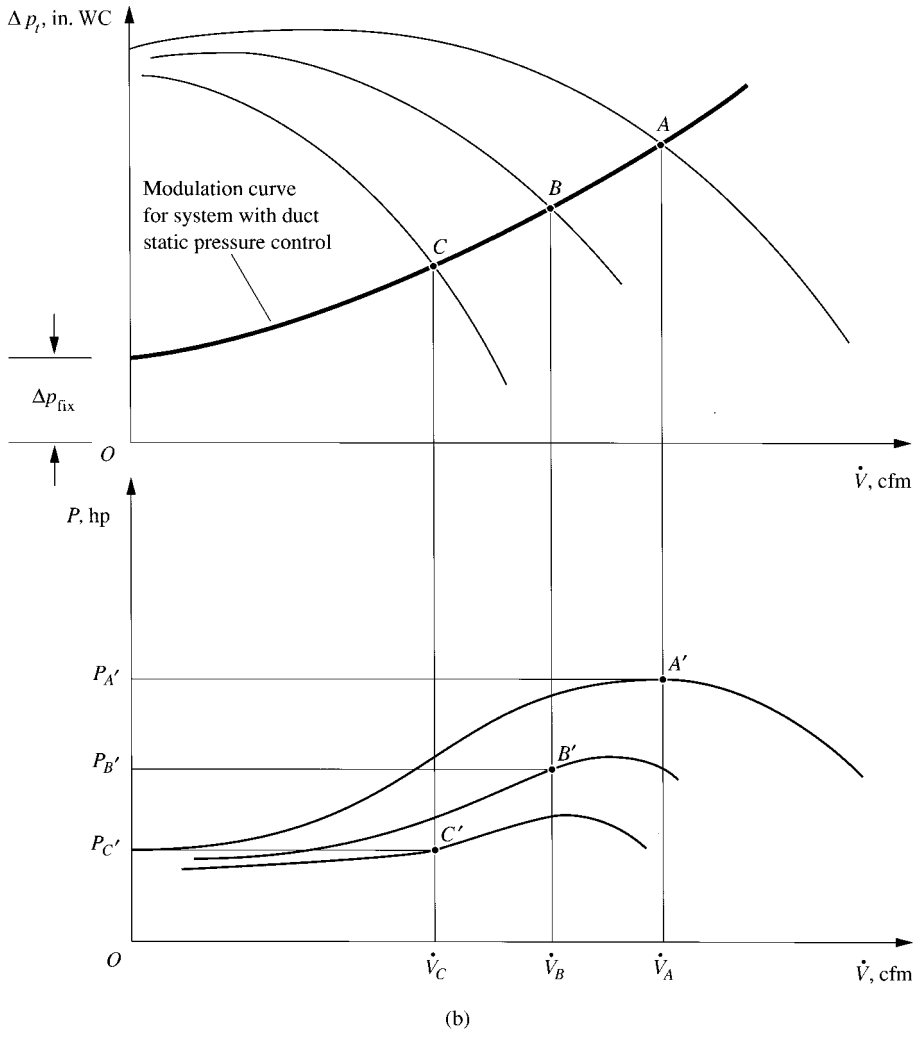
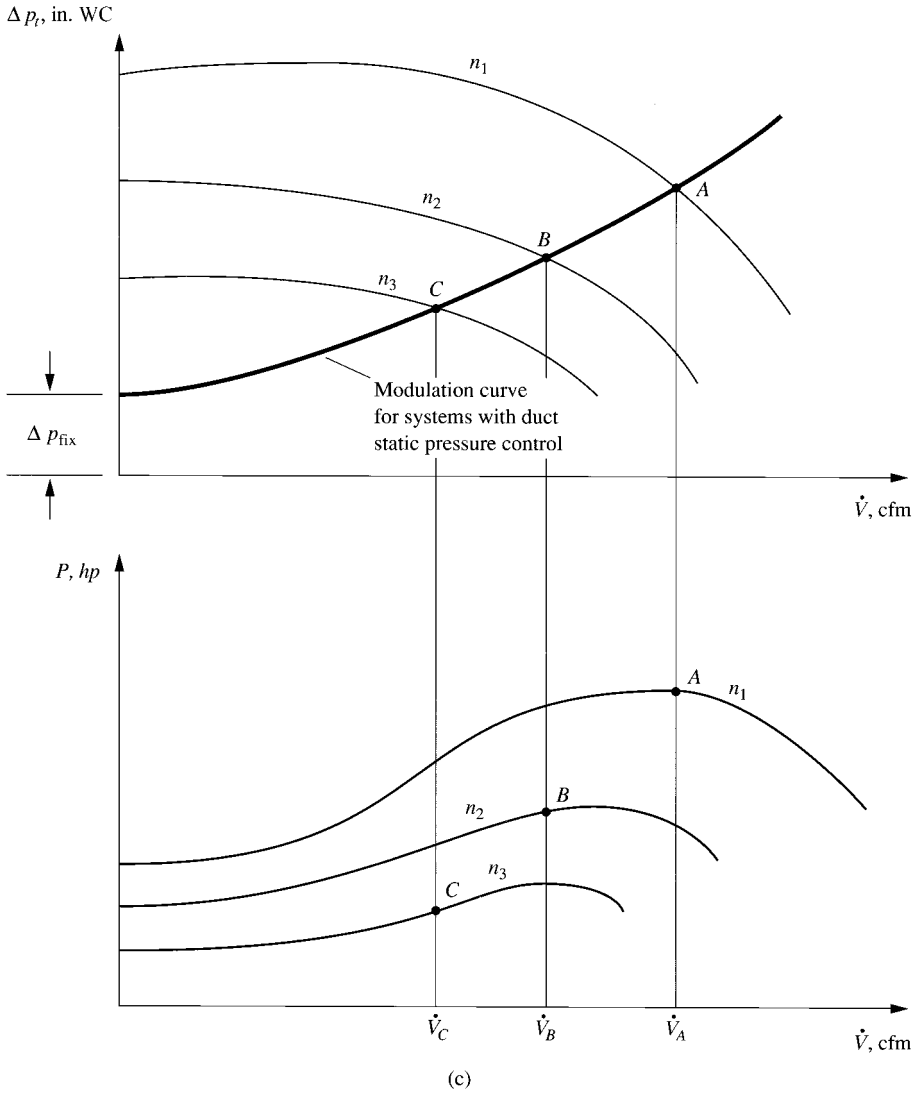


FIGURE 20.14 (*Continued*)

Modulation Using Inlet Vanes. Modulation by varying the opening of inlet vanes at the centrifugal fan inlet gives different fan performance curves and therefore different system operating points. It is widely used in many VAV systems. The surge area of a fan with inlet vanes is smaller than that of a fan without inlet vanes. For a VAV system with duct static pressure control, a modulation curve and the corresponding fan power inputs are shown by points A', B', and C' on the power-volume curves, as shown in Fig. 20.14*b*.

Compared to modulation by damper, modulation using inlet vanes has a moderately higher installation cost and considerable energy savings.

Modulation Using Inlet Cone. Moving the inlet cone of a backward-curved or airfoil centrifugal fan also gives different fan performance curves, so that its modulation curve is similar to that of



(c)

FIGURE 20.14 (*Continued*)

inlet vanes. Compared to modulation by inlet vanes, modulation using inlet cone blocks far less fan inlet area.

Modulation by inlet cone for backward-curved and airfoil centrifugal fans has a comparatively lower installation cost and also considerable energy savings.

Modulation by Blade Pitch Variation of Axial Fan. Modulation of the volume flow of a fan-duct system by blade pitch (blade angle) variation of an axial fan changes the fan characteristics and, therefore, the system operating points. Modulation by automatic blade pitch variation significantly

lowers the fan energy consumption at reduced volume flow rate. Although automatic blade pitch variation is rather expensive, it has become more popular then before.

Modulation by Varying Fan Speed. Modulation of the fan-duct system by varying the fan speed renders new fan performance curves n_2 and n_3 and, therefore, new system operating points *B* and *C* at reduced volume flow rates, as shown in Fig. 20.14*c*. The modulation curve of a VAV system with duct static pressure control can be obtained in the same way as for inlet vanes.

Fan power input for fan speed modulation is lower than for damper or inlet vane modulations. Because of the fan speed reduction at reduced $\dot{V}$, the fan sound power level is reduced accordingly.

Fan speed variation using adjustable-frequency variable-speed drives in a VAV system has become more popular. An adjustable-frequency ac inverter consumes only 7 to 8 percent more energy input to the fan motor. The high initial cost of adjustable-frequency variable-speed drives is often cost-effective for large centrifugal fans. The initial cost may drop further in the future as more and more variable-speed drives are installed.

Because of the differences in fan characteristics, inlet vane, inlet cone, and variable-speed drive modulation is widely used in centrifugal fan-duct systems, and blade-pitch and variable-speed drive modulation is used for axial fan-duct systems.

Example 20.3. A variable-air-volume system equipped with a backward-curved centrifugal fan with airfoil blades operated at 1700 rpm has the following pressure-volume and power-volume characteristics:

$\dot{V}$, cfm	5000	10,000	15,000	20,000	25,000
Δp_t, in. WC	6.0	6.0	6.0	4.65	2.70
Power, hp	10.0	15.2	18.7	21.0	20.0

At design condition, this system has a volume flow rate of 20, 000 cfm (9438 L/s) and a system total pressure loss of 4.65 in. WC (1156 Pa). When it is operated at 50 percent of its design flow rate and its flow rate is modulated by inlet vanes, its performance curves have the following characteristics:

$\dot{V}$, cfm	5000	7500	10,000	12,500	15,000
Δp_t, in. WC	4.88	4.55	4.0	3.12	1.87
P_f, hp	9.6	11.2	12.0	12.3	12.0

Determine the fan power input if the volume flow rate is reduced to 50 percent of its design flow rate by the following methods:

1. Volume control damper
2. Inlet vanes, where system total pressure loss at 50 percent design flow is 4 in. WC
3. Fan speed variation by means of an adjustable-frequency variable-speed drive

Solution

1. From the given data, the flow resistance of this VAV system at design condition is

$$R_d = \frac{\Delta p_t}{\dot{V}^2} = \frac{4.65}{(20{,}000)^2} = 1.16 \times 10^{-8} \text{ in. WC/(cfm)}^2$$

The system total pressure loss at various volume flow rates can then be calculated as

$$S = \Delta p_t = R_d \dot{V}^2 = 1.16 \times 10^{-8}\, \dot{V}^2$$

Plot the fan performance curve F_t and the system curve S. The intersection of these curves is the system operating point P at design condition, as shown in Fig. 20.15*a*.

2. When the volume flow rate is reduced by a damper to 50 percent of its design volume flow, that is, $0.5 \times 20{,}000 = 10{,}000$ cfm (4719 L/s), its operating point must lie on the fan performance curve with a $\dot{V} = 10{,}000$ cfm (4719 L/s), point Q. The fan power input at point Q is 15.2 hp (11.3 kW), as shown in Fig. 20.15*a*.

3. When the volume flow is reduced by modulation of inlet vanes to 10,000 cfm (4719 L/s), the system total pressure loss is 4 in. WC (994 Pa), so the flow resistance of this VAV system at reduced flow using inlet vanes is

$$R_{iv} = \frac{4}{(10{,}000)^2} = 4 \times 10^{-8} \text{ in. WC}/(\text{cfm})^2$$

From the given data, plot the fan performance curve using inlet vanes F_{iv} and the system curve $S_{iv} = R_{iv}\,\dot{V}^2$, as shown in Fig. 20.15*b*. Also plot the power-volume curve. The intersection point P_{iv} of the F_{iv} and S_{iv} curves is the system operating point at 10,000 cfm (4719 L/s) volume flow, and its fan power input is 12 hp (8.95 kW), as shown in Fig. 20.15*b*.

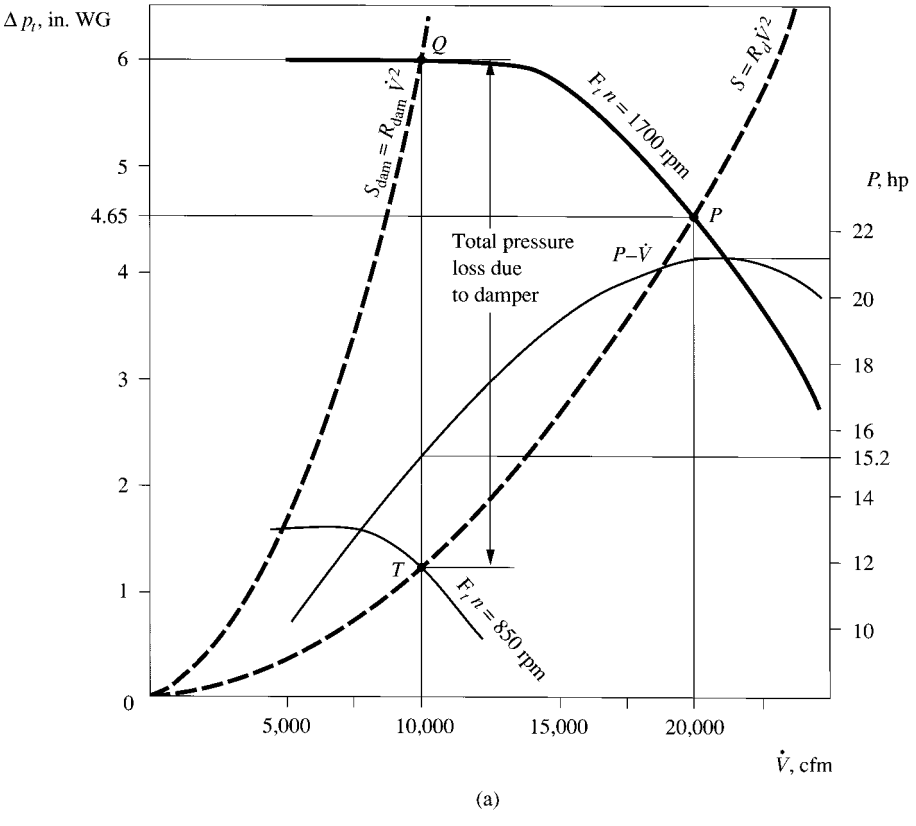


FIGURE 20.15 Modulation of a VAV system using dampers, inlet vanes, and fan-speed variation: (*a*) using dampers and fan-speed variation; (*b*) using inlet vanes.

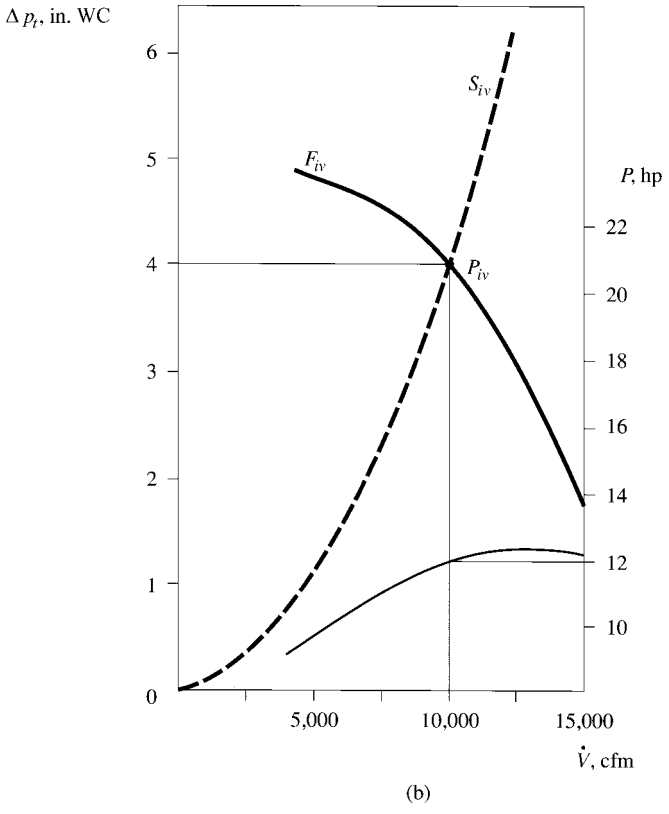


FIGURE 20.15 (*Continued*)

4. For the same fan-duct system, from the fan law, the fan speed required at reduced volume flow of 10,000 cfm (4719 L/s) is

$$n_2 = n_1\left(\frac{\dot{V}_2}{\dot{V}_1}\right) = 1700\left(\frac{10{,}000}{20{,}000}\right) = 850 \text{ rpm}$$

Fan power input at a fan speed $n_2 = 850$ rpm is

$$P_2 = P_1\left(\frac{n_2}{n_1}\right)^3 = 21\left(\frac{10{,}000}{20{,}000}\right)^3 = 2.63 \text{ hp } (1.96 \text{ kW})$$

The difference in fan power inputs between using damper and variable-speed drive fan modulation is the power consumption at the damper: 15.2 − 2.63 = 12.5 hp (9.32 kW). At reduced fan speeds, fan total efficiency may be reduced. Because of the degradation of motor efficiency and ac inverter loss at part-load operations actual power input at 50 percent of part-load operation for variable-speed drive modulation is far higher than 2.63 hp (1.96 kW).

20.9 CLASSIFICATION OF AIR SYSTEMS

Air systems, in a narrower sense, can be classified into the following categories according to their system characteristics:

- Air-handling systems, air handlers, and air distribution systems.
- Supply air systems, return air systems, recirulating air systems, and exhaust air systems. A supply air system may supply conditioned air, outdoor ventilation air, makeup air (either conditioned air or nonconditioned air), or recirculating air. A return air system often returns the conditioned space air to the fanroom or machinery room. A recirculating air system transports the conditioned space air to the AHU, PU, fan-coil unit, or water-source heat pump for conditioning or mixing with outdoor air again. An exhaust system exhausts the contaminated air or space air to the outside atmosphere.
- Single-zone or multizone systems. In a single-zone system, there is no terminal. Space temperature, relative humidity, and volume flow rate are controlled directly by the coils, humidifiers, and inlet vanes or ac inverter in the air-handling unit or packaged unit. In a multizone system, zone temperature or zone supply volume flow rate is controlled by terminals.
- Fan combination systems. Three fan combination systems are often used: supply and exhaust fan combination, supply and relief fan combination, and supply and return fan combination systems.

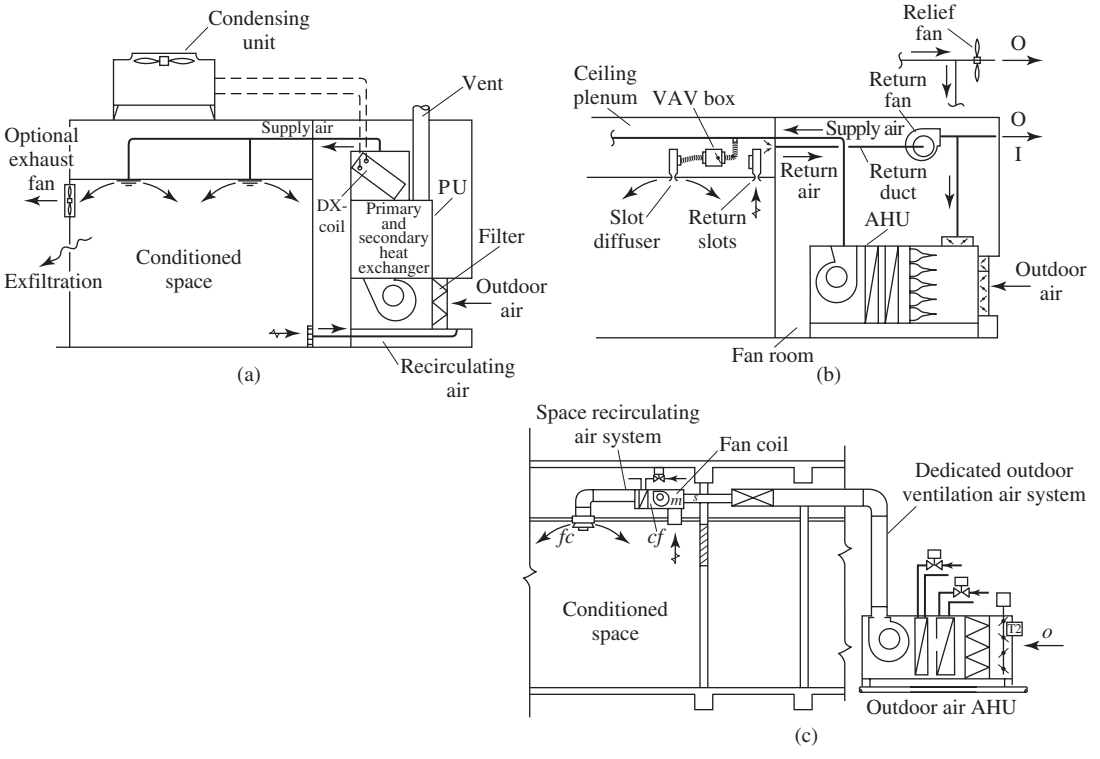


FIGURE 20.16 Various types of air systems: (*a*) constant-volume system; (*b*) variable-volume system; (*c*) dedicated ventilation and space recirculating system.

- Single-duct or dual-duct systems. In a single-duct system, conditioned air is supplied to the conditioned space by a single supply duct. In a dual-duct system, conditioned air is supplied to the conditioned space in the perimeter zone through two supply ducts: a warm air duct and a cold air duct. In the interior zone, only a cold air duct is needed.
- Constant-volume (CV) or variable-air-volume systems. In a constant-volume system, the temperature of supply air is modulated to match the variation of space load during part-load operation, as shown in Fig. 20.16*a*. In a VAV system, the supply volume flow rate is modulated to maintain a predetermined space temperature as space load varies, as shown in Fig. 20.16b.
- Dedicated ventilation and space recirculating system. In a dedicated ventilation and space recirculating system, a required amount of outdoor ventilation air is supplied by a separate ventilation air system; at the same time, there is a parallel space recirculating system to condition the space recirculating air to offset the space heating or cooling load, as shown in Fig. 20.16*c*. Outdoor ventilation air is either mixed with the space recirculating air first or directly supplied to the conditioned space. When the outdoor and recirculating air are mixed in the mixing box (mixed plenum) first; the mixture is then conditioned and supplied to the conditioned space.

Because a constant-volume system is often a single-duct and outdoor recirculating air mixing system, a variable-air volume system is usually also an outdoor recirculating air mixing system, and a space conditioning system always employs a separate ventilation air and a parallel space recirculating air system. Therefore, air systems can be mainly classified into following three primary categories:

- Constant-volume, single-duct outdoor recirculating air mixing systems, or simply constant-volume systems
- Variable-air-volume outdoor recirculating air mixing systems, or simply variable-air-volume systems
- Dedicated ventilation and space recirculating systems

Each category of air system has supply, return, recirculating, or exhaust systems and other specific functions. Dedicated ventilation and space recirculating systems are discussed in Chap. 28.

20.10 CONSTANT-VOLUME SYSTEMS

System Characteristics

A single-zone, single-duct, constant-volume, and outdoor air and recirculating air mixing system is shown in Fig. 20.16*a*. A constant-volume system means that there is a constant volume of supply air throughout the operating period. Supply air temperature is varied when the space load reduces in part-load operation. Single-zone states indicates that the system serves a conditioned space which is controlled to maintain a unique indoor temperature, relative humidity, cleanliness, and pressure differential.

Single-zone constant-volume systems have been widely used in residential buildings and small retail stores in commercial buildings. They often employ a packaged unit with a DX coil instead of water cooling coil in an air-handling unit. In the PU, there is often a single supply fan, and the conditioned space is generally positively pressurized at 0.005 to 0.03 in. WC (1.25 to 7.5 Pa), so that the amount of outdoor ventilation air intake at the packaged unit can be exfiltrated through the cracks and gaps on the building shell; or sometimes a small exhaust fan is installed in the restroom. For a shopping mall or an indoor stadium, an AHU with a return fan is often used. Many air systems that serve clean rooms are single-zone constant-volume systems. These air systems are demanding, complicated, and expensive, and are discussed in Chap. 30. In health care facilities and industrial applications, a single-zone constant-volume system or sometimes a multizone system with reheat is used.

To save energy, most AHU and PU manufacturers also offer air economizers for single-zone constant-volume systems when the supply volume flow rate of the air system is 2000 cfm (1000 L/s) and greater. Most single-zone constant-volume systems employ zone temperature control, night setback, diagnostics, emergency stop, and safety controls. Air economizer and air system controls are discussed in the chapters that follow.

Energy per Unit Volume Flow

As specified in ASHRAE/IESNA Standard 90.1-1999 and discussed in Sec. 17.2, the fan power input to the fan motors per unit volume flow $P_{sy}/\dot{V}_{sd}$ in hp/cfm (W·s/L), for constant-volume air systems as calculated by Eq. (17.24*a*):

$$\frac{P_{sy}}{\dot{V}_{sd}} \le 0.0012 \text{ hp/cfm } (0.0019 \text{ W}\cdot\text{s/L}) \qquad \text{when } \dot{V}_{sd} < 20{,}000 \text{ cfm } (9440 \text{ L/s})$$
$$\frac{P_{sy}}{\dot{V}_{sd}} \le 0.0011 \text{ hp/cfm } (0.00174 \text{ W}\cdot\text{s/L}) \qquad \text{when } \dot{V}_{sd} \ge 20{,}000 \text{ cfm} \tag{20.29}$$

For variable-air-volume (VAV) systems, the fan power input per unit volume flow can be calculated by Eq. (17.24*b*) as:

$$\frac{P_{sy}}{\dot{V}_{sd}} \le 0.0017 \text{ hp/cfm } (0.0027 \text{ W}\cdot\text{s/L}) \qquad \text{when } \dot{V}_{sd} < 20{,}000 \text{ cfm}$$
$$\frac{P_{sy}}{\dot{V}_{sd}} \le 0.0015 \text{ hp/cfm } (0.0024 \text{ W}\cdot\text{s/L}) \qquad \text{when } \dot{V}_{sd} \ge 20{,}000 \text{ cfm}$$

20.11 AIR CONDITIONING PROCESSES

An air conditioning process determines the change in thermodynamic properties of moist air between the initial and final states of conditioning and also the corresponding energy and mass transfer between the moist air and a medium, such as water, refrigerant, or moist air itself during this change. The energy balance and the conservation of mass in nonnuclear processes are the two principles most often used in the analysis and calculation of the change of thermodynamic properties in air conditioning processes.

In general, for a single air conditioning process, heat transfer or mass transfer is positive. However, for calculations that involve several air conditioning processes, the heat supplied to the moist air is taken to be positive, and the heat rejected from the moist air is taken to be negative.

Sensible Heat Ratio

The sensible heat ratio (SHR) of an air conditioning process is defined as the ratio of the absolute value of sensible heat to the absolute value of total heat, or

$$\text{SHR} = \frac{|q_{\text{sen}}|}{|q_{\text{total}}|} = \frac{|q_{\text{sen}}|}{|q_{\text{sen}}| + |q_l|} \tag{20.30}$$

In Eq. (20.30), total heat, in Btu/h (W), is given as

$$q_{\text{total}} = |q_{\text{sen}}| + |q_l|$$

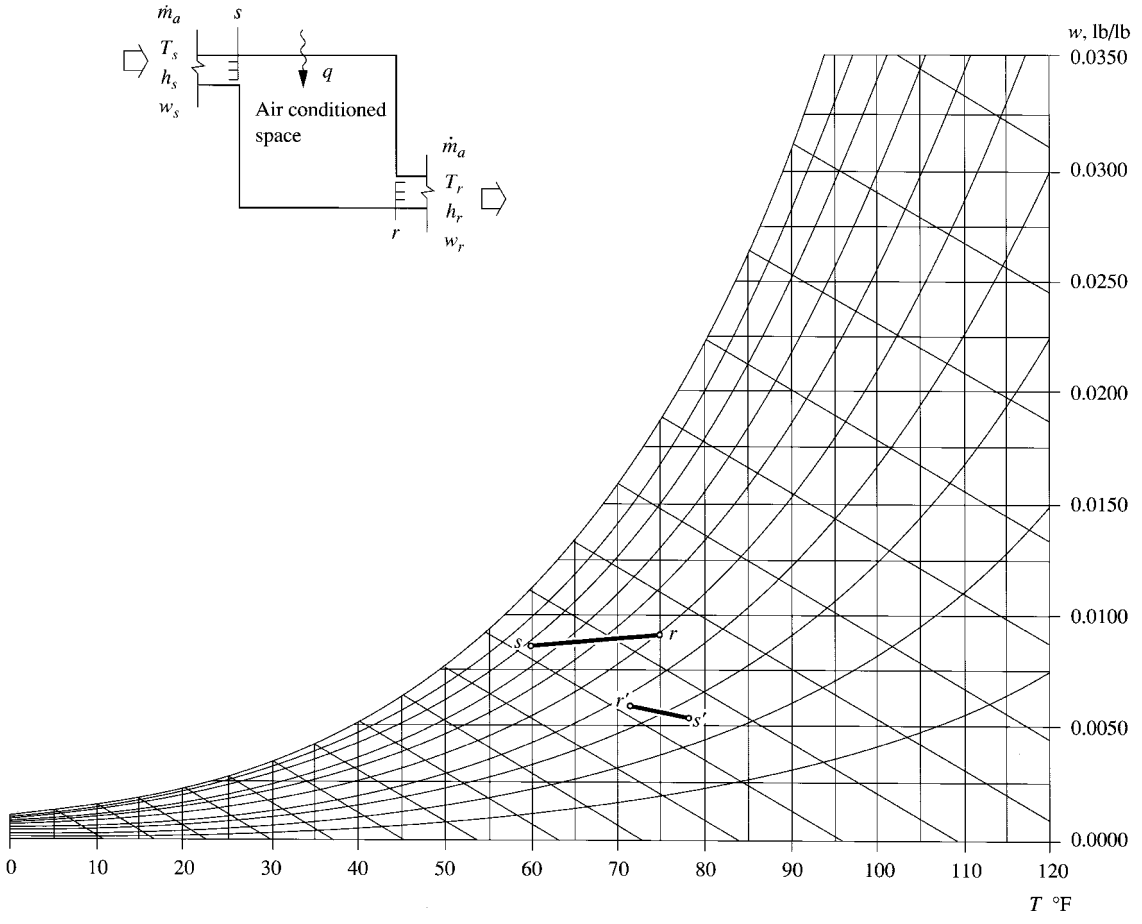


FIGURE 20.17 Space conditioning lines.

Figure 20.17 shows an air conditioning process from an initial state s to final state r. Refer to Secs. 2.9, 2.10, and Appendix Table B.4 for the SI unit conversion of the parameters on the psychrometric chart in Fig. 20.17 as well as other figures.

The sensible heat change q_{sen} in this process, in Btu/h (W), can be calculated as

$$q_{\text{sen}} = 60\dot{V}_s\rho_s c_{pa}(T_r - T_s) = 60\dot{m}_a c_{pa}(T_r - T_s) \tag{20.31}$$

where $\dot{V}_s$ = volume flow rate of supply air, cfm [m^3/(60 s)]
$\dot{m}_a$ = mass flow rate of supply air, lb/min [kg/(60 s)]
ρ_s = density of supply air, lb/ft^3(kg/m^3)
c_{pa} = specific heat of moist air, Btu/lb · °F (J/kg · °C)
T_s,T_r = moist air temperature at initial and final states, °F (°C)

Latent heat change q_l in this process (in Btu/h or W) is given by

$$q_l = 60\dot{V}_s\rho_s(w_r - w_s)h_{fg}$$

where w_s, w_r = humidity ratio at the initial and final states, lb/lb (kg/kg). Theoretically, h_{fg}, in Btu/lb (J/kg), is the latent heat of vaporization of water at the temperature where vaporization or condensation occurs. But because the sensible heat change of $\dot{m}_a c_{pa}(w_r - w_s)(T_r - T_s)$ is already included in Eq. (20.31), it is convenient to use $h_{fg,32} + c_{ps}(T - 32)$ to replace h_{fg}. The error is usually less than 0.03 percent. Then the latent heat change q_l can be calculated as

$$\begin{aligned} q_l &= 60\dot{V}_s\rho_s(w_r - w_s)h_{fg,32} = 60 \times 1075\dot{V}_s\rho_s(w_r - w_s) \\ &= 60\dot{m}_a(w_r - w_s)\,(1075) = 64{,}500\dot{V}_s\rho_s(w_r - w_s) \end{aligned} \tag{20.32}$$

In Fig. 20.17, the slope of the air conditioning process *sr* is given by

$$\tan\alpha = \frac{L_w}{L_t} = \frac{C_t(w_r - w_s)}{C_w(T_r - T_s)} \tag{20.33}$$

where α = angle between air conditioning process and horizontal line on psychrometric chart, deg
L_w, L_t = vertical and horizontal distance between state points *r* and *s*, ft (m)
C_w, C_t = scale factor for humidity ratio lines and temperature lines, lb/lb · ft (kg/kg · m) and °F/ft (°C/m)

Substituting Eqs. (20.31) and (20.32) into Eq. (20.30) gives

$$\begin{aligned} \text{SHR} &= \frac{\dot{m}_a c_{pa}(T_r - T_s)}{\dot{V}_a\rho_a c_{pa}(T_r - T_s) + \dot{m}_a(w_r - w_s)h_{fg,32}} \\ &= \frac{1}{1 + (h_{fg,32}C_w \tan\alpha)/(c_{pa}C_t)} \end{aligned} \tag{20.34}$$

In ASHRAE's psychrometric chart, as shown in App. Fig. B.1, additional SHR lines are given by joining the intersection of $T = 78°\text{F}$, $\varphi = 50$ percent, and the outer sensible heat ratio scale. An air conditioning process that is parallel to any one of these SHR lines has the same SHR.

20.12 SPACE CONDITIONING AND SENSIBLE COOLING AND HEATING PROCESSES

Space Conditioning Process

A space conditioning process is an air conditioning process in which either:

1. Heat and moisture are absorbed by the supply air and removed from the space, *or*
2. Heat, or sometimes heat and moisture, is supplied to the space to compensate for the transmission and infiltration losses through the building shell, with moisture being given up by the supply air to the space. The purpose of these processes is to maintain a desirable space temperature and relative humidity.

Figure 20.17 shows two lines, *sr* and *s′r′*, to indicate these two space conditioning processes. These lines are called space conditioning lines. The upper line, *sr* denotes the absorbtion of space heat and moisture during summer, and the bottom line indicates the supply of heat and absorption of moisture during winter.

In space conditioning processes, assuming that the kinetic energy difference between the supply inlet *s* and return exit *r* is negligible, and that there is no work being done during these processes,

the steady flow energy equation can be simplified to

$$60\,\dot{m}_a h_s + q_{\text{rc}} = 60\,\dot{m}_a h_r \tag{20.35}$$

where h_r, h_s = enthalpy of space air and supply air, Btu/lb (J/kg)
Q_{rc} = heat to be removed from conditioned space, or space cooling load, Btu/h (W)

Rearranging the terms, then, we can calculate the space cooling load as

$$Q_{rc} = 60\,\dot{m}_a(h_r - h_s) = 60\,\dot{V}_s\rho_s(h_r - h_s) \tag{20.36}$$

where $\dot{V}_s$ = volume flow rate of supply air, cfm [m^3/(60 s)]
ρ_s = density of supply air, lb/ft^3 (kg/m^3)

Considering the mass balance during the space conditioning process

$$\dot{m}_a w_s + \dot{m}_g = \dot{m}_a w_r$$

where w_r, w_s = humidity ratio at exit r and inlet s, lb/lb (kg/kg)
$\dot{m}_g$ = rate of moisture gain in conditioned space, lb/min [kg/(60 s)]

Again, by rearranging the terms, the rate of space moisture gain can be evaluated as

$$\dot{m}_g = \dot{m}_a(w_r - w_s) = \dot{V}_s\rho_s(w_r - w_s) \tag{20.37}$$

Sensible Heating and Cooling Processes

A sensible heating process is a process in which heat is added to the moist air, resulting in an increase in its temperature, while its humidity ratio remains constant, as discussed in Sec. 15.9. This process is represented by line *el* in Fig. 15.29. The sensible heating process occurs when moist air flows through a heating coil in which heat is transferred from the hot water inside the coil tubes to the moist air, or through a heat exchanger where heat transfer takes place between two fluid streams. The rate of heat transfer from the hot fluid to the cold fluid Q_{ch}, in Btu/h (W), is called the heating coil load or the heating capacity of the heat exchanger. From Eq. (15.24) the heating coil load can be evaluated as

$$Q_{ch} = 60\dot{m}_a(h_l - h_e) = 60\dot{V}_s\rho_s(h_l - h_e) \tag{20.38}$$

where h_e,h_l = enthalpy of moist air entering and leaving coil (or heat exchanger), Btu/lb (J/kg). From Eq. (2.29),

$$h = c_{pd}T + w(h_{g0} + c_{ps}T)$$

and from Eq. (2.36)

$$c_{pa} = c_{pd} + wc_{ps}$$

Substituting Eq. (2.29) into Eq. (20.38), assuming that the differences between c_{pd1} and c_{pd2}, and c_{ps1} and c_{ps2}, are negligible, and that for a sensible heating process the humidity ratio of moist air entering is equal to leaving air $w_e = w_l$,

$$Q_{ch} = 60\dot{m}_a c_{pa}(T_l - T_e) = 60\dot{V}_s\rho_s c_{pa}(T_1 - T_2) \tag{20.39}$$

A sensible cooling process removes heat from the moist air, resulting in a drop in its temperature while maintaining a constant humidity ratio of the moist air, as discussed in Sec. 15.9. This process is represented by line $e'\ l'$ in Fig. 15.29. The sensible cooling process occurs when moist air flows through a cooling coil and there is an indirect heat transfer from the moist air to the chilled water inside the cooling coil; or in an air-to-air heat exchanger where two airstreams do not make contact.

In a sensible cooling process, the rate of heat transfer Q_{cs}, in Btu/h (W), from the moist air to the chilled water inside the cooling coil or from one airstream to another airstream in the heat

exchanger is called the sensible cooling coil load or sensible cooling capacity of the heat exchanger. It can be calculated as

$$Q_{cs} = 60\dot{m}_a c_{pa}(T_{e'} - T_{l'}) = 60\dot{V}_s \rho_s c_{pa}(T_{e'} - T_{l'}) \tag{20.40}$$

20.13 HUMIDIFYING PROCESSES

In a humidifying process, water vapor is added to the moist air, which increases the humidity ratio if the initial moist air is unsaturated. As discussed in Chap. 15, air is humidified by the following methods:

1. Steam injection or submerged heating element
2. Atomizing devices
3. Wetted media evaporation or spraying air washers

Steam Injection and Heating Element Humidifier

In a humidifying process using steam injection, steam is often supplied from the main line to a grid-type humidifier and then injected directly into the air through small holes, as discussed in Sec. 15.19 and shown in Fig. 20.18. The steam injection humidifying process is indicated by line 12,

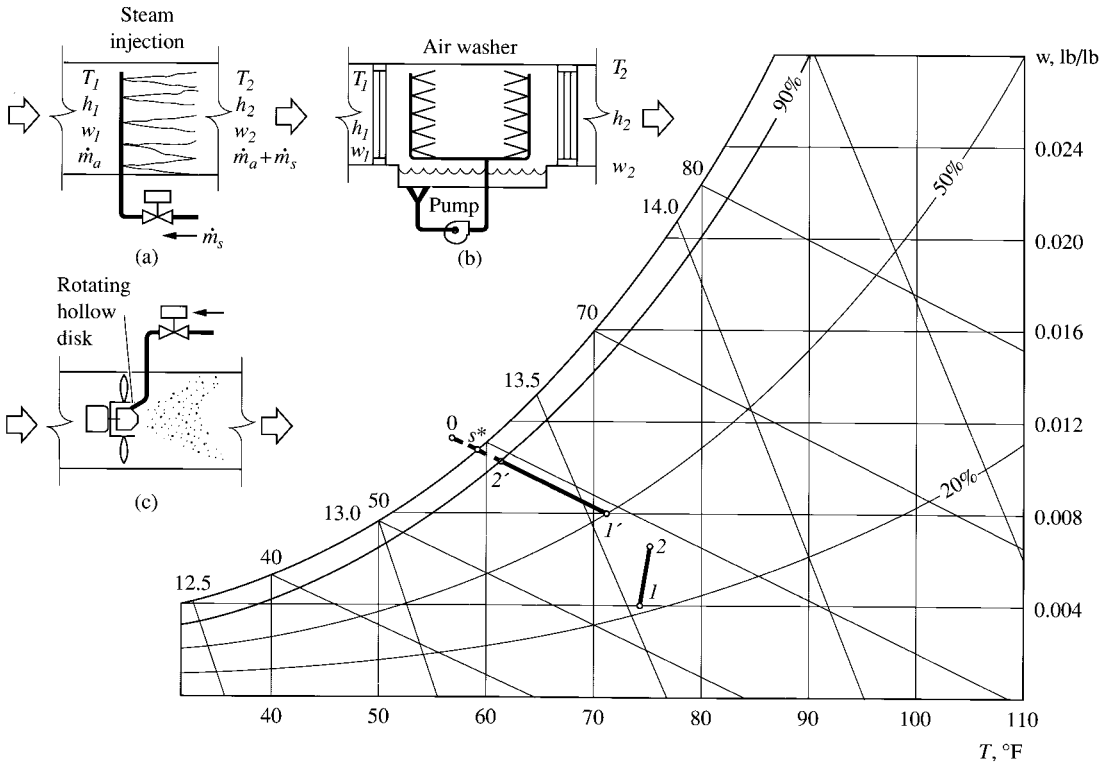


FIGURE 20.18 Humidifying processes.

which is approximately parallel to the constant temperature lines on the psychrometric chart. The slight inclination toward the right-hand side at the top of line 12 is due to the high temperature of the injected steam. The increase in the moist air temperature because of the steam injection can be calculated as follows:

When a mass flow rate of $\dot{m}_s$ lb/h [kg/(60 s)] of dry saturated steam at low pressure is injected into a moist airstream of a mass flow rate of $\dot{m}_a$, lb/h [kg/(60 s)], according to the principle of heat balance

$$\dot{m}_a h_1 + \dot{m}_s h_s = \dot{m}_a h_2 \tag{20.41}$$

where h_1, h_2 = enthalpy of moist air entering and leaving steam injection humidifier, Btu/lb (J/kg)
h_s = enthalpy of injecting steam, Btu/lb (J/kg)

The mass flow rate of injecting steam is defined as

$$\dot{m}_s = \dot{m}_a(w_2 - w_1) \tag{20.42}$$

where w_1, w_2 = humidity ratio of moist air entering and leaving steam injection humidifier, lb/lb (kg/kg). Let $w_{sm} = \dot{m}_s/\dot{m}_a = w_2 - w_1$; then

$$h_1 + w_{sm} h_s = h_2 \tag{20.43}$$

Assuming that c_{pd} and c_{ps} are constants, as $wc_{ps} \ll c_{pd}$, let $w_{12} = (w_1 + w_2)/2$ and replace w_1 and w_2 by w_{12}. Then

$$T_2 - T_1 = \frac{w_{sm} c_{ps} T_s}{c_{pd} + w_{12} c_{ps}} \tag{20.44}$$

where T_s = steam temperature, °F (°C)
T_1, T_2 = temperature of moist air entering and leaving steam injection humidifier, °F (°C)

A heating element humidifier employs a steam coil or electric heating element to provide the latent heat of vaporized water. The saturated water vapor added to the moist air is generally at a temperature higher than that of the airstream.

Air Washer

An air washer is a device that sprays water into air to humidify, to cool and dehumidify, and to clean the air, as discussed in Sec. 15.21. When moist air flows through an air washer sprayed with recirculating water, as shown in Fig. 20.18*b*, or water is distributed over a wetted medium, the moist air is humidified, tending to approach saturation. This actual adiabatic saturation process follows the thermodynamic wet-bulb temperature line on the psychrometric chart represented by line 1′2′ in Fig. 20.18. This process increases the humidity ratio of the airstream while resulting in a reduced air temperature. The cooling effect of the adiabatic saturation process is often called direct evaporative cooling and is covered in greater detail in Chap. 27.

A saturated end state is not usually achieved by such a process. The saturation efficiency ε_{sat} of the moist air leaving the air washer is defined by Eq. (15.54) and discussed in Sec. 15.21.

Oversaturation

In water atomization for air humidification, an atomizing device breaks water into fine particles that are injected into the airstream. Whether the atomization is accomplished by the centrifugal force,

compressed air, or ultrasonic force, the humidification process is an actual adiabatic saturation process. With some atomizing devices, moist air leaving the air washer or pulverizing fan (as shown in Fig. 20.18*c*) can contain unevaporated water particles w_w great enough to exceed the humidity ratio of saturated air at the thermodynamic wet-bulb temperature w_s^*. The excess amount of water particles present in the moist air is called oversaturation. Oversaturation is defined as

$$w_o - w_s^* = w_2' + w_w - w_s^* \tag{20.45}$$

The quantity of unevaporated water particles at state point 2′, in lb/lb (kg/kg) is

$$w_w = w_o - w_2' \tag{20.46}$$

where w_o = sum of humidity ratio w_2' and minute water particles w_w at state point 2′, lb/lb (kg/kg)
w_2' = humidity ratio at state point 2′, lb/lb (kg/kg)

When adiabatic heat transfer occurs between the airstream and the minute water particles, some evaporation takes place, so the humidity ratio of the moist air will increase. Such a transformation still follows the thermodynamic wet-bulb temperature line 1′0, as shown in Fig. 20.18.

As the moist air flows through an air washer or an atomizing device, there can be minute water particles present in the moist air, even if the water eliminators are employed. The magnitude of w_w depends mainly on the construction of the humidifying device and the water eliminators, as well as the air velocity flowing through them. When moist air flows through an air washer, w_w may vary from 0.0002 to 0.001 lb/lb (kg/kg). If a pulverizing fan is used, w_w may be as high as 0.00135 lb/lb (kg/kg).

Oversaturation is beneficial to a space or to a process where humidification is required. However, oversaturation causes wetted surfaces and dampness and often facilitates the growth of mold and fungus, resulting in a serious IAQ problem. Necessary remedies must be taken to eliminate hazardous oversaturation and dampness along the airflow passage.

20.14 COOLING AND DEHUMIDIFYING PROCESS

In a cooling and dehumidifying process, both the temperature and the humidity ratio of the moist air will drop. This process is represented by curve *m-cc* on the psychrometric chart in Fig. 20.19, where *m* is the entering mixture temperature of the outside and recirculating air.

There are three types of heat exchangers commonly used in the cooling and dehumidifying process:

1. Direct-expansion DX coil in which refrigerant evaporates directly inside the coil's tubes, as discussed in Sec. 10.2
2. Water cooling coils with chilled water flowing inside the coil's tubes, as discussed in Sec. 15.10
3. Air washer where chilled water rather than recirculated water is used for spraying, as discussed in Sec. 15.21

The evaporating temperature of the refrigerant T_{ev} in a DX coil and the outer surface temperature of the coil's tubes $T_{s,t}$ are usually lower than the dew point of entering air T_{ae}''. If the temperature of the entering chilled water in a water cooling coil T_{we} and the outer surface temperature of the coil's tubes $T_{s,t}$ are lower than the dew point of entering air T_{ae}'', condensation occurs on the outer surface of the coil tubes on both DX and water cooling coils. Temperature, humidity ratio, and enthalpy of moist air are changed during such a cooling and dehumidifying process.

Based on the principle of heat balance,

$$\begin{matrix}\text{Total enthalpy}\\ \text{of entering air}\end{matrix} = \begin{matrix}\text{total enthalpy}\\ \text{of leaving air}\end{matrix} + \begin{matrix}\text{Cooling coil load (or cool-}\\ \text{ing capacity of washer)}\end{matrix} + \begin{matrix}\text{Heat energy}\\ \text{of condensate}\end{matrix}$$

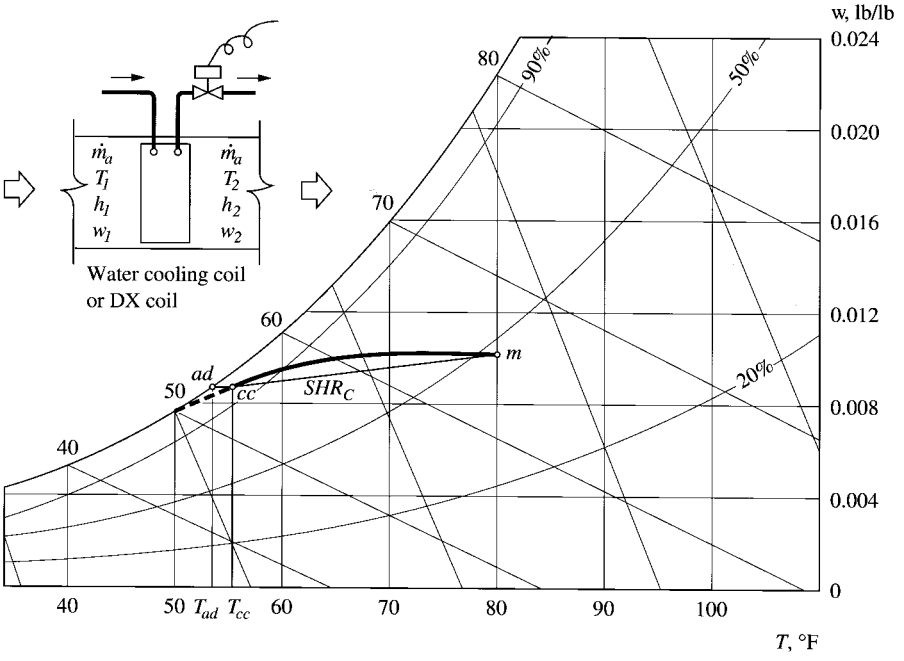


FIGURE 20.19 Cooling and dehumidifying process.

That is,

$$60\dot{m}_a h_{ae} = 60\dot{m}_a h_{cc} + Q_{cc} + 60\dot{m}_w h_w$$

where h_{ae}, h_{cc} = enthalpy of moist air entering and leaving the cooling coil or air washer, Btu/lb (J/kg)
$\dot{m}_w$ = mass flow rate of condensate, lb/min [kg/(60 s)]
h_w = enthalpy of condensate, Btu/lb (J/kg)

The cooling coil load or the cooling capacity of the air washer Q_{cc}, in Btu/h (W), can be calculated as

$$\begin{aligned} Q_{cc} &= 60\dot{m}_a(h_{ae} - h_{cc}) - 60\dot{m}_w h_w \\ &= 60\dot{V}_s\rho_s(h_{ae} - h_{cc}) - 60\dot{m}_w h_w \end{aligned} \tag{20.47}$$

The mass flow rate of the condensate can be evaluated as

$$\dot{m}_w = \dot{m}_a(w_{ae} - w_{cc})$$

where w_{ae}, w_{cc} = humidity ratio of moist air entering and leaving coil or air washer, lb/lb (kg/kg). Assuming that the temperature of the condensate $T_w = T_{cc}$, then we see that Eq. (20.47) becomes

$$\begin{aligned} Q_{cc} &= 60\dot{m}_a[h_{ae} - h_{cc} - (w_{ae} - w_{cc})c_{pw}T_{cc}] \\ &= 60\,\dot{V}_s\rho_s[h_{ae} - h_{cc} - (w_{ae} - w_{cc})c_{pw}T_{cc}] \end{aligned} \tag{20.48a}$$

where c_{pw} = specific heat of water, Btu/lb·°F (J/kg·°C). In most cases, $\dot{m}_w h_w$ is less than

0.02 Q_{cc}. Because $\dot{m}_w h_w$ is small compared to $\dot{m}_a(h_{ae} - h_{cc})$, for practical calculations, $\dot{m}_w h_w$ is often neglected, so

$$Q_{cc} = 60\dot{m}_a(h_{ae} - h_{cc}) = 60\dot{V}_s\rho_s(h_{ae} - h_{cc}) \tag{20.48b}$$

A straight line has been used to represent a cooling and dehumidifying process, and the intersection of this straight line and the saturation curve is considered the effective surface temperature of the coil and is called the apparatus dew point. This is a misconception. First, the cooling and dehumidifying process is actually a curve instead of a straight line. Second, the effective surface temperature as a fictitious reference point is affected by many factors and is difficult to determine in heat- and mass-transfer calculations.

It is more accurate to define the apparatus dew point as the dew point of the moist air leaving the conditioning apparatus, the coil, the air washer, or other heat exchangers. Apparatus dew point is represented by point *ad* on the saturation curve of the psychrometric chart in Fig. 20.19.

However, the sensible heat ratio of the cooling and dehumidifying process SHR_c can be indicated by the slope of the line joining points *m* and *cc*, as shown in Fig. 20.19. SHR_c can be calculated as the ratio of sensible heat removed during cooling and dehumidifying process Q_{cs} to the

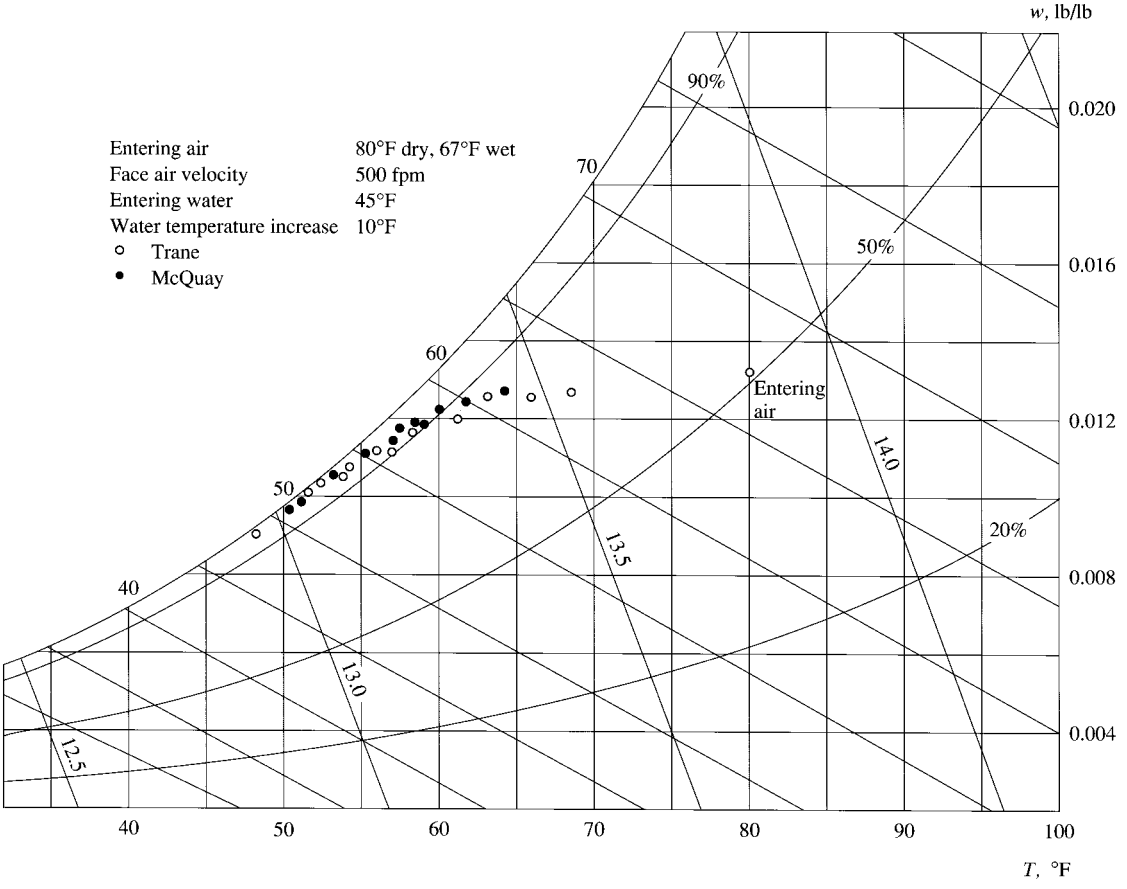


FIGURE 20.20 Cooling and dehumidifying process based on manufacturers' data.

total heat removed Q_{cc}, both in Btu/h (W)

$$\text{SHR}_c = \frac{Q_{cs}}{Q_{cc}} \tag{20.49}$$

According to the data published by U.S. manufacturers, a cooling and dehumidifying curve can be plotted for the moist air flowing through a water cooling coil. Fig. 20.20 shows such a curve. The operating conditions are as follows:

Entering air	80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb temperatures
Face air velocity	500 fpm (2.5 m/s)
Entering water temperature	45°F (7.2°C)
Water temperature rise	10°F (5.6°C)

The number of rows of depth, i.e., the number of coil tubes along the airflow, varies from 2 to 10 rows, and the fin spacing varies from 8 to 18 fins per inch (1.4- to 3.2-mm fin spacing). Both of these factors affect the outer surface area of the coil.

Note that the relative humidity of the air leaving the coil approaches saturation as the outer surface area of the coil increases. The cooling and dehumidifying process for a DX coil is similar to that of the water cooling coil. Therefore, for coils with fin spacing of less than 10 fins per inch and for entering air at 80°F (26.7°C) dry-bulb and 68°F (20°C) wet-bulb temperatures, the relative humidity of the conditioned air leaving the cooling coil may be estimated approximately as follows:

4-row coils	90–95 percent
6-row and 8-row coils	96–98 percent

Many factors affect the final state of the conditioned air leaving the coil. Use manufacturer's data or refer to Secs. 10.2 and 15.10 for a more detailed analysis.

20.15 ADIABATIC MIXING AND BYPASS MIXING PROCESSES

Two-Stream Adiabatic Mixing Process

In a two-stream adiabatic mixing process, two moist airstreams (say, 1 and 2) are mixed together adiabatically, forming a uniform mixture m in a mixing chamber. Such a mixing process is illustrated by line 1-m1-2 on the psychrometric chart in Fig. 20.21. In actual practice, for the mixing of either recirculating and outdoor airstreams or the conditioned airstream and bypass airstream, the rate of heat transfer between the mixing chamber and the ambient air is small. It is usually ignored. Based on the principle of heat balance and the continuation of mass

$$\begin{aligned} \dot{m}_1 h_1 + \dot{m}_2 h_2 &= \dot{m}_m h_m \\ \dot{m}_1 w_1 + \dot{m}_2 w_2 &= \dot{m}_m w_m \\ \dot{m}_1 + \dot{m}_2 &= \dot{m}_m \end{aligned} \tag{20.50}$$

In Eqs. (20.50), subscripts 1, 2, and m represent airstream 1, airstream 2, and the mixture, respectively. Because $\dot{m}_1 = \dot{m}_m - \dot{m}_2$ and $\dot{m}_2 = \dot{m}_m - \dot{m}_1$, substituting into Eq. (20.50) gives

$$\dot{m}_1/\dot{m}_m = \frac{h_2 - h_m}{h_2 - h_1} = \frac{w_2 - w_m}{w_2 - w_1} = K_1$$

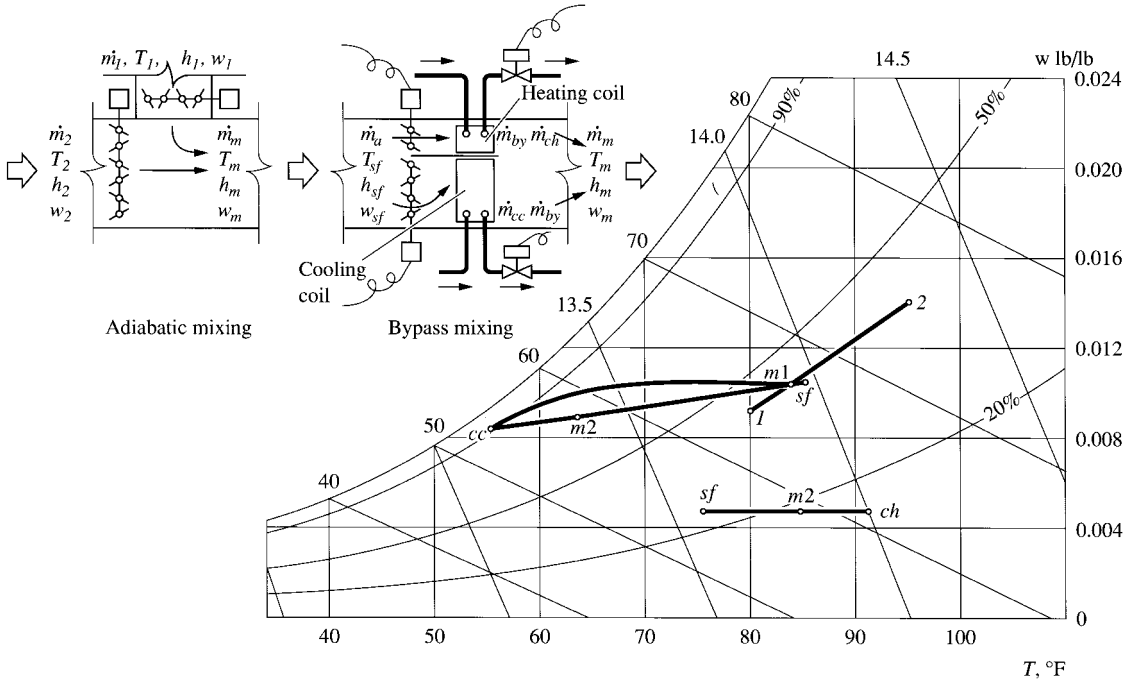


FIGURE 20.21 Adiabatic mixing and bypass mixing processes.

and
$$\dot{m}_2/\dot{m}_m = \frac{h_m - h_1}{h_2 - h_1} = \frac{w_m - w_1}{w_2 - w_1} = K_2 \tag{20.51}$$

where K_1, K_2 = fraction of airstreams 1 and 2 in the mixture.

From the above-mentioned relationships

1. The mixing point m must lie on the line that connects state points 1 and 2.
2. The ratio of the mass flow rate of airstream 1 to the mass flow rate of the mixture is exactly equal to the ratio of segment m1-2 to segment 1-2, as shown in Fig. 20.21, that is,

$$\frac{\dot{m}_1}{\dot{m}_m} = \frac{m1\text{-}2}{1\text{-}2} \tag{20.52}$$

Similarly, the ratio of the mass flow rate of airstream 2 to the mass flow rate of the mixture is exactly equal to the ratio of segment m1-1 to segment 1-2, that is,

$$\frac{\dot{m}_2}{\dot{m}_m} = \frac{m1\text{-}1}{1\text{-}2} \tag{20.53}$$

As the mass flow rate $\dot{m} = \dot{V}\rho$, substituting into Eqs. (20.50) gives

$$\begin{aligned} \dot{V}_1\rho_1 h_1 + \dot{V}_2\rho_2 h_2 &= \dot{V}_m\rho_m h_m \\ \dot{V}_1\rho_1 w_1 + \dot{V}_2\rho_2 w_2 &= \dot{V}_m\rho_m w_m \\ \dot{V}_1\rho_1 + \dot{V}_2\rho_2 &= \dot{V}_m\rho_m \end{aligned} \tag{20.54}$$

Because the magnitude of ρ_m always lies between ρ_1 and ρ_2, if the differences between ρ_m and ρ_1, ρ_m and ρ_2, and the differences in specific heat c_{pa} are negligible, then

$$\begin{aligned} \dot{V}_1 h_1 + \dot{V}_2 h_2 &= \dot{V}_m h_m \\ \dot{V}_1 w_1 + \dot{V}_2 w_2 &= \dot{V}_m w_m \\ \dot{V}_1 T_1 + \dot{V}_2 T_2 &= \dot{V}_m T_m \\ \dot{V}_1 + \dot{V}_2 &= \dot{V}_m \end{aligned} \tag{20.55}$$

Bypass Mixing Process

In many air-handling units, one airstream is divided into an upper hot deck air-stream and a lower cold deck airstream, as shown in Fig. 20.21. During the cooling season, the lower airstream is cooled and dehumidified by the cooling coil, and the hot water supply to the heating coil in upper hot deck is shut off. Therefore, the upper airstream becomes a bypass airstream, i.e., it bypasses the cooling coil. After the coils, the conditioned airstream is mixed with the bypass airstream and forms a mixture at state point *m*2. This bypass mixing process is shown by line *cc*-*m*2-*sf* in Fig. 20.21.

During the heating season, the cooling coil is not energized, but the heating coil is. Under such circumstances, the upper conditioned warm airstream and the lower bypass airstream combine to form a mixture *m*2, as indicated by line *ch*-*m*2-*sf* in Fig. 20.21.

Similar to the two-stream adiabatic mixing process, the ratio of the mass flow rate of the conditioned stream $\dot{m}_{cc}$ to the mixture $\dot{m}_m$ is

$$\frac{\dot{m}_{cc}}{\dot{m}_m} = \frac{sf\text{-}m2}{sf\text{-}cc} = K_{cc}$$

and

$$\frac{\dot{m}_{ch}}{\dot{m}_m} = \frac{sf\text{-}m2}{sf\text{-}ch} = K_{ch} \tag{20.56}$$

where K_{cc}, K_{ch} = fraction of conditioning stream flowing through the cooling coil or heating coil. Also

$$\begin{aligned} \dot{m}_m &= \dot{m}_{cc} + \dot{m}_{by} = \dot{m}_{ch} + \dot{m}_{by} \\ \dot{m}_m w_m &= \dot{m}_{cc} w_{cc} + \dot{m}_{by} = w_{sf} \end{aligned} \tag{20.57}$$

Subscript cc represents the condition off the cooling coil, ch the heating coil, by the bypass airstream, which is the same as the state point of air at the supply fan discharge sf. According to the principle of energy balance

$$\begin{aligned} h_m &= \frac{\dot{m}_{cc} h_{cc} + \dot{m}_{by} h_{sf}}{\dot{m}_m} \\ &= \frac{\dot{m}_{ch} h_{ch} + \dot{m}_{by} h_{sf}}{\dot{m}_m} \\ w_m &= \frac{\dot{m}_{cc} w_{cc} + \dot{m}_{by} w_{sf}}{\dot{m}_m} \end{aligned} \tag{20.58}$$

Based on the mass flow rate of the mixture, the cooling coil load can be calculated as

$$Q_{cc} = K_{cc} V_s \rho_s (h_{sf} - h_{cc}) \tag{20.59}$$

and the heating coil load is

$$Q_{ch} = K_{ch}\dot{V}_s\rho_s c_{pa}(T_{ch} - T_{sf}) \tag{20.60}$$

20.16 SINGLE-ZONE, CONSTANT-VOLUME SYSTEMS—COOLING MODE OPERATION

Air Conditioning Cycle

An air conditioning cycle consists of several air conditioning processes connected in a serial order. Strictly speaking, such processes determine the operating performance of various components of the air system in an air conditioning system including the heat, mass, and other energy transfer and the changes in property of the moist air.

Different air systems are characterized by different types of air conditioning cycles. The psychrometric analysis of an air conditioning cycle is an important tool to determine the properties of moist air at various state points, the volume flow rates, and the capacities of the major components of the air system.

Air conditioning cycles can be divided into two categories: open and closed cycles. In an open air conditioning cycle, the moist air at the end state will not resume its original state. An air conditioning cycle of a 100 percent outdoor air makeup system is an open cycle. In a closed air conditioning cycle, moist air at the end state will resume its original state. The cycle of conditioning the mixture of recirculating and outdoor air, supplying it to the space, maintaining it at a required temperature and relative humidity, and recirculating it again is a closed cycle.

Based on space load characteristics, the air conditioning cycles for constant-volume single-zone systems operate mainly in two modes: cooling mode or heating mode.

1. *Cooling mode.* When a single-zone, constant-volume system is turned on and its space temperature T_r is raised above a predetermined limit, such as 75°F (24.4°C), the control system calls for cooling (cold air supply), and the air economizer and refrigeration will be turned on in sequence to maintain T_r within required limits. The constant-volume system is said to be operated at cooling mode. Its equipment is selected to satisfy the summer outdoor and indoor design conditions.
2. *Heating mode.* When a single-zone, constant-volume system is turned on and its T_r drops below a predetermined limit, such as 70°F (21.1°C), the control system calls for heating (warm air supply) to maintain a required T_r within predetermined limits. The constant-volume system is said to be operated at heating mode. The same equipment is also selected to satisfy winter outdoor and indoor design conditions.

Cooling Mode Operation in Summer

In Fig. 20.22, the upper air conditioning cycle is the summer cooling mode cycle for a constant-volume single-zone system in a factory. Return air from the conditioned space may pass through a return fan. Recirculating air enters the air-handling unit or packaged unit at state point *ru* and is mixed with the outdoor air at point *o* in the mixing box. The mixture *m* is then cooled and dehumidified at the cooling coil and leaves it at point *cc*. After that, the conditioned air flows through the supply fan *sf*, the supply duct, and then supplies to the conditioned space through the supply outlet at state point *s*. When supply air absorbs the sensible and latent loads from the space, it becomes space air at point *r*. Space air then flows through ceiling plenum *rp*, absorbs heat released from the light fixtures and the roof, and flows through the return duct/return fan *rf*. The recirculated portion enters the AHU or PU again at point *ru*.

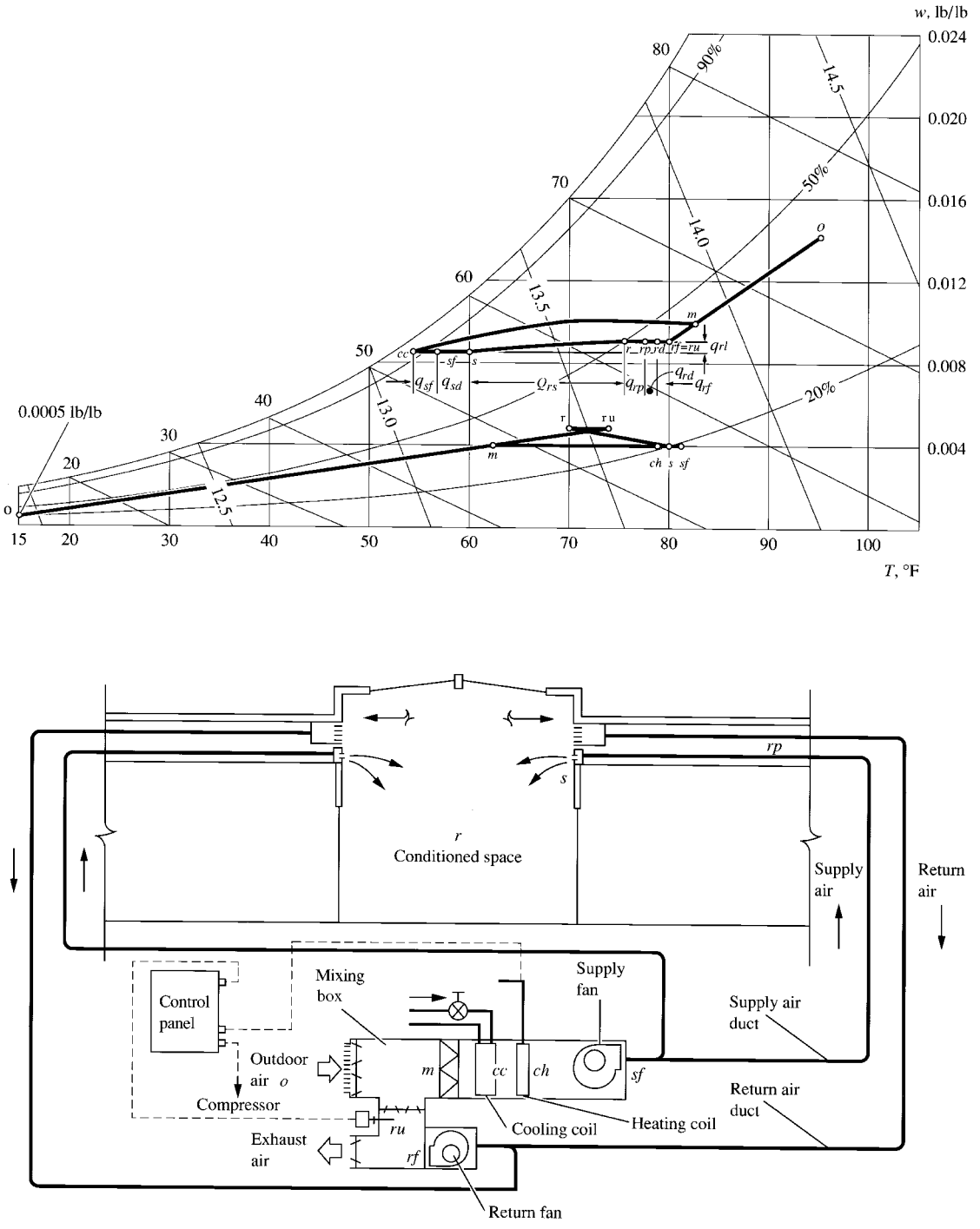


FIGURE 20.22 Summer cooling mode air conditioning cycle for a constant-volume single-zone system in a factory.

Cooling mode operation in summer consists of the following processes:

1. Sensible heating process *r-ru* from the return system heat gain $q_{r,s}$, in Btu/h (W), when return air flows through the ceiling plenum, the return duct, and the return fan. It includes

- Heat gain from the electric lights recessed on the ceiling plenum q_{rp}
- Return fan power heat gain q_{rf}, if there is a return fan
- Return duct heat gain q_{rd}

That is;

$$q_{r,s} = q_{rp} + q_{rf} + q_{rd} \tag{20.61}$$

It is often convenient to use overall temperature rise to denote the system heat gain, such as, $\Delta T_{r,s}$, in °F (°C). Then the return system heat gain is

$$q_{r,s} = 60\dot{V}_{\text{ret}}\rho_{\text{ret}}c_{pa}(T_{ru} - T_r) = 60\dot{V}_{\text{ret}}\rho_{\text{ret}}c_{pa}\Delta T_{r,s} \tag{20.62}$$

Subscript ret indicates return air and *ru* the recirculating air entering the AHU or PU. The volume flow rate of return air is often higher than that of the recirculated air entering the mixing box if there is leakage or exhaust air through the exhaust/relief damper.

2. Adiabatic mixing process of recirculating air at state point *ru* and the outdoor air at state point *o* in the mixing box, which is represented by line *ru-m-o* on the psychrometric chart in Fig. 20.22.

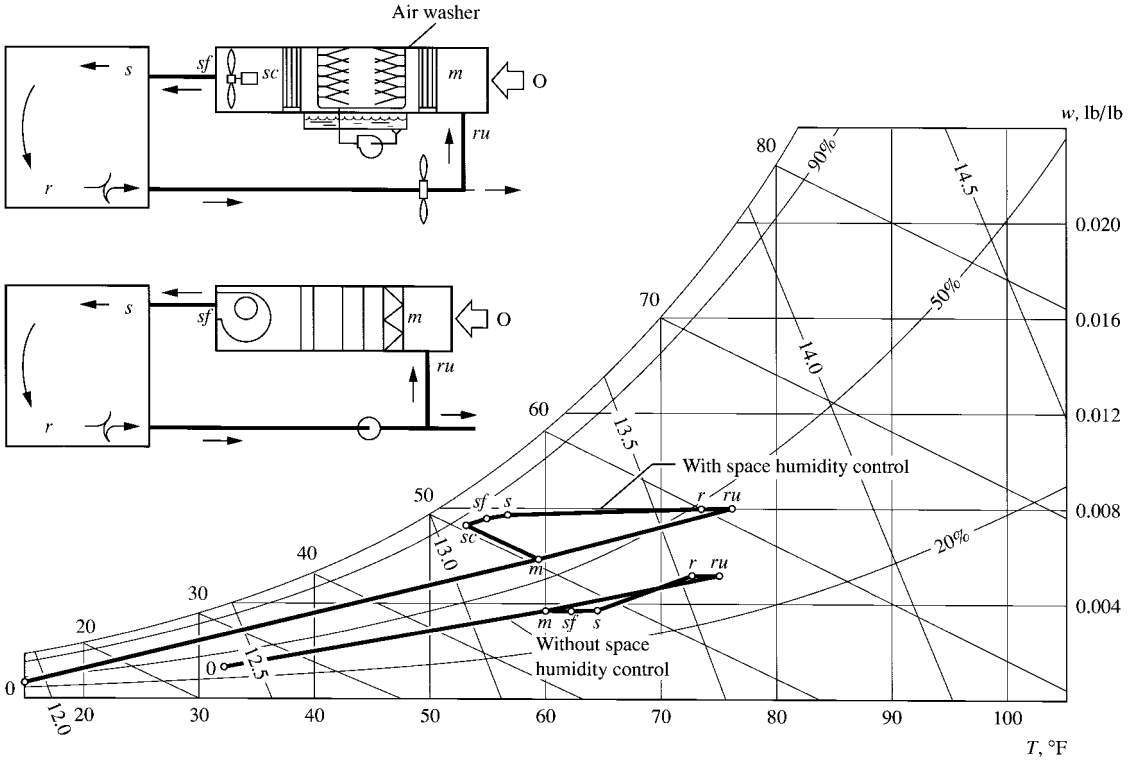


FIGURE 20.23 Air conditioing cycle—winter cooling mode operation for a constant-volume single-zone system.

After mixing, the condition of the mixture is *m*, which is also the state point of the moist air entering the cooling coil.

3. Cooling and dehumidifying process, which is indicated by the curve *m-cc*. The cooling coil load can be calculated by Eq. (20.48*b*).

4. Sensible heating process *cc-s* from the supply system heat gain $q_{s,s}$ when supply air flows through the supply fan and supply duct. It includes

- Supply fan power heat gain q_{sf} represented by line *cc-sf*
- Supply duct heat gain q_{sd} indicated by line *sf-s*.

Supply system heat gain is then given as

$$q_{s,s} = q_{sf} + q_{sd} \tag{20.63}$$

The supply fan power heat gain and the supply duct heat gain are far greater than the heat gains of the return fan and duct. These differences are caused by the higher fan total pressure in the supply fan and the larger temperature difference between the air inside and outside the supply duct.

Like the return system heat gain the supply system heat gain can be represented by an overall temperature rise $\Delta T_{s,s}$, in °F (°C).

5. Space conditioning process, which is represented by line *sr*. From Eqs. (20.31) and (20.36), the sensible heat removed from the conditioned space or the space sensible load Q_{rs}, in Btu/h (W), is

$$Q_{rs} = 60\dot{m}_a c_{pa}(T_r - T_s) = 60\dot{V}_s \rho_s c_{pa}(T_r - T_s) \tag{20.64}$$

From Eq. (20.32), the latent heat removed from the conditioned space or the space latent load $Q_{rl,}$ in Btu/h (W), is

$$Q_{rl} = 60\dot{m}_a(w_r - w_s)h_{fg,32} = 60\dot{V}_s \rho_s(w_r - w_s)h_{fg,32} \tag{20.65}$$

Therefore, the space cooling load Q_{rc}, in Btu/h (W), is calculated as

$$Q_{rc} = Q_{rs} + Q_{rl} \tag{20.66}$$

Cooling Mode Operation in Winter Without Space Humidity Control

In winter, many factories that employ constant-volume single-zone systems still have a space cooling load, need a cold air supply, and operate at cooling mode operation. These systems can be divided into two categories: systems without space humidity control, as shown by the lower air conditioning cycle in Fig. 20.23, and systems with space humidity control, as shown by the upper air conditioning cycle in Fig. 20.23.

The air conditioning cycle of a constant-volume single-zone system without space humidity control operated at cooling mode winter design conditions consists of the following air conditioning processes, and most are similar to those in summer design conditions:

- Sensible heating process *r-ru* from the return system heat gain $q_{r,s}$
- Adiabatic mixing process of recirculating air *ru* and the outdoor air *o* in the mixing box
- Sensible heating process *m-sf-s* from the supply system heat gain $q_{s,s}$
- Space conditioning process *sr*

Most of the constant-volume single-zone systems operated at cooling mode winter design conditions do not need a cooling and dehumidifying process because the mixture of outdoor and

recirculating air m is usually lower than the supply temperature plus the temperature rise due to supply system heat gain. This winter cooling mode free cooling cycle can be constructed as follows:

1. From Eq. (20.65), calculate the humidity ratio difference $w_r - w_s$ from the known space latent load and supply volume flow rate.
2. The volume flow rate of outdoor air $\dot{V}_o$ should be greater than the occupant requirements, and sufficient to offset the space cooling load during winter without refrigeration. Based on the same supply volume flow rate as in summer, use Eq. (20.31) to calculate the supply air temperature difference $T_r - T_s$ from the design winter sensible cooling load, and determine T_s.
3. Temperature of the recirculating air entering the AHU or PU T_{ru} can be calculated as

$$T_{ru} = T_r + \text{temperature rise due to return system heat gain}$$

   Mixing temperature of outdoor and recirculating air can be calculated as

$$T_m = T_s - \text{temperature rise due to supply system heat gain}$$

4. Because $w_r = w_{ru}$ and $w_m = w_s$, the ratio

$$\frac{w_{ru} - w_m}{w_{ru} - w_o} = \frac{w_r - w_s}{w_r - w_o} = \frac{T_{ru} - T_m}{T_{ru} - T_o} \tag{20.67}$$

From Eq. (20.67), $w_r - w_o$ and w_r can be determined.

Cooling Mode Operation In Winter with Space Humidity Control

Constant-volume single-zone systems using cold air supply with space humidity control during the winter season have been adopted in many industrial applications, such as spinning and weaving rooms in textile mills. In these workshops, high internal sensible heat loads are released by the spinning machines and weaving looms, requiring cold air supply and a specific controlled relative humidity, even in winter.

The upper air conditioning cycle of Fig. 20.23 shows such a cooling mode cycle in winter with space humidity control using a spray-type air washer for a constant-volume single-zone system. The adiabatic saturation process of evaporative cooling humidification in an air washer saves energy to evaporate the liquid water which is especially suitable for conditioned space that needs a cold air supply in winter. Air washers also cool and dehumidify the air in summer by chilled water and clean the air by water spraying.

The following are the air conditioning processes in this cycle and their characteristics:

1. Process *m-sc* is an adiabatic saturation process. It proceeds along the thermodynamic wet-bulb temperature line on the psychrometric chart.
2. State point *sc* indicates the conditioned air leaving the air washer. Process lines *sc-sf*, *sf-s*, and *s-r* all undergo heating and humidifying processes. Because the airstream leaving the air washer is oversaturated with minute water particles, as the airstream picks heat from the fan, ductwall, and space machine loads, more evaporation occurs. The increase in the humidity ratios in each of these processes varies from 0.0001 to 0.0003 lb/lb (kg/kg). These processes can be determined from both the increase of the enthalpy and the humidity ratios.
3. Because of the high machine load, the sensible heat ratio of the space conditioning line SHR_s often exceeds 0.90.
4. Outdoor air is used for free cooling as well as for occupants and the dilution of air contaminants.

Part-Load Operation and Controls

In a constant-volume single-zone system operated at cooling mode, when the space cooling load reduces, two types of control are used to maintain the required space temperature or other controlled variable during part-load operation:

Water Flow Rate Modulation Control. For an air system employing a water cooling coil in an air-handling unit, the reduction of space sensible cooling load causes a drop in space temperature T_r. When a drop of T_r is sensed, the controller closes the two-way valve of the chilled water supply to maintain the required space temperature. The reduction of chilled water flowing through the coil raises the air temperature and humidity ratio off the coil. The result is a higher space relative humidity φ_{rp} at part-load operation than at design load when the space temperature is maintained at a nearly constant value.

The part-load performance of a water cooling coil, as calculated in Example 15.2, is shown below:

	Design load	Part load
Load ratio	1.0	0.8
Entering air	80°F dry/67°F wet	79°F dry/68°F wet
Leaving air	57.5°F dry/56.5°F wet	61.3°F dry/59.5°F wet
Water velocity	4 fps	2.5 fps
Water temperature rise	10°F (5.6°C)	16°F (8.9°C)
Space temperature	75°F (23.9°C)	75°F (23.9°C)
Space relative humidity	55 percent	59 percent

If the load ratio drops to 0.5, then space relative humidity at part load φ_{rp} may increase to 65 percent.

Two-Position or Cycling Control. For small constant-volume single-zone systems, such as using a packaged unit whose supply volume flow rate is 1500 cfm (708 L/s) or less, on/off control of the refrigeration compressors or the packaged unit is often used to adjust the cooling capacity of the system within an operating cycle of a few minutes to 20 min. Both the air temperature off the coil and the space temperature fluctuate. Because of the heat storage capacity of the air system and the building structure in the conditioned space, the space temperature may fluctuate by 1 to 2°F (0.56 to 1.1°C).

For large packaged units, the fan is continuously operating. Only the refrigeration compressors are cycling in groups, so that the cooling capacity is reduced at part-load operation. At the same time, the amplitude of the fluctuation of the space temperature becomes smaller accordingly.

Outdoor Ventilation Air and Exhaust Fans

Because the supply volume flow is constant year-round for a constant-volume single-zone system only when the filter is loaded with dirt, the fan total pressure increases and the supply volume flow rate decreases accordingly.

Manually Adjusted Dampers. Many small constant-volume single-zone systems for residential and retail stores using a rooftop or split packaged unit with a supply volume flow rate of 1500 cfm (708 L/s) or less employ manually adjusted outdoor air dampers and operate as follows:

- Outdoor damper is field-adjustable to allow preset amount of outdoor air for year-round ventilation purposes.

- Outdoor damper activates with supply fan to preset position and closes when supply fan is shut off.
- Outdoor damper can be preset to admit up to 50 percent of outdoor air.

Air Economizer. For constant-volume systems with a supply volume flow rate of 2000 cfm (944 L/s) or greater, many packaged unit manufacturers offer air economizers to use outdoor air free cooling for energy savings as well as to improve indoor air quality. An air economizer is always used as the first stage of cooling before refrigeration is energized. Air economizers are discussed in Chap. 21.

Exhaust Systems. In most small constant-volume single-zone systems using a PU, part of space air is exfiltrated through the cracks and gaps on the building shell and the opening of entrance doors because of the positive space pressure to balance the amount of outdoor air intake at the PU. Sometimes an additional small exhaust fan is installed on the external window or wall of the restroom.

For a constant-volume system operated in an air economizer cycle that extracts 100 percent outdoor air for free cooling, a return or relief fan must be used to exhaust the space air outdoors.

20.17 SUPPLY VOLUME FLOW RATE

Supply air volume flow rate and the refrigeration capacity are two primary air system characteristics.

Based on Space Cooling and Heating Load

For most air systems including constant-volume systems that are designed to maintain a healthy and comfortable space temperature T_r, the supply air volume flow rate is often calculated from the space cooling load. From Eqs. (20.31) and (20.36), the mass flow rate of the supply air, in lb/min (g/min), can be calculated as

$$\dot{m}_a = \frac{Q_{rc}}{60(h_r - h_s)} = \frac{Q_{rs}}{60c_{pa}(T_r - T_s)} \tag{20.68}$$

The supply volume flow rate, in cfm [m^3/(60 s)], can be calculated as

$$\dot{V}_s = \frac{Q_{rc}}{60\rho_s(h_r - h_s)} = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_s)} \tag{20.69}$$

In Eq. (20.69), T_r is the space temperature and T_s the supply temperature at the outlet, both in °F (°C). For a constant-volume single-zone system operated at cooling mode summer design conditions, T_r is often the required indoor design temperature in summer, and T_s is the representative supply temperature of this single zone, as shown in Fig. 20.22. For a multizone air system for comfort air conditioning, T_r is often the same summer indoor design temperature, and T_s is the supply temperature of the control zone that has the greatest supply system heat gain. In Eq. (20.69), Q_{rs} indicates the design sensible cooling load, in Btu/h (W). For a constant volume system Q_{rs} is the peak cooling load of that specific area which the air system serves because $\dot{V}_s$ is not reduced at part-load operation. For a VAV system, Q_{rs} is the block cooling load.

If Q_{rs} and T_r and T_s are properly selected, Eq. (20.68) can be applied to both single-zone and multizone as well as constant-volume and variable-air-volume air systems. The supply density $\dot{V}_{s,d}$, in cfm/ft^2 (L/s · m^2) is defined as

$$\dot{V}_{s,d} = \frac{\dot{V}_s}{A_{\text{fl}}}$$

where A_{fl} = floor area of the conditioned space, ft^2 (m^2). Supply air density is often used in air volume flow estimates.

The following items should be considered during the calculation of the supply air flow rate from Eqs. (20.68) and (20.69):

1. *Volumetric versus mass flow rate*. The supply volume flow rate $\dot{V}_s$ is widely used in calculations to determine the size of fans, grilles, outlets, air-handling units, and packaged units. However, mass flow rate $\dot{m}_a$ is sometimes simpler to apply in cooling coil load calculations, or it may be more appropriate when the variation of air density affects the result.

2. *Temperature difference versus enthalpy difference*. Theoretically, in Eq. (20.69), it is more accurate to use the enthalpy difference to calculate the supply volume flow rate. This is due to the elimination of the term c_{pa}, which is actually a variable depending on the temperature and humidity ratio. In actual practice, it is more convenient to use temperature differences instead of enthalpy difference because usually the space temperature is controlled and maintained at a preset value. The variation of c_{pa} is often not significant enough to affect the calculated results.

3. *Summer cooling load versus winter heating load*. Summer cooling load is usually used to calculate the supply volume flow rate because it is generally greater than using the winter heating load. But if the winter weather is very cold, this may require a greater volume flow rate. Based on winter heating load, the supply volume flow rate can be calculated as

$$\dot{V}_s = \frac{Q_{rh}}{\rho_s c_{pa}(T_s - T_r)} \tag{20.70}$$

where Q_{rh} = space heating load, Btu/h (W). Use the greater volume flow rate of the two to select the fans, AHU or PU, and other components.

Based on Requirements Other Than Cooling Load

The supply volume flow rate of many air conditioning systems can also be governed by the following requirements:

1. *To dilute the concentration of the air contaminants*. Based on the principle of conservation of mass, the mass generated rate of the air contaminants in the conditioned space $\dot{m}_{par}$, is given by

$$\dot{m}_{par} = \frac{\dot{V}_s(C_i - C_s)}{2118}$$

And the supply volume flow rate $\dot{V}_s$, in cfm (m^3/s), can be calculated as

$$\dot{V}_s = \frac{2118\,\dot{m}_{par}}{C_i - C_s} \tag{20.71}$$

where C_i, C_s = concentrations of air contaminants of space air and supply air, usually in mg/m^3;
1 mg = 0.01543 gr
$\dot{m}_{par}$ = rate of contaminants generated in space, mg/s

2. *To maintain a required relative humidity φ_r as well as a humidity ratio w_r at a specific temperature for the conditioned space.* Then the supply volume flow rate $\dot{V}_s$, in cfm [m^3/(60 s)], can be calculated as

$$\dot{V}_s = \frac{Q_{rl}}{60\rho_s(w_r - w_s)h_{fg,32}} \tag{20.72}$$

3. *To provide a desirable air velocity v_r, in fpm (m/s), within the working area in a clean room.* The supply volume flow rate $\dot{V}_s$, in cfm (m^3/s), is then

$$\dot{V}_s = A_r v_r \tag{20.73}$$

where A_r = cross-sectional area perpendicular to airflow in working area, ft^2 (m^2).

4. *To exceed the outdoor air requirement for acceptable air quality for occupants*. The supply volume flow rate can then be calculated as

$$\dot{V}_s = n\dot{V}_{oc} \tag{20.74}$$

where n = number of occupants
$\dot{V}_{oc}$ = outdoor air requirement per person, cfm (m^3/s)

5. *To exceed the sum of the volume flow rates of the exhaust air $\dot{V}_{ex}$ and the space air relief $\dot{V}_{exf}$* through exfiltration to maintain a positive pressure in the conditioned space, both in cfm (m^3/s). Then the supply volume flow rate is

$$\dot{V}_s = \dot{V}_{\text{ex}} + \dot{V}_{\text{exf}} \tag{20.75}$$

The supply volume flow rate should be the one that is the greatest from any of the above requirements, the space cooling load, or heating load, or outdoor air requirements.

Rated Volume Flow of Supply and Return Fans

The supply volume flow rate $\dot{V}_s$ is determined based on the mass flow rate of supply air $\dot{m}_a$ required to offset the space load at the specific supply condition, with a specific supply air density ρ_s. Because the supply volume flow rate must be provided by the supply fan, or the fan's rated volumetric capacity $\dot{V}_{\text{sf},r}$ against a certain fan total pressure Δp_t, it is important to determine the relationship between $\dot{V}_s$ and $\dot{V}_{\text{sf},r}$.

If the supply fan is located downstream from the cooling coil, the humidity ratios of the moist air between state points *cc* and *s* are constant, as shown in Fig. 20.22. According to the principle of conservation of mass, the mass flow rate of supply air $\dot{m}_a$, in lb/min (kg/min), remains constant, but the volume flow changes as its state changes. The relationships are

$$\dot{m}_a = \dot{V}_{\text{cc}}\rho_{cc} = \dot{V}_{\text{sf}}\rho_{\text{sf}} = \dot{V}_s\rho_s \tag{20.76}$$

Subscript *cc* represents the state point of conditioned air leaving the cooling coil, sf represents the moist air at the outlet of the supply fan, and *s* represents the supply air at the outlets.

Fans are rated at standard air condition. Standard air means dry air at a temperature of 70°F (21.1°C) and a barometric pressure of 29.92 in. (760 mm) Hg (14.697 psi) with a density of 0.075 lb/ft^3 (1.20 kg/m^3). From Eqs. (20.76) and (20.68), the volume flow rate at the supply fan outlet $\dot{V}_{\text{sf}}$, in cfm (m^3/s), is calculated as

$$\dot{V}_{\text{sf}} = \frac{\dot{V}_s\rho_s}{\rho_{\text{sf}}} = \frac{Q_{rs}}{60\rho_{\text{sf}}c_{pa}(T_r - T_s)} \tag{20.77}$$

For the same fan at a given speed, its volume flow rate remains unchanged against a fixed Δp_t even if the air density varies during rated conditions. Therefore,

$$\dot{V}_{\text{sf},r} = \dot{V}_{\text{sf}} \tag{20.78}$$

where $\dot{V}_{\text{sf},r}$ = rated volume flow rate, cfm (m^3/s). From Eq. (2.32), the moist volume of 1 lb of moist air v is

$$v = \frac{R_a T_R(1 + 1.6078w)}{p_{\text{at}}}$$

For normal summer and winter comfort applications, the humidity ratio varies less than 0.006 lb/lb (kg/kg). Therefore, $v < 1.01R_aT_R/p_{at}$, that is, the influence of variation of humidity ratio is smaller than 1 percent. Therefore, it can be ignored. If there is no infiltration and exfiltration at the conditioned space, Eq. (20.76) can be rewritten as

$$\dot{m}_a = \dot{V}_{cc}\rho_{cc} = \dot{V}_{sf}\rho_{sf} = \dot{V}_s\rho_s = \dot{V}_r\rho_r = \dot{V}_{rf}\rho_{rf} \tag{20.79}$$

Subscript rf represents the outlet of the return fan.

Similarly, the rated volume flow rate of the return fan $\dot{V}_{rf,r}$ is given by

$$\dot{V}_{rf,r} = \dot{V}_{rf} = \frac{Q_{rs}}{60\rho_{rf}c_{pa}(T_r - T_s)} \tag{20.80}$$

For air temperature leaving the cooling coil T_{cc} at 55°F (12.8°C) with a relative humidity of 92 percent and T_{sf} at 57°F (13.9°C), the psychrometric chart shows the moist volume v_{sf} = 13.20 ft^3/lb (0.824 m^3/kg). From Eq. (20.77), the rated volume flow rate of supply fan is

$$\dot{V}_{sf} = \frac{13.20\ Q_{rs}}{60 \times 0.243(T_r - T_s)} = \frac{Q_{rs}}{1.105(T_r - T_s)} \tag{20.81}$$

For cold air distribution, which is covered in detail in Chap. 18, T_{cc} may equal 40°F (4.4°C) with a relative humidity of 98 percent; and if T_{sf} = 42°F (5.6°C), then, v_{sf} = 12.8 ft^3/lb (0.8 m^3/kg), and the rated supply volume flow is

$$\dot{V}_{sf,r} = \dot{V}_{sf} = \frac{12.8\ Q_{rs}}{60 \times 0.243(T_r - T_s)} = \frac{Q_{rs}}{1.14(T_r - T_s)} \tag{20.82}$$

For an air system in which the supply fan is upstream from the cooling coil, and if T_{sf} = 82°F (27.8°C) and φ_{sf} = 43 percent, then, from the psychrometric chart, v_{sf} = 13.87 ft^3/lb (0.866 m^3/kg), and the supply volume flow rate is

$$\dot{V}_{sf,r} = \dot{V}_{sf} = \frac{13.87\ Q_{rs}}{60 \times 0.243(T_r - T_s)} = \frac{Q_{rs}}{1.05(T_r - T_s)} \tag{20.83}$$

Thus, an upstream fan requires 9 percent greater fan capacity than a downstream fan with cold air distribution even though Q_{rs} and $T_r - T_s$ are the same.

20.18 DETERMINATION OF THE SUPPLY AIR CONDITION

The following parametrs are usually known before the constant-volume system cooling mode air conditioning cycle is developed:

1. Summer and winter outdoor design temperatures: dry-bulb T_o and wet-bulb T'_o
2. Summer and winter indoor space air temperature T_r and relative humidity φ_r
3. Space sensible and latent load Q_{rs} and Q_{rl}
4. Outdoor air requirement $\dot{V}_o$
5. Estimated supply system heat gain $q_{s,s}$
6. Estimated return system heat gain $q_{r,s}$

If the state point of the supply air can be determined from the psychrometric chart, then the constant-volume system cooling mode air conditioning cycle can be developed. Eventually, the supply volume flow rate $\dot{V}_s$ and the cooling coil load Q_{cc} can also be determined.

Air Conditioning Rules

To determine the supply air condition, the following air conditioning rules are helpful:

1. For a given summer indoor design temperature T_r, design space sensible cooling load Q_{rs}, and specific supply system heat gain $q_{s,s}$,
   - A lower air off-coil temperature T_{cc} and supply air temperature T_s always result in a greater supply air temperature differential $\Delta T_s = T_r - T_s$ and a lower space relative humidity φ_s.
   - On the other hand, a higher T_{cc} means a smaller T_s and a higher φ_r.
2. A greater ΔT_s reduces the volume flow rate and, therefore, fan size, fan energy use, and terminal and duct sizes. These reduce both first cost and operating costs. At the same time, a greater ΔT_s, needs a lower T_{cc} to maintain an indoor design temperature T_r. Excluding the cold air distribution in the ice storage system, a ΔT_s between 15 and 20°F (8.3 and 11.1°C) is most widely used.
3. Temperature of air leaving the cooling coil T_{cc} is closely related to the temperature of the chilled water entering and leaving the coil, T_{we} and T_{wl}, or to the evaporating temperature inside the DX coil T_{ev}. A lower T_{cc} or T_{we} needs a correspondingly lower T_{ev} and, therefore, higher power input to the refrigeration compressors.
4. For a given T_{cc}, Q_{rs}, and $q_{s,s}$, a supply volume flow rate $\dot{V}_s$ that is greater than the required value results in a lower space temperature T_r, a higher space relative humidity φ_r, and a smaller ΔT_s. On the other hand, a smaller than required $\dot{V}_s$ results in a higher T_r, a lower φ_r, and a greater ΔT_s.
5. When an air system is used to serve a single-zone conditioned space, T_s and T_{cc} should be properly arranged so that the indoor design temperature and relative humidity can be maintained for energy-efficient operation.
6. For an air system serving multizones, with many space temperatures and relative humidity requirements, T_{we} and T_{cc} or T_{ev} should satisfy the lowest T_{cc} requirement. In actual practice, T_{we} and T_{cc}, or T_{ev}, are often determined by previous experience with similar applications.

Graphical Method

The following graphical method may be used to determine the condition of supply air *s* for an air system serving a single zone (including constant-volume single-zone systems) whose T_r and φ_r must be maintained.

1. According to the data given in Sec. 20.14, the relative humidity of the air leaving a cooling coil φ_{cc} with 10 or more fins per inch can be estimated as

4-row coil:	$\varphi_{cc} = 93$ percent
6-row coil:	$\varphi_{cc} = 96$ percent
8-row coil:	$\varphi_{cc} = 98$ percent

2. Draw the space conditioning line *sr* from the state point *r*, parallel to the line of known sensible heat ratio SHR_s, as shown in Fig. 20.24.
3. State point of supply air *s* must lie on line *sr*.
4. The horizontal line *cc-s* represents the magnitude of the supply system heat gain. As segment *cc-s* is moved up and down, there exists a position where *cc* lies exactly on the selected φ_{cc} curve for the coil, *s* lies on line *sr*, and $T_s - T_{cc}$ exactly matches the temperature increase caused by supply system heat gain.
5. In practice, this required T_{cc} must be reconciled with the selected coil's performance for a given coil configuration, T_{we}, and air and water velocities.

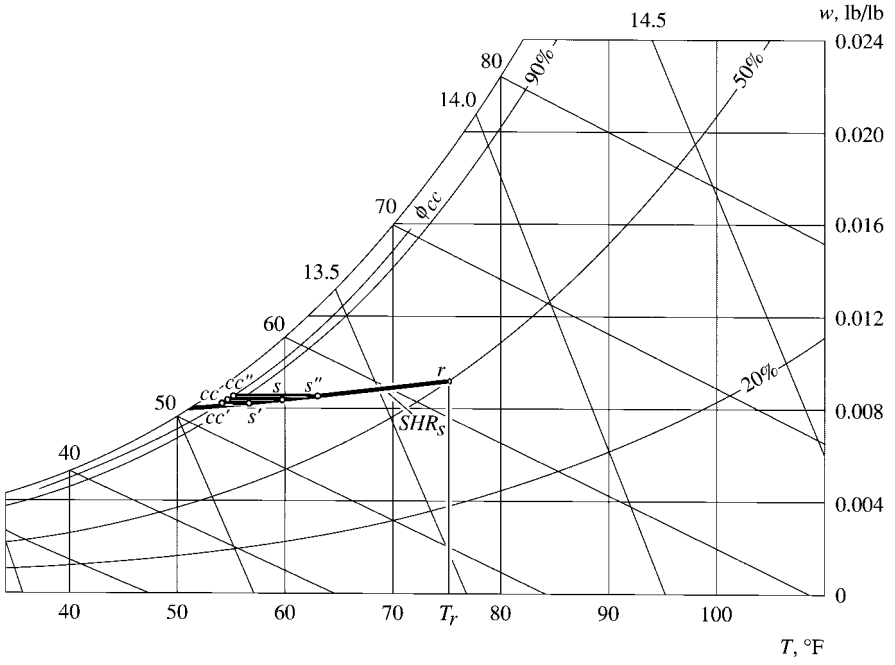


FIGURE 20.24 Determination of state point of supply air.

Influence of Sensible Heat Ratio

For a specific design T_r and φ_r, the sensible heat ratio of the space conditioning line SHR_s has a significant effect on T_s and T_{cc}. Fig. 20.25 shows such an influence. As SHR_s becomes smaller, for the sake of maintaining the required T_r and φ_r, the temperature of conditioned air leaving the cooling coil T_{cc} must be lowered. This requires a lower T_{we} or a lower evaporating temperature T_{ev} in a DX coil, which eventually increases the energy input to the refrigeration compressors, but reduces the air volume flow that must be handled.

For a specific space cooling load Q_{rc}, a lower SHR_s always indicates a smaller Q_{rs} and a greater ΔT_s. A higher ΔT_s means a lower supply volume flow rate $\dot{V}_s$, smaller equipment size, lower fan energy use, and reduced construction and operating costs.

In Fig. 20.25, when $SHR_s = 0.7$, the dew point of space air of 75°F (23.9°C), and 50 percent relative humidity is 55°F (12.8°C), which is higher than the 54°F (12.2°C) supply air temperature. Under such circumstances, condensation may occur at the supply outlet when its cold metal frame contacts the space air at dew point 55°F (12.8°C) or higher.

Condensation can be avoided by the following measures:

- Add an insulation layer with vapor barrier, on the surface of ducts and outlets with sufficient thickness, typically 1 to 2 in. (25 to 50 mm) that are exposed to space air or ambient air whose dew point is higher than the external temperature of the exposed surface.
- Adopt supply outlet that can induce sufficient space air to mix with low-temperature supply air in order to raise T_s.

For an air system to serve a single-zone conditioned space (including constant-volume single-zone systems) with specific indoor T_r and φ_r, say, 75°F (23.9°C) and 50 percent, and a fixed SHR_s, there will be only one supply air condition with a corresponding ΔT_s that can provide the required indoor T_r and φ_r (see Fig. 20.26). If T_r remains constant at 75°F (23.9°C), an increase of ΔT_s due to the

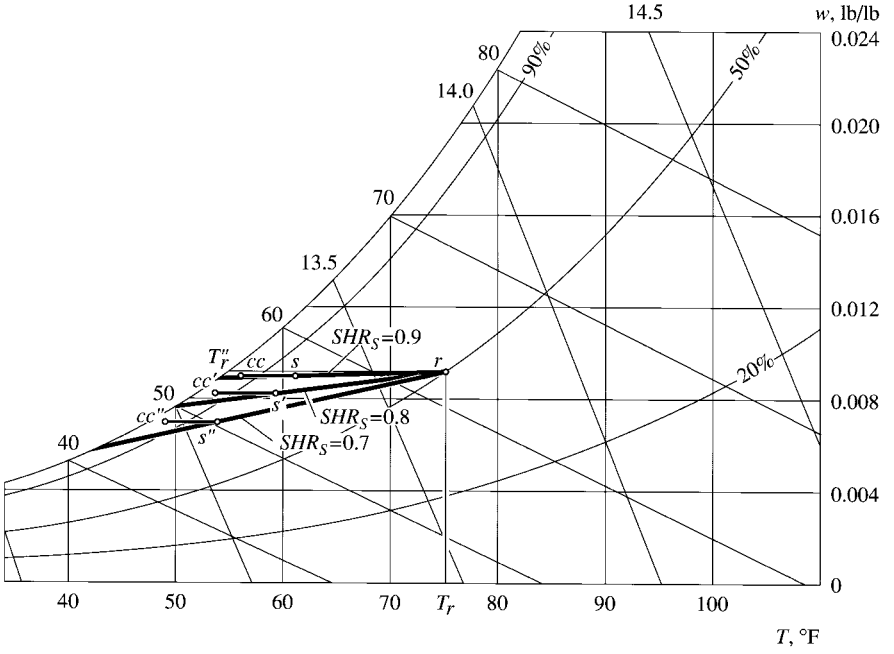


FIGURE 20.25 Effect of sensible heat ratio of space conditioning line on the determination of the condition of supply air.

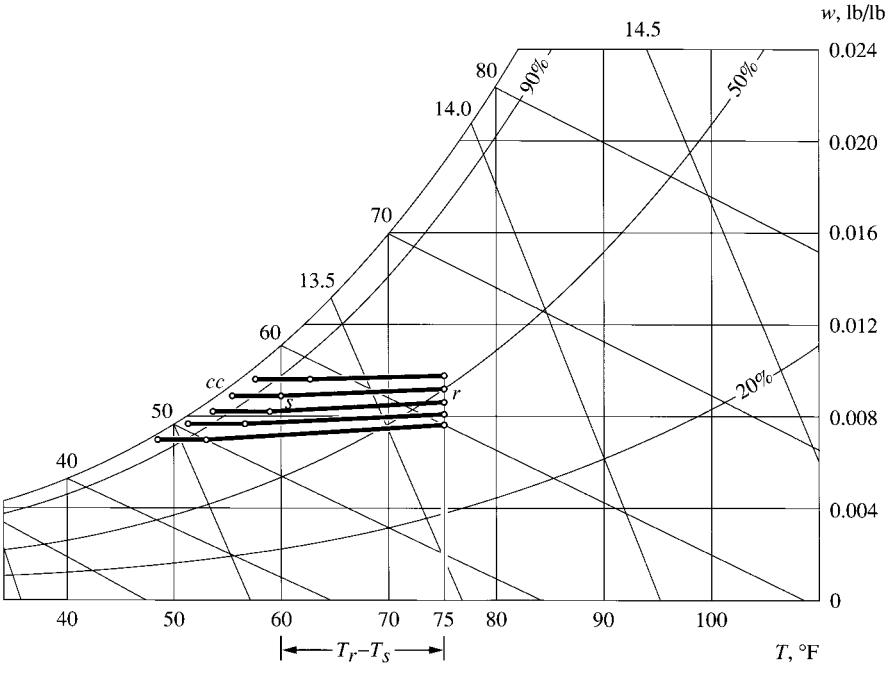


FIGURE 20.26 Influence of supply temperature difference $T_r - T_s$ on T_{cc} and φ_r.

reduction of supply volume flow rate $\dot{V}_s$ at a fixed space sensible load Q_{rs} is possible only when the temperature of air leaving the cooling coil T_{cc} and supply temperature T_s are lowered accordingly. The space relative humidity φ_r also will be lower. This again needs a lower chilled water temperature entering the coil T_{we} or a lower evaporating temperature inside the DX coil T_{ev}.

Example 20.4. One workshop in a factory employs a constant-volume single-zone air system. The summer space sensible cooling load is 350,000 Btu/h (102,550 W), and the summer space latent load is 62,000 Btu/h (18,166 W). Other design data are as follows:

Summer outdoor design temperatures:	
Dry-bulb	95°F (35°C)
Wet-bulb	75°F (23.9°C)
Summer indoor space conditions:	
Air temperature	75°F (23.9°C)
Relative humidity	50 percent
Temperature rise:	
In supply fan	2°F (1.11°C)
In supply duct	3°F (1.67°C)
Because of heat released from light fixtures and roof	3°F (1.67°C)
In return duct and return fan	2°F (1.11°C)
Relative humidity of air leaving cooling coil	93 percent
Outdoor air requirement	3200 cfm (1510 L/s)

Determine

1. The condition of supply air at summer design temperatures
2. The rated volume flow rate of supply fan
3. The cooling coil load

Solution

1. From Eq. (20.30) and the given data, the sensible heat ratio of the space conditioning line can be calculated as

$$\text{SHR}_s = \frac{|Q_{rs}|}{|q_{rs}| + |q_{rl}|} = \frac{350{,}000}{350{,}000 + 62{,}000} = 0.85$$

On the psychrometric chart in the upper summer cool mode air conditioning cycle in Fig. 20.22, draw a space conditioning line *sr* from point *r* with $\text{SHR}_s = 0.85$. From the given data, the total temperature rise of the supply system heat gain is 2 + 3 = 5°F. Point *cc* will lie on φ_{cc} 93 percent line, and point *s* must lie on line *sr*. Let segment *cc*-*s* = 5°F. Move *cc*-*s* until point *cc* is on the 93 percent curve and point *s* is on line *sr*. The state point of supply air *s* and the condition of air leaving the cooling coil, point *cc*, are then determined. From the psychrometric chart,

$T_s = 60°\text{F}\ (15.6°\text{C})$ $\quad \varphi_s = 78$ percent and $w_s = 0.0086$ lb/lb (kg/kg)

$T_{cc} = 55°\text{F}\ (12.8°\text{C})$ $\quad \varphi_{cc} = 93$ percent $\quad w_{cc} = 0.0086$ lb/lb (kg/kg)

$h_{cc} = 22.6$ Btu/lb (52.6 kJ/kg)

2. Because the air temperature at the supply fan outlet $T_{sf} = 55 + 2 = 57°F$, and $w_{sf} = w_{cc} = 0.0086$ lb/lb, from the psychrometric chart, moist volume $v_{sf} = 13.21$ ft^3/lb. From Eqs. (20.77) and (20.78) the rated supply volume flow rate of the supply fan is

$$\dot{V}_{sf,r} = \dot{V}_{sf} = \frac{Q_{rs}}{60\rho_{sf}c_{pa}(T_r - T_s)}$$

$$= \frac{13.21 \times 350{,}000}{60 \times 0.243(75\text{-}60)} = 21{,}140 \text{ cfm } (9976 \text{ L/s})$$

3. State point *m* can be graphically determined or calculated. The total temperature rise due to the return system heat gain is $3 + 2 = 5°F$; therefore, $T_{rf} = T_{ru} = 75 + 5 = 80°F$. Because line *r-ru* is a horizontal line, point *ru* can be plotted on the psychrometric chart. From the given data, the summer outdoor temperature $T_o = 95°F$, and the summer design outdoor wet-bulb temperature $T'_o = 75°F$, so the state point of outdoor air *o* can also be plotted. Connect line *ru-o*. Ignore the difference in density between points *ru*, *m* and *o*, Then

$$\frac{ru\text{-}m}{ru\text{-}o} = \frac{3200}{21{,}140} = 0.15$$

From the psychrometric chart, the length of line section *ru-o* is 1.8 in.; then point *m* can be determined as

$$T_m = 82°F \text{ } (27.8°C) \qquad \varphi_m = 44 \text{ percent} \qquad \text{and} \qquad h_m = 30.9 \text{ Btu/lb } (71.9 \text{ kJ/kg})$$

Mixing point *m* can also be determined by using the analytical method. From the psychrometric chart, the enthalpy of outdoor air $h_o = 38.4$ Btu/lb and the enthalpy $h_{ru} = 29.8$ Btu/lb. From Eq. (20.55),

$$h_m = \frac{\dot{V}_o h_o + (\dot{V}_s - \dot{V}_o)h_{ru}}{\dot{V}_m}$$

$$= \frac{3200 \times 38.4 + (21{,}140 - 3200)29.8}{21{,}140}$$

$$= 31.10 \text{ Btu/lb } (72.3 \text{ kJ/kg})$$

$$T_m = \frac{\dot{V}_o T_o + (\dot{V}_s - \dot{V}_o)T_{ru}}{\dot{V}_m}$$

$$= \frac{3200 \times 95 + (21{,}140 - 3200)80}{21{,}140} = 82.3°F(27.95°C)$$

The differences between the enthalpies and temperatures determined from the graphical and analytical methods are small, less than 1 percent. But the analytical method is more accurate because the line segment lengths are small. Because $\dot{V}_s\rho_s = \dot{V}_{sf}\rho_{sf}$, from Eq. (20.48*b*), the cooling coil load is calculated as

$$Q_{cc} = 60\dot{V}_{sf}\rho_{sf}(h_m - h_{cc})$$

$$= 60 \times \frac{21{,}140(31.1 - 22.6)}{13.21} = 816{,}154 \text{ Btu/h } (239{,}133 \text{ W or } 239 \text{ kW})$$

Example 20.5. A factory uses a constant-volume single-zone system without space humidity control. The operating parameters at winter design conditions are as follows:

Winter outdoor design conditions	
Air temperature	32°F (0°C)
Relative humidity	30 percent
Winter indoor design conditions	
Air temperature	72°F (22.2°C)
Space sensible cooling load	800,000 Btu/h (234,400 W)
Space latent load	600,000 Btu/h (175,800 W)
Temperature rise due to supply system heat gain	4°F (2.22°C)
Temperature rise due to return system heat gain	2°F (1.11°C)
Supply volume flow rate	100,000 cfm (47,190 L/s)
Minimum outdoor air requirement	26,250 cfm (12,387 L/s)

At winter design condition, a mixture of outdoor air and recirculating air forms the cold air supply (cooling mode operation) to offset the space cooling load.

Determine the space relative humidity at winter design conditions and the actual amount of outdoor air intake.

Solution. Assume that the moist volume of the supply air $v_s = 13.3$ ft^3/lb. From Eq. (20.65) and the given data, the humidity ratio difference between the space and supply air can be calculated as

$$w_r - w_s = \frac{Q_{rl}}{60\dot{V}_s\rho_s h_{fg,32}}$$

$$= \frac{600{,}000 \times 13.3}{60 \times 100{,}000 \times 1075} = 0.00125 \text{ lb/lb (kg/kg)}$$

From Eq. (20.31) and the given data, the supply air temperature difference $T_r - T_s$ can be calculated as

$$T_r - T_s = \frac{Q_{rs}}{60\dot{V}_s\rho_s c_{pa}}$$

$$= \frac{800{,}000 \times 13.3}{60 \times 100{,}000 \times 0.243} = 7.30 \text{ °F (4.1°C)}$$

From the given data

$$T_{ru} = T_r + 2 = 72 + 2 = 74°\text{F}$$

and

$$T_m = T_r - (T_r - T_s) - 4 = 72 - 7.3 - 4 = 60.7°\text{F (15.9°C)}$$

Because $w_{ru} = w_r$ and $w_m = w_s$, from Eq. (20.67)

$$\frac{w_r - w_s}{w_{ru} - w_o} = \frac{T_{ru} - T_m}{T_{ru} - T_o}$$

$$= \frac{74 - 60.7}{74 - 32} = 0.317$$

Then

$$w_r - w_o = \frac{w_r - w_s}{0.317} = 0.00394 \text{ lb/lb (kg/kg)}$$

From the psychrometric chart, at a temperature of 32°F and a relative humidity of 30 percent, $w_o = 0.0012$ lb/lb, so

$$w_r = w_o + 0.00394 = 0.0012 + 0.00394 = 0.00514 \text{ lb/lb}$$

At a temperature of 72°F (22.2°C) and $w_r = 0.00514$ lb/lb (kg/kg), the space relative humidity $\varphi_r = 31$ percent, as shown in the lower part of the psychrometric chart in Fig. 20.23. The amount of outdoor air intake can be calculated as

$$\dot{V}_o = \frac{\dot{V}_s(T_{ru} - T_m)}{T_{ru} - T_o}$$
$$= 100{,}000 \times 0.317 = 31{,}700 \text{ cfm } (14{,}959 \text{ L/s})$$

This amount is greater than the amount of outdoor air required 26,250 cfm (12,387 L/s). At $T_s = 72 - 7.3 = 64.7$°F and $w_s = 0.00514 - 0.00125 = 0.00389$ lb/lb, from the psychrometric chart, $v_s = 13.3$ ft^3/lb (0.830 m^3/kg), which is equal to the assumed value.

20.19 CONSTANT-VOLUME SINGLE-ZONE SYSTEMS—HEATING MODE OPERATION

Heating Mode without Space Humidity Control

In many residential and small commercial buildings, the sum of the heat loss due to transmission through the windows, walls, and roofs and the heat loss due to infiltration in winter is greater than the sum of the internal heat gains from occupants, electric lights, equipment, and appliances. Then a warm air supply is required to maintain a desirable indoor temperature. A constant-volume single-zone system operated in the heating mode without space humidity is often used.

The lower air conditioning cycle in Fig. 20.22 shows such a heating mode cycle in winter. Return air at state point *r* flows through the return grilles, return duct, and return fan. After that, recirculating air at point *ru* enters the air-handling unit and mixes with the outdoor air. The mixture, at point *m*, passes through the cooling coil which is not energized, and is then heated at the heating coil. Warm air leaves the heating coil at point *ch*. It leaves the supply fan at point *sf* after absorbing the fan power heat gain. The warm supply air loses heat in the supply duct and supplies to the conditioned space at point *s*. At the conditioned space, the supply air absorbs the latent heat from the occupants; supplies heat to compensate for transmission loss through external windows, walls, and roofs; and finally changes to the point *r*.

The air conditioning cycle in heating mode operation without space humidity control during winter design condition (as shown in Fig. 20.22) consists of the following processes:

1. Sensible heating process *r-ru* due to the return system heat gain q_{rs}. In winter, the return system heat gain consists of mainly the heat released from the electric lights q_{rp} and the return fan power heat gain q_{rf}, if any. That is,

$$q_{r,s} = q_{rp} + q_{rf} \tag{20.84}$$

   The temperature difference between the air inside and outside the return duct is generally small in winter and can be ignored.

2. Adiabatic mixing process *ru-m-o*. The ratio of the minimum volume flow rate of outdoor air to the supply volume flow rate is the same as in the cooling mode in summer design condition in order to provide the required amount of outdoor air for occupants.

3. Sensible heating process *m-ch* at the heating coil. The heating coil load can be calculated according to Eq. (20.39) as

$$Q_{ch} = 60\ \dot{V}_s \rho_s c_{pa}(T_{ch} - T_m) \tag{20.85}$$

   where T_m, T_{ch} = temperature of mixture entering coil and air off heating coil,°F (°C).

4. Sensible heating process *ch-sf* due to the supply fan power heat gain q_{sf}.

5. Sensible cooling process *sf-s* due to the heat loss from the warm air inside the supply duct through the duct wall to the ambient air q_{sd}.
6. Space conditioning process *sr*. During heating mode operation in winter, heat is supplied to the space to compensate for the transmission loss, and moisture is absorbed by the supply air due to the space latent load. Supply air at point *s* is then changed to *r*.
   The heat supplied to the conditioned space to offset the space heating load Q_{rh}, in Btu/h (W), can be calculated as

$$Q_{rh} = 60\,\dot{V}_s \rho_s c_{pa}(T_s - T_r) \tag{20.86}$$

   And the latent heat absorbed by the supply air Q_{rl}, in Btu/h (W), is

$$Q_{rl} = 60\dot{V}_s \rho_s (w_r - w_s) h_{fg,32} \tag{20.87}$$

Example 20.6. For the same constant-volume single-zone system as that in Example 20.4, shown in Fig. 20.22, the following winter design conditions are given:

Winter outdoor design conditions:	
Dry-bulb temperature	15°F (9.4°C)
Relative humidity	30 percent
Winter indoor design temperature	72°F (22.2°C)
Outdoor air requirement	3200 cfm (1510 L/s)
Temperature rise:	
Due to heat release from light fixtures	2°F (1.11°C)
Return fan power heat gain	1°F (0.56°C)
Supply fan	2°F (1.11°C)
Temperature drop of the supply duct	1°F (0.56°C)
Space heating load	225,000 Btu/h (65,925 W)
Space latent load	60,000 Btu/h (17,580 W)
Supply volume flow rate	21,140 cfm (9976 L/s)

Determine

1. The space relative humidity at winter design conditions
2. The temperature of supply air
3. The heating coil load

Solution

1. To develop a winter heating mode air conditioning cycle for a constant-volume single-zone system, the space humidity ratio w_r must be determined. Because the winter indoor temperature is given, the space relative humidity can then be found from the psychrometric chart.

Assume that the moist volume of the warm supply air $v_s = 13.75$ ft^3/lb (0.86 m^3/kg). Because $\rho_s = 1/v_s$, from Eq. (20.87)

$$w_r - w_s = \frac{Q_{rl}}{60\dot{V}_s \rho_s h_{fg,32}}$$

$$= \frac{60{,}000 \times 13.75}{60 \times 21{,}140 \times 1075}$$

$$= 0.000605 \text{ lb/lb}$$

Because $w_m = w_s$,

$$\frac{w_r - w_s}{w_r - w_o} = \frac{3200}{21{,}140} = 0.15$$

Therefore,

$$w_r - w_o = \frac{0.000605}{0.15} = 0.0041 \text{ lb/lb}$$

From the psychrometric chart, at $T_o = 15°\text{F}$ and $\varphi_o = 30$ percent, $w_o = 0.0005$ lb/lb. Then

$$w_r = w_o + 0.0041 = 0.0005 + 0.0041 = 0.0046 \text{ lb/lb (kg/kg)}$$

Point *r* can thus be plotted on the psychrometric chart. The relative humidity of the space air at winter design condition is 30 percent.

2. From the given data, $T_{ru} = T_r + 2 + 1 = 72 + 2 + 1 = 75°\text{F}$. Because $w_{ru} = w_r$, points *ru* and *o* can be plotted from the psychrometric chart, as shown in Fig. 20.22, upper figure. Connect *ru-o*. For an adiabatic mixing process

$$\frac{ru\text{-}m}{ru\text{-}o} = \frac{3200}{21{,}140} = 0.15$$

Section *ru-o* measured from the psychrometric chart is 6.15 in. Then mixing point *m* can be determined from the psychrometric chart:

$$T_m = 64.3°\text{F (17.9°C)} \qquad \text{and} \qquad w_m = 0.004 \text{ lb/lb (kg/kg)}$$

From Eq. (20.86), the supply air temperature

$$T_s = T_r + \frac{Q_{rh}}{60\,\dot{V}_s \rho_s c_{pa}}$$

$$= 70 + \frac{225{,}000 \times 13.75}{60 \times 21{,}140 \times 0.243} = 80.0°\text{F (26.7°C)}$$

Because $w_m = w_{sf} = 0.004$ lb/lb, point *s* can be plotted. From the psychrometric chart, $v_s = 13.72$ ft^3/lb (0.856 m^3/kg) which is approximately equal to the assumed value. If the assumed value is too high or too low, the calculation will be repeated with a new value of v_s.

3. Warm air temperature off the heating coil at winter heating mode can be calculated by

$$T_{ch} = T_s + 1 - 2 = 80.0 + 1 - 2 = 79.0°\text{F}$$

And from Eq. (20.85), if $\rho_s = 1/13.70$, the heating coil load is

$$Q_{ch} = 60\,\dot{V}_s \rho_s c_{pa}(T_{ch} - T_m)$$

$$= 60 \times 21{,}140 \times \frac{1}{13.70}(0.243)(79.0 - 64.3) = 330{,}719 \text{ Btu/h (96,901 W)}$$

Heating Mode with Space Humidity Control

In health care buildings with special requirements, or in industrial buildings humidification may have to be incorporated in the warm air supply in winter. Figure. 20.27 shows an air conditioning cycle operated in winter heating mode with steam injection humidification and the corresponding schematic diagram of the constant-volume single-zone system. This cycle consists of the following processes:

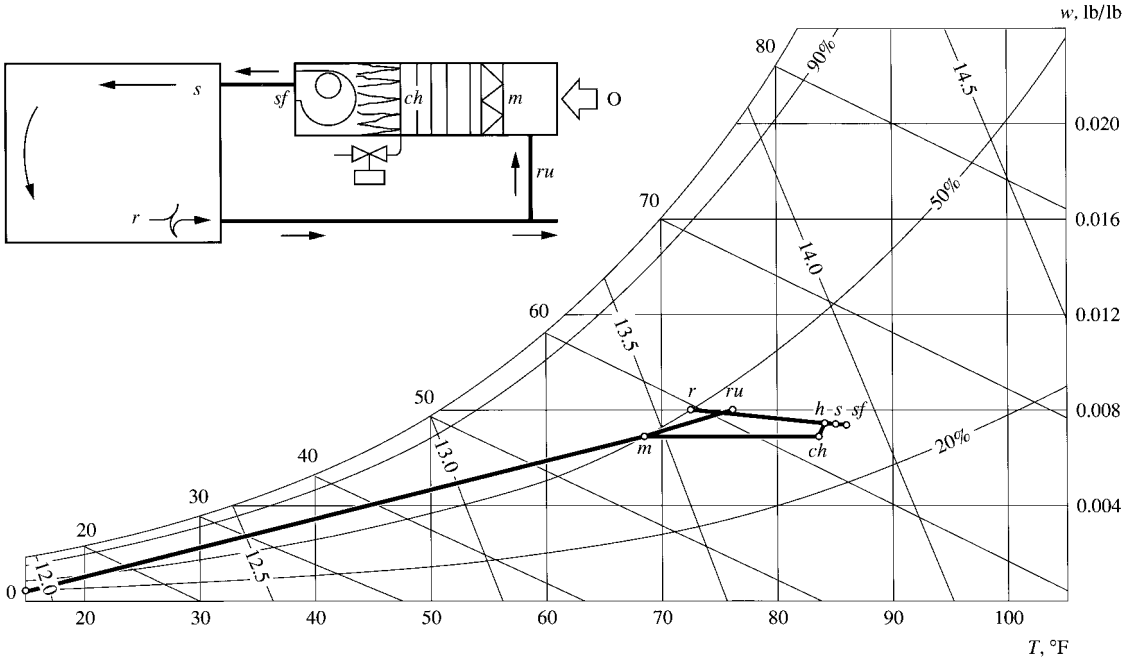


FIGURE 20.27 Air conditioning cycle—winter heating mode operation with space humidity control for a constant-volume single-zone system.

1. Sensible heating process *r-ru* from return system heat gain
2. Mixing process *ru-m-o* of recirculated air and outdoor air
3. Sensible heating process *m-ch* at the heating coil
4. Steam injection or heated element humidifying process *ch-h*
5. Sensible heating process *h-sf* from supply fan power heat gain
6. Sensible cooling process *sf-s* from duct heat loss
7. Space conditioning process *sr*

The procedure for constructing this cycle is as follows:

1. Plot the state points of the conditioned space *r* and outdoor air *o*.
2. Draw the space conditioning line *sr* from point *r* with known sensible heat ratio SHR_s.
3. From Eq. (20.87), calculate $w_r - w_s$. Since w_r, Q_{rl}, and the supply volume flow rate from summer cooling mode calculations are known values, w_s can be calculated. Point *s* must lie on line *sr*, so the intersection of *sr* and w_s line determines point *s*.
4. Draw a horizontal line *r-ru* from point *r*, representing the return system sensible heat gain. Plot point *ru* from its known temperature rise.
5. Connect points *ru* and *o*. From the ratio of the volume flow rate of the outdoor air to the supply air $\dot{V}_o/\dot{V}_s$, point *m* can be found and located on line *ru-o*.
6. Draw horizontal line *m-ch* from point *m*. Draw also a horizontal line *sf-s-h* through point *s*. Temperature difference $T_{\text{sf}} - T_h$ represents the temperature rise of the supply fan heat gain, and $T_{\text{sf}} - T_s$ is the supply duct heat loss. This defines points *sf* and *h*.

7. Draw a vertical line through *ch* to line *h-sf* extended from point *h*. The intersection of lines *h-ch* and *m-ch* locates point *ch*. Because $w_m = w_{ch}$, the mass flow rate of the injected steam is the mass flow rate of the water vapor $\dot{m}_s$, in lb/min (kg/s), added to the moist air, and can be calculated from Eq. (20.42)

$$\dot{m}_s = \dot{V}_s \rho_s (w_h - w_m) \tag{20.88}$$

where w_h = humidity ratio of air leaving humidifier, lb/lb (kg/kg)
w_m = humidity ratio of mixture, lb/lb (kg/kg)

Part-Load Operation

For a constant-volume single-zone system, when the space heating load is reduced during part-load operation, as in the the cooling mode operation, hot water flow rate modulation is used to reduce the supply air temperature and the heating capacity if a hot water heating coil is employed in an air-handling unit. For a packaged unit using a gas-fired furnace as a heating source, a DDC unit controller will directly adjust the amount of gas supplied to the burner when the temperature senser senses the rise of space temperature during part-load operation.

Dual-Thermostat Year-Round Zone Temperature Control

For most of the single-zone constant-volume systems employed in residential buildings and small retail stores, two separate thermostats with one set point for cooling mode operation and another set point for heating mode operation are often used, as shown in Fig. 20.28. During cooling mode operation, when the zone temperature drops below the cooling set point, the refrigeration system is shut down; or during heating mode operation, when the zone temperature rises above the heating set point, the gas furnace is shut off. On/off control is most often used.

Between the cooling and heating mode operation, there is a dead band in which neither cooling nor heating is provided. During the dead band mode, the constant-volume single-zone system is

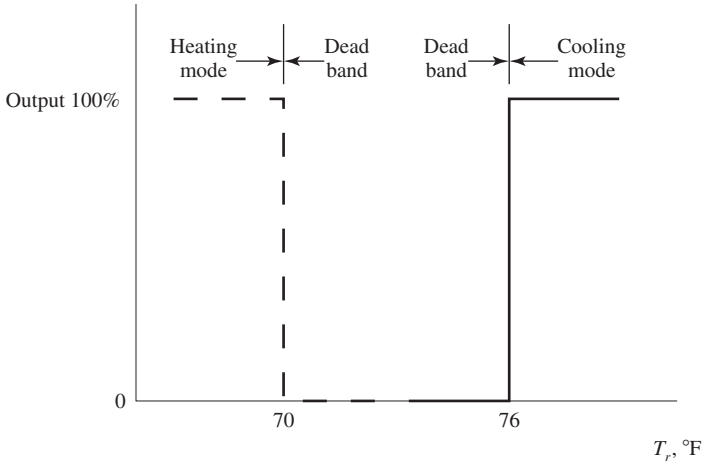


FIGURE 20.28 Dual-thermostat year-round zone temperature control.

often shut down, and sometimes the fan remains continuously operating to provide necessary outdoor ventilation air for interior spaces. The on/off control of the cooling or heating mode operation for such a constant-volume single-zone system is performed manually or automatically. It is possible to have a dead band width of 5°F (2.8°C) or even wider for such a dual dual-thermostat year-round zone-temperature control system.

If the cooling capacity of a single-zone constant-volume system is greater than 5 tons (17.6 kW), multiple single-zone constant-volume packaged units may be used.

20.20 CONSTANT-VOLUME MULTIZONE SYSTEM WITH REHEAT

Reheating, Recooling, and Mixing

In zone controls for multizone systems, reheating is a process in which air is reheated after it has been cooled. Recooling is a process in which air is cooled after it has been previously heated by using combustion fuels, and in the mixing process a cooler airstream is often mixed with warmer recirculating air, or a warmer airstream mixed with cooler outdoor air. Reheating, recooling, and mixing processes are all simultaneous cooling and heating processes in air conditioning cycles, and are energy-inefficient air conditioning processes.

ASHRAE/IESNA Standard 90.1-1999 specifies that zone thermostatic controls (either in constant-volume or VAV systems) shall be capable of operating in sequence the supply of heating and cooling energy to the zone. Zone controls shall prevent (1) reheating, (2) recooling, (3) mixing or simultaneously supplying air that has been previously mechanically heated with air that has been previously cooled by refrigeration or by an economizer, and (4) other simultaneous operation of heating and cooling systems to the same zone except the following (exceptions for VAV systems will be listed and discussed in Chap. 21):

a. Zones in which the volume flow of reheated, recooled, or mixed air is no greater than the larger of the following:

- The volume flow of outside air to meet the ventilation requirement of ASHRAE Standard 62-1999.
- 0.4 cfm/ft^2 (2L/s · m^2) of the zone conditioned floor area provided that primary air temperature is controlled:

  0 to 12°F (0 to 6.7°C) below the design space heating temperature $T_{r,h}$ when outdoor air temperature $T_o > 60$°F (15.6°C) for reheat systems and the cold deck mixing systems.

  0 to 12°F (0 to 6.7°C) above design space temperature T_r when $T_o > 60$°F (15.6°C) for recooling system and the hot deck mixing systems.

  Each zone need not comply with this exception provided the average of all zones served by this system that have both heating and cooling ability comply.

- 300 cfm (142 L/s). This exception is for zones whose peak flow rate totals no more than 10 percent of the total fan system (air system) volume flow rate.
- Any higher rate that is satisfied by the authority having jurisdication to reduce overall air system annual energy usage by offsetting reheat/recool energy losses through a reduction in outdoor air intake in accordance with the multiple space requirements in ASHRAE Standard 62-1999.

b. Zones where special pressurization relationships, cross-contamination requirements, or code-required minimum circulation rates are such that variable-air-volume systems are impractical.

c. Zones where at least 75 percent of the energy for reheating or for warm air mixing systems is provided from a site recovered (including condenser heat) or site solar energy source.

Reheating is a simple and effective means of controlling space temperature and relative humidity at part load. Reheating is used with compliance to ASHRAE/IESNA Standard 90.1-1999 for VAV systems at minimum volume flow rate settings for comfort air conditioning, and in hospitals, laboratories, computer rooms, museums, and precision manufacturing environments (such as semiconductor manufacturing plants) in order to maintain a close tolerance in space temperature ($\pm$ 1°F or $\pm$ 0.56°C) or space relative humidity for processing air conditioning.

Constant-Volume Multizone System with Reheat

A constant-volume, multizone system with reheat is shown in Fig. 20.29. During cooling mode operation, as shown in Fig. 20.29, outdoor air at point *o* is mixed with recirculating air at point *ru*. The mixture *m* flows through the filter. It is then cooled and dehumidified to point *cc* at the cooling coil and discharged from the AHU or PU at the supply fan outlet *sf*. After absorbing the duct heat gain, the conditioned air enters the reheating coils 1 and 2 at point *s*.

During summer design load operation, theoretically, reheating coil 1 is not energized. Air is supplied to room 1 at condition *s*1. Conditioned air is reheated at reheating coil 2 to maintain the required room temperature in room 2. After absorbing the space cooling load, the supplied air becomes room air *r*1 and *r*2. Returned air from rooms 1 and 2 is mixed together and forms mixture *rm* in the ceiling plenum. The return air is then returned to the AHU or PU, and the recirculating air enters the AHU or PU at point *ru*. A portion of return air is exhausted through the relief fan to the outdoors.

During winter operation, outdoor air is mixed with recirculating air in a ratio such that its amount is always greater than the minimum outdoor air requirement. The mixture *m* flows through the filter, the deenergized cooling coil, and the supply duct and is reheated at the reheating coils 1 and 2. The heated air is supplied to rooms 1 and 2 to offset the heating load there. Both hot water and electric reheating coils can be used in multizone systems. When electric reheating coils are used, the design and installation requirements described in Sec. 8.4 and the requirements of the National Electric Code and local codes must be followed.

Control Systems

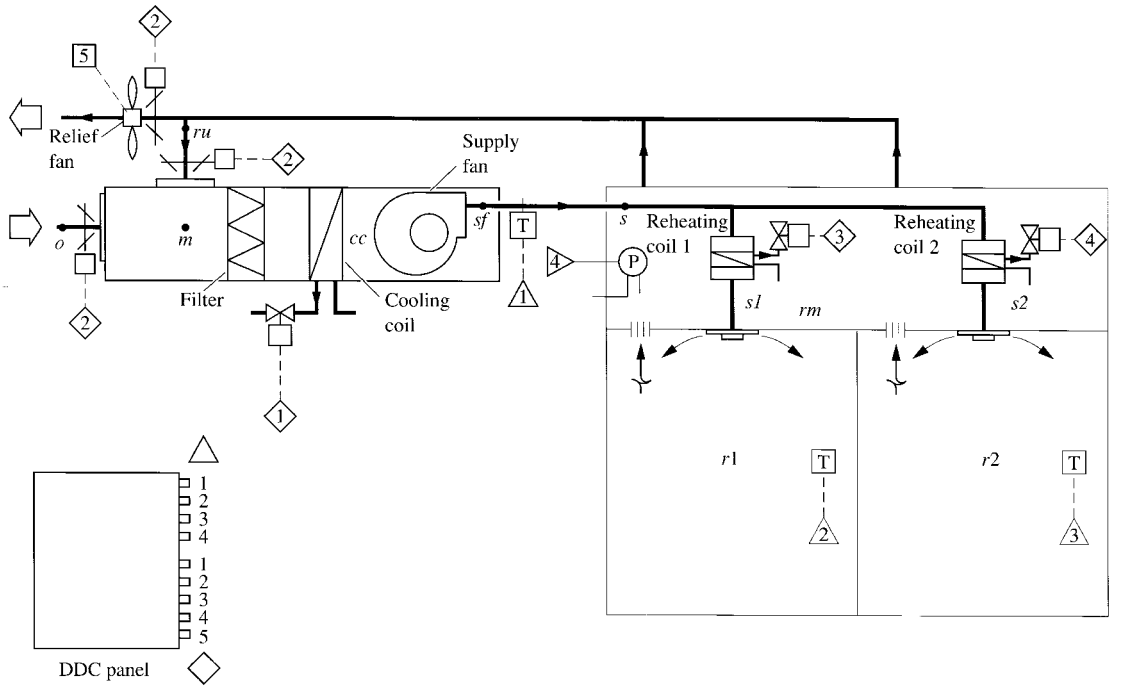
As described in Sec. 5.14, the controls for a constant-volume multizone system with reheat using an AHU or a PU may include the following:

- Discharge air temperature control
- Outdoor ventilation air control
- Space pressurization control
- Air economizer control
- Zone smoke control

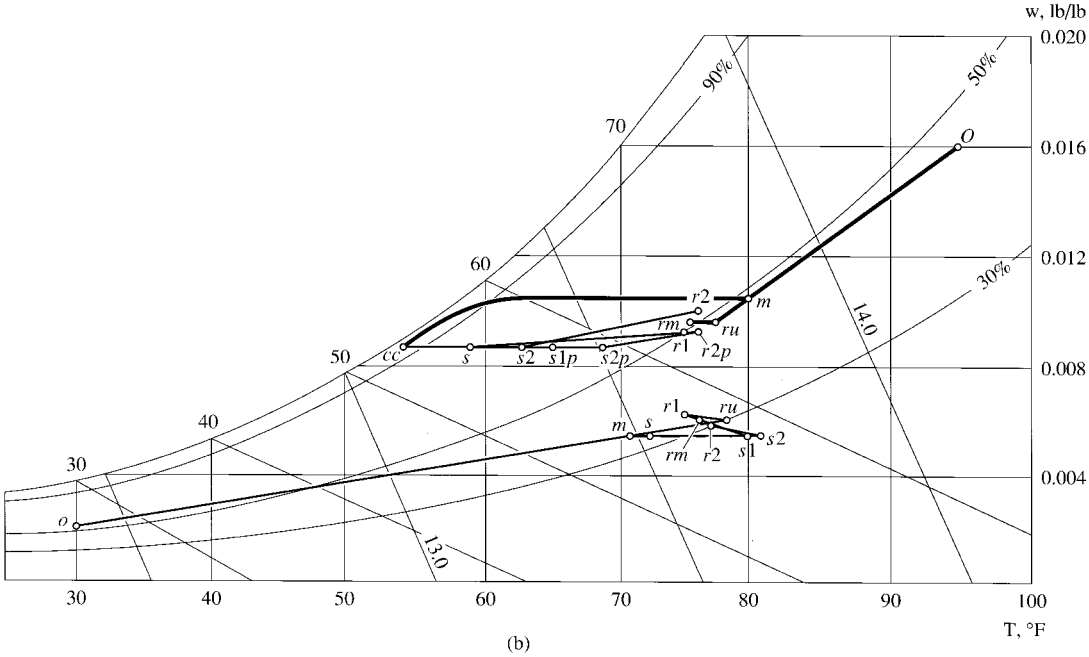
All these functional controls for an air system are discussed in Chap. 23 except zone smoke control, discussed in Chap. 22.

If there are only a few reheating coils, both the AHU or PU and the reheating coils can be controlled by a DDC unit controller. If there are many reheating coils, the AHU or PU and the reheating coils should be controlled by a DDC system controller.

When room temperature sensors T2 and T3 sense a drop in either room temperature below a limit, a signal is sent to the DDC controller. The reheating coil is then energized. For a hot water reheating coil, the two-way valve is modulated so that room temperature is maintained within predetermined limits. For an electric reheating coil, heating is added or decreased in steps. To prevent fires, electric heating is energized only when airflow is sensed by a flow switch.



(a)



(b)

FIGURE 20.29 Constant-volume multizone system with reheat: (*a*) schematic diagram; (*b*) air conditioning cycle.

Operating Parameters and Calculations

1. The zone supply air condition should be determined so that the space temperature and relative humidity in each zone or room can be maintained at design load operation by using the reheating processes.

2. The zone n supply mass flow rate $\dot{m}_{an}$, in lb/min [kg/(60 s)], and volume flow rate $\dot{V}_{sn}$, in cfm [kg/(60 s)], at summer design condition can be calculated as

$$\dot{m}_{an} = \frac{Q_{rsn}}{60c_{pa}(T_{rn} - T_s)} \tag{20.89}$$

$$\dot{V}_{sn} = v_s\dot{m}_{an} \tag{20.90}$$

where Q_{rsn} = sensible cooling load for zone n, Btu/h (W)
T_{rn} = temperature of zone n,°F (°C)
T_s = supply temperature of zone n,°F (°C)
$v_s = 1/\rho_s$, moist volume of supply air, ft^3/lb (m^3/kg)

3. The zone n supply temperature at minimum part-load operation T_{snp}, in°F (°C), can be calculated as

$$T_{snp} = T_{rnp} - \frac{Q_{rsn}}{60\dot{m}_{an}c_{pa}} \tag{20.91}$$

where T_{rnp} = temperature of zone at minimum part-load, operation °F (°C).

4. The zone n reheating coil load Q_{hn}, in Btu/h (W), at minimum part-load operation can be calculated as

$$Q_{hnp} = 60\dot{m}_{an}c_{pa}(T_{snp} - T_s) \tag{20.92}$$

System supply temperature T_s varies along the supply main duct. For simplicity, T_s can be taken as an average value.

5. The weighted mean value of zone temperature T_{rm}, in °F (°C), and mean humidity ratio w_{rm}, in lb/lb (kg/kg), can be calculated as

$$T_{rm} = \frac{\dot{m}_{a1}T_{r1} + \dot{m}_{a2}T_{r2} + \cdots + \dot{m}_{an}T_{rn}}{\dot{m}_{a1} + \dot{m}_{a2} + \cdots + \dot{m}_{an}} \tag{20.93}$$

and

$$w_{rm} = \frac{\dot{m}_{a1}w_{r1} + \dot{m}_{a2}w_{r2} + \cdots + \dot{m}_{an}w_{rn}}{\dot{m}_{a1} + \dot{m}_{a2} + \cdots + \dot{m}_{an}} \tag{20.94}$$

where $\dot{m}_{a1}, \dot{m}_{a2}, \cdots, \dot{m}_{an}$ = mass flow rate of supply air for zones 1, 2, . . . , n, lb/min (kg/s)
$T_{r1}, T_{r2}, \ldots, T_{rn}$ = air temperatures in zones 1, 2, . . . , n,°F (°C)
$w_{r1}, w_{r2}, \ldots, w_{rn}$ = air humidity ratio in zones 1, 2, . . . , n, lb/lb (kg/kg)

6. The system supply volume flow rate $\dot{V}_s$, in cfm (m^3/s), of the AHU or PU is calculated as

$$\dot{V}_s = \dot{V}_{s1} + \dot{V}_{s2} + \cdots + \dot{V}_{sn} \tag{20.95}$$

where $\dot{V}_{s1}, \dot{V}_{s2}, \ldots, \dot{V}_{sn}$ = supply volume flow rate for zones 1, 2, . . . , n, cfm (m^3/s).

7. The condition of recirculating air entering the AHU or PU, point *ru*, and the mixture of outdoor and recirculating air *m* can be determined graphically on the psychrometric chart.The cooling coil load Q_{cc}, in Btu/h (W), can therefore be calculated as

$$Q_{cc} = 60(\dot{m}_{a1} + \dot{m}_{a2} + \cdots + \dot{m}_{an})(h_m - h_{cc}) \tag{20.96}$$

where h_m, h_{cc} = enthalpy of mixture and conditioned air off coil, Btu/h (W)
$\dot{m}_a$ = mass flow rate of mixture, cfm [kg/(60 s)]

8. At part-load operation, space cooling load reduces and more reheating is required. The condition of zone supply air point *snp* can be determined because *snp* must lie on horizontal line *ccsnp*. Zone supply air temperature at part load T_{snp} can be calculated from Eq. (20.91). Draw a line from *snp* with the known sensible heat ratio for zone *n* at part load SHR_{np}. This line intersects zone temperature line T_{rnp} at point *rnp*, which is the state point of zone air at part load. Zone relative humidity φ_{rnp} can then be determined.

The operating parameters during winter heating mode operation at design load can be determined as follows:

- At winter design load, $w_{rm} = w_{ru}$ and $w_s = w_m$. Here w_{ru} indicates the humidity ratio of the recirculating air, w_s is the humidity ratio of system supply air before entering the reheating coil, and w_m is the humidity ratio of the mixture of the outdoor air and recirculating air, all in lb/lb (kg/kg). Then

$$w_{ru} = w_s + \frac{Q_{rl}}{60(\dot{m}_{a1} + \dot{m}_{a2} + \cdots + \dot{m}_{an})h_{fg,32}} \tag{20.97}$$

Also,
$$\frac{w_{ru} - w_m}{w_{ru} - w_o} = \frac{\dot{V}_o}{\dot{V}_s} \tag{20.98}$$

where w_o = humidity ratio, lb/lb (kg/kg); $\dot{V}_o$ = volume flow rate of outdoor air, cfm [kg/ (60 s)].

- Zone supply temperature T_{sn}, in °F (°C), can be calculated as

$$T_{sn} = T_{rn} + \frac{Q_{rhn}}{60\dot{V}_{sn}\rho_s c_{pa}} \tag{20.99}$$

where Q_{rhn} = space heating load for zone *n*, Btu/h (W).

- The reheating coil load for zone *n* during winter heating mode operation Q_{hn}, in Btu/h (W), is

$$Q_{hn} = 60\dot{m}_{an}c_{pa}(T_{sn} - T_s) \tag{20.100}$$

where T_s = supply air temperature before reheating coil, °F (°C). Usually, the reheating coil load during winter heating mode operation is greater than in summer cooling mode minimum part-load operation, and should be taken as the design capacity of the reheating coil. However, it is necessary to select the greater value of these two as the reheating coil design capacity.

- At part-load operation, space heating load reduces, and the reheating coil load decreases accordingly.

System Characteristics

If the humidity ratio of the supply temperature T_s remains the same during summer cooling mode or winter heating mode operation, the space relative humidity of a constant-volume multizone system with reheat at part load is always lower than that at design load.

There is also another constant-volume multizone system that has a hot deck and a cold deck. For each control zone, a separate air duct is connected to the hot deck, and another separate air duct is connected to the cold deck; each has a damper. Constant-volume multizone systems waste energy and also result in a very complicated control system. They are now replaced by other more energy-efficient systems.

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CHAPTER 21

AIR SYSTEMS: VARIABLE-AIR-VOLUME SYSTEMS

21.1 SYSTEM CHARACTERISTICS OF VARIABLE-AIR-VOLUME SYSTEMS 21.2
- Variable-Air-Volume Systems 21.2
- Types of Vairable-Air-Volume Systems 21.2

21.2 SINGLE-ZONE VAV SYSTEMS 21.3
- System Description 21.3
- Year-Round Operation of a Single-Zone VAV System 21.5
- Economizer Cycle and Economizers 21.8
- Differential Enthalpy, Electronic Enthalpy, and Fixed Enthalpy Economizer Control 21.8
- Fixed Dry-Bulb and Differential Dry-Bulb Economizer Control 21.9
- Comparison of Enthalpy-Based and Temperature-Based Economizer Control 21.10
- Water Economizer and Control 21.12
- Comparison of Air and Water Economizers 21.14
- ASHRAE / IESNA Standard 90.1-1999 Economizer Control Specifications 21.14
- Air Conditioning Cycle and System Calculations 21.16
- Zone Temperature Control—Sequence of Operations 21.17

21.3 VAV COOLING, VAV REHEAT, AND PERIMETER HEATING VAV SYSTEMS 21.18
- VAV Cooling Systems 21.19
- VAV Reheat Systems 21.20
- System Description 21.20
- Perimeter Heating VAV System 21.20
- VAV Box 21.21
- Reheating VAV Box 21.23
- Sound Power Level of a VAV Box 21.23
- VAV Reheat Zone Temperature Control and Sequence of Operations 21.25
- Stability of Zone Control Using VAV Boxes and Reheating VAV Boxes 21.26
- Case Study: A VAV Reheat System 21.27
- Conditioned Air Off-Coil and Supply Temperature Differential 21.28
- System Volume Flow Rate and Coil Load 21.28
- Summer Cooling Mode Part-Load Operation 21.29
- Winter Reheating in Perimeter Zone 21.29
- Winter Cooling Mode Operation in Interior Zone 21.32

21.4 DUAL-DUCT VAV SYSTEMS 21.33
- System Description 21.33
- Number of Supply Fans 21.36
- Mixing VAV Box 21.36
- Mixing Mode Operation 21.38
- Zone Controls and Sequence of Operations of Dual-Fan Dual-Duct VAV System 21.38
- Discharge Air Temperature Control 21.40
- Zone Supply Volume Flow Rate 21.41
- Case Study: A Dual-Fan Dual-Duct VAV System 21.42
- Winter Heating and Winter Cooling Mode Operation 21.43
- Part-Load Operation 21.43

21.5 FAN-POWERED VAV SYSTEMS 21.44
- System Description 21.44
- Fan-Powered VAV Boxes 21.48
- Fan Characteristics in Parallel Fan-Powered Boxes 21.50
- Zone Control and Sequence of Operations of a Fan-Powered VAV System with Parallel Fan-Powered Box 21.52
- Supply Volume Flow Rate 21.53
- Fan Energy Use 21.54
- Design Considerations 21.55

21.6 COMPARISON BETWEEN VARIOUS VAV SYSTEMS 21.56

REFERENCES 21.56

21.1 SYSTEM CHARACTERISTICS OF VARIABLE-AIR-VOLUME SYSTEMS

Variable-Air-Volume Systems

A variable-air-volume (VAV) system is an air system that varies its supply air volume flow rate to match the reduction of space load during part-load operation to maintain a predetermined space parameter, usually air temperature, and to conserve fan power at reduced volume flow. A constant-volume system varies its supply air temperature to match the reduction of space load during part-load operation to maintain a predetermined space air temperature.

Compared with a constant-volume system, a VAV system has mainly the following advantages:

- Reduced fan energy use during part-load operation when the supply volume flow rate is reduced
- A slightly lower or nearly the same zone relative humidity when the supply volume flow rate is reduced during summer cooling mode part-load operation
- More individual control zones
- Reduction of the construction cost because of taking into consideration of the supply air volume flow diversity factor instead of the sum of zone peak loads
- Capability of self-balancing of zone supply volume flow rates
- Convenience during the relocation of the terminals and space diffusion devices during future expansion or retrofit

Compared with a constant-volume system, the primary disadvantages of a VAV system are:

- Inadequate outdoor ventilation air when the supply volume flow rate is reduced
- A more complicated system structure and controls, which need more demanding design, installation, operation, and maintenance

VAV systems are applicable to air systems whose space load varies significantly so that there are fan energy savings. VAV systems became popular after the energy crisis in 1973. They are widely used in large commercial buildings in the United States.

Types of Variable-Air-Volume Systems

Most medium-size and large buildings need multizone air systems. However, many indoor stadiums, convention centers, factories, residential buildings, and small retail stores employ single-zone air systems. Currently used variable-air-volume systems can be mainly classified into the following types:

- Single-zone VAV systems
- VAV cooling systems
- VAV reheat systems
- Dual-duct VAV systems
- Fan-powered VAV systems

Recently, another VAV system has been developed called the variable diffuser VAV system. In variable diffuser systems, the aperture of each diffuser can be varied so that the discharge velocity is relatively constant while the supply volume flow is reduced, and the throw from the variable diffuser may also be maintained above a certain limit. Variable diffuser VAV systems provide more individual control zones as well as a more complicated diffuser construction and controls. More field perfomance data and experience are needed to make an appropriate selection.

Conditioned spaces in buildings are primarily divided into two categories: (1) perimeter zone, where there are external wall, roofs, and windows and space load varies depending on solar heat, outdoor-indoor temperature difference and internal load; and (2) interior zone where space loads are mainly internal loads. In an interior zone, the air system often operates at cooling mode in both summer and winter except for morning warm-up. A perimeter zone can often be subdivided into four subzones according to their orientations. Although all these subzones operate at cooling mode in summer, in fall and winter, zones facing south in the northern hemisphere may need cooling, whereas zones facing north may need heating. Under such circumstances, a VAV system that serves a perimeter zone is preferable to having a modulating cooling device and a modulating heating device to meet the variable cooling and heating requirements for each of its subzones facing different orientations.

The system characteristics of different types of VAV systems vary mainly because of the zone control of their different types of VAV boxes to serve perimeter zone, except a single-zone VAV system which directly varies the fan speed or the inlet vanes and a dual-duct VAV system in which two supply ducts (a cold air duct and a warm air duct) are used. A VAV cooling system with VAV boxes usually serves an interior zone, and reheating VAV boxes or a VAV cooling system plus a perimeter heating system (such as a hot water heating system) will serve a perimeter zone. A single-zone VAV system can be used for either a perimeter zone or an interior zone. Excluding the terminals and zone controls, the following system characteristics of different VAV systems are similar to each other, and are discussed in later chapters.

- Fan combination
- System pressure analysis and space pressurization
- Smoke control
- Outdoor ventilation air
- Controls, including discharge temperature, duct static pressure, and safety controls

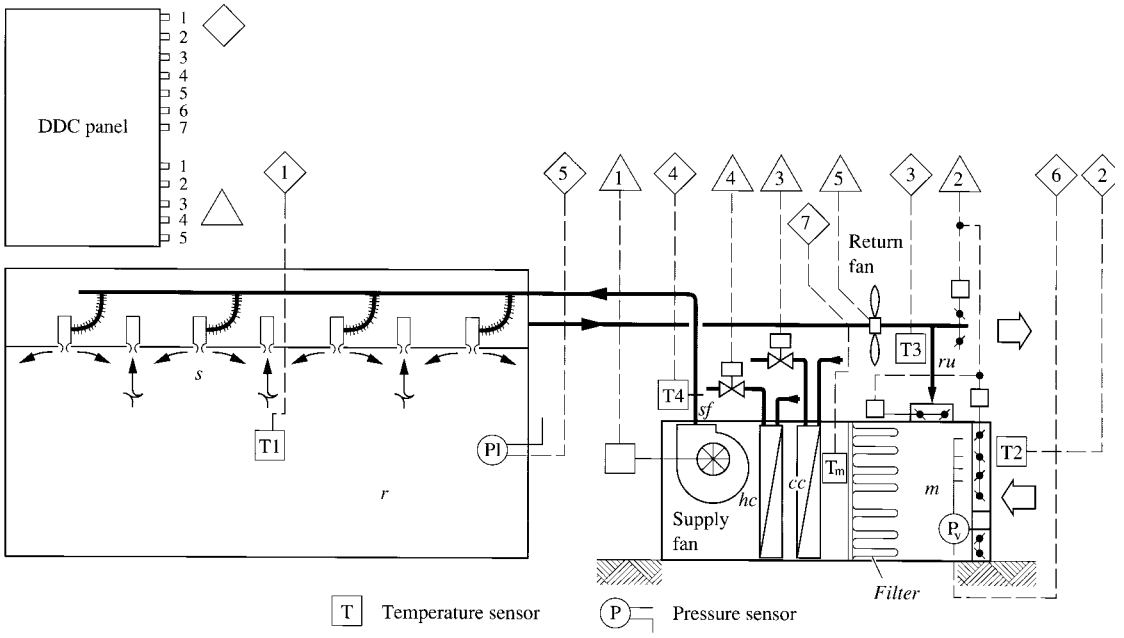
ASHRAE/IESNA Standard 90.1-1999, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* specifies that individual variable-air-volume (VAV) fans with 30-hp (22-kW) motors and larger shall have other controls and devices (such as adjustable-frequency, variable-speed drives) that will result in fan motor demand of no more than 30 percent of design wattage at 50 percent of design volume flow rate when fan static pressure equals one-third of the total design static pressure based on manufacturer's certified data.

21.2 SINGLE-ZONE VAV SYSTEMS

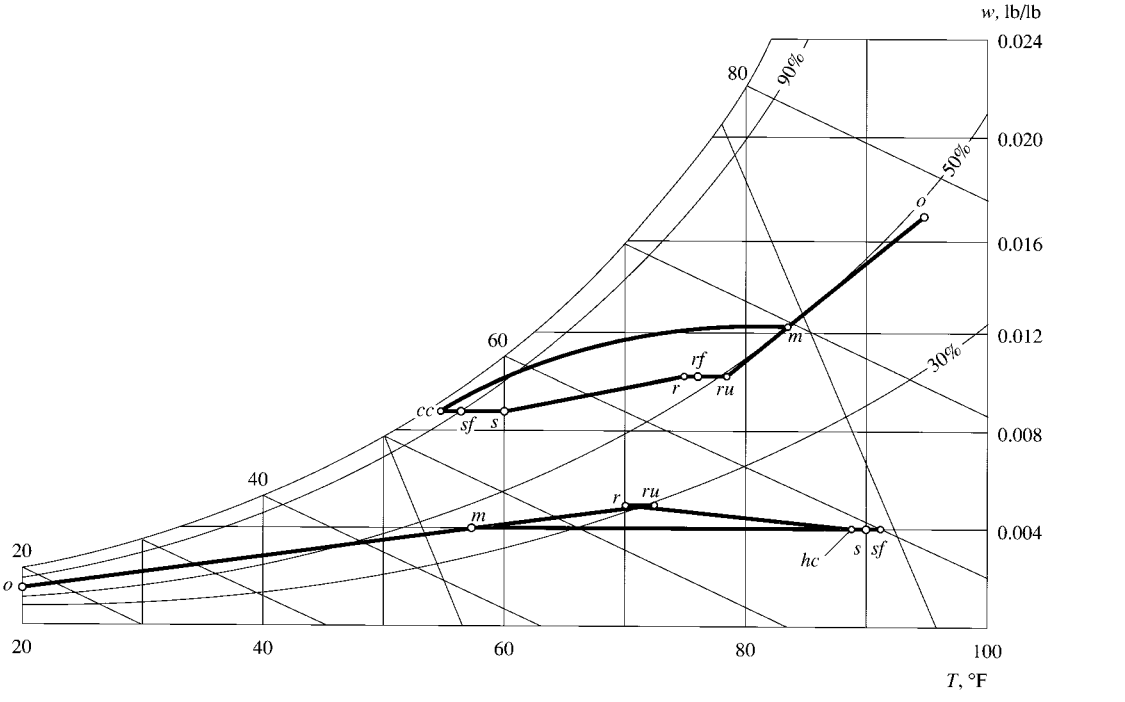
A single-zone VAV system is a VAV system that varies the air volume flow rates supplied to and returned from a single-zone conditioned space to maintain a predetermined space parameter at reduced load and to conserve fan power. Single-zone systems are widely used in indoor stadiums, assembly halls, shopping malls, and factories.

System Description

Figure 21.1 shows the schematic diagram and the air conditioning cycles for a typical single-zone VAV system. During the summer cooling mode operation, the mixture of outdoor air and recirculating air at point *m* is filtered at the filter and cooled and dehumidified at the cooling coil. The conditioned air leaves the coil at point *cc*. It then flows through the supply fan of the air-handling unit (AHU) or packaged unit (PU) and the supply duct before it is discharged to the single-zone conditioned space at point *s*. After absorbing the space cooling load, the supply air becomes the space air *r* and returns to the AHU or PU through the return air system.



(a)



(b)

FIGURE 21.1 A single-zone VAV system: (*a*) schematic diagram; (*b*) air conditioning cycle.

When air economizer cycles are used for free cooling during spring and fall seasons, outdoor air intake may vary from minimal to 100 percent. A relief fan should be installed in combination with the supply fan, if the pressure drop of the return duct system is small. If the pressure drop of the return duct system is greater than 0.6 in. WC (149 Pa), a return fan in series with a supply fan should be considered.

During summer cooling mode part-load operation, the supply and return volume flow rates are reduced to match the lower space cooling load. The temperature rise from the fan power heat gain is reduced accordingly. At the same time, duct heat gain remains approximately the same, which results in a higher temperature rise for supply and return ducts during summer cooling mode part-load operation.

During winter, if the solar heat and internal load is greater than the space transmission and infiltration loss, then a cooling mode operation of a cold air supply is still needed, such as in fully occupied indoor stadium and arenas. Outdoor air is often mixed with the recirculating air to form a mixture at a temperature around 55°F (12.8°C) during the air economizer cycle for free cooling. Warm air may be needed only during the morning warm-up period to raise the space temperature to an acceptable value before the space is to be occupied.

If solar heat and the internal load are less than the transmission and infiltration loss, the mixture of outdoor and recirculating air is heated at the heating coil to point *hc*. It is extracted by the supply fan and forced through the supply duct. Warm air is then supplied to the conditioned space at point *s*, and the recirculating air is returned to the AHU or PU through the ceiling plenum and connecting return ducts.

For comfort air conditioning without humidification in winter, the space relative humidity depends mainly on the space latent load and the volume flow ratio of outdoor air to supply air.

For many VAV systems in commercial buildings, the ceiling plenum is often used as the return plenum. The primary sound transmission path of supply fan noise from the AHU or PU to the space is often duct-borne noise through the return system. A sound trap or silencer should be added in the fan room between the AHU or PU and the connecting duct before penetrating the fan room partition wall.

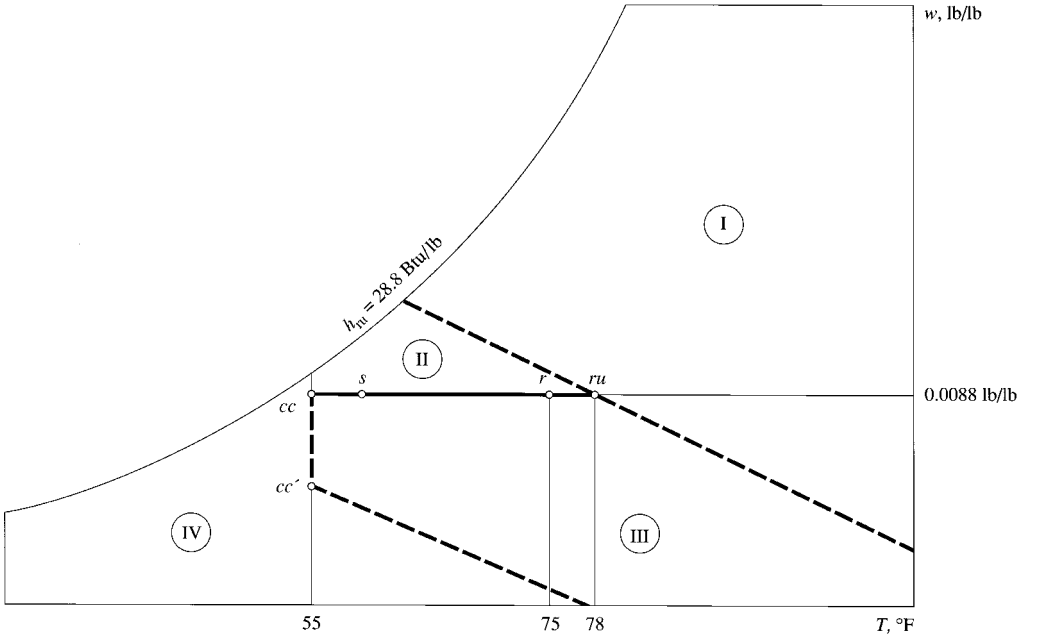
Year-Round Operation of a Single-Zone VAV System

In this single-zone VAV system, cooling is provided by a refrigerant plant and evaporative coolers, and heating is provided by the hot water boiler through a hot water heating coil. For convenience in analysis, consider an interior zone with zone air temperature maintained at 75°F (23.9°C) year-round. During cooling mode operation, the required off-coil temperature is 55°F (12.8°C) with a relative humidity of 95 percent. The recirculating air enters the mixing box at 78°F (25.5°C). It is mixed with an amount of outdoor air approximately equal to 20 percent of the total design supply volume flow rate and is cooled and dehumidified in a cooling coil. During winter heating mode operation, cold air supply is still required to maintain a year-round space temperature of 75°F (23.9°C); outdoor air and recirculating air are mixed together to provide a mixture of 55°F (12.8°C). It is also assumed that the building is in a location where no humidification is required to maintain a space relative humidity no lower than 30 percent during the winter season.

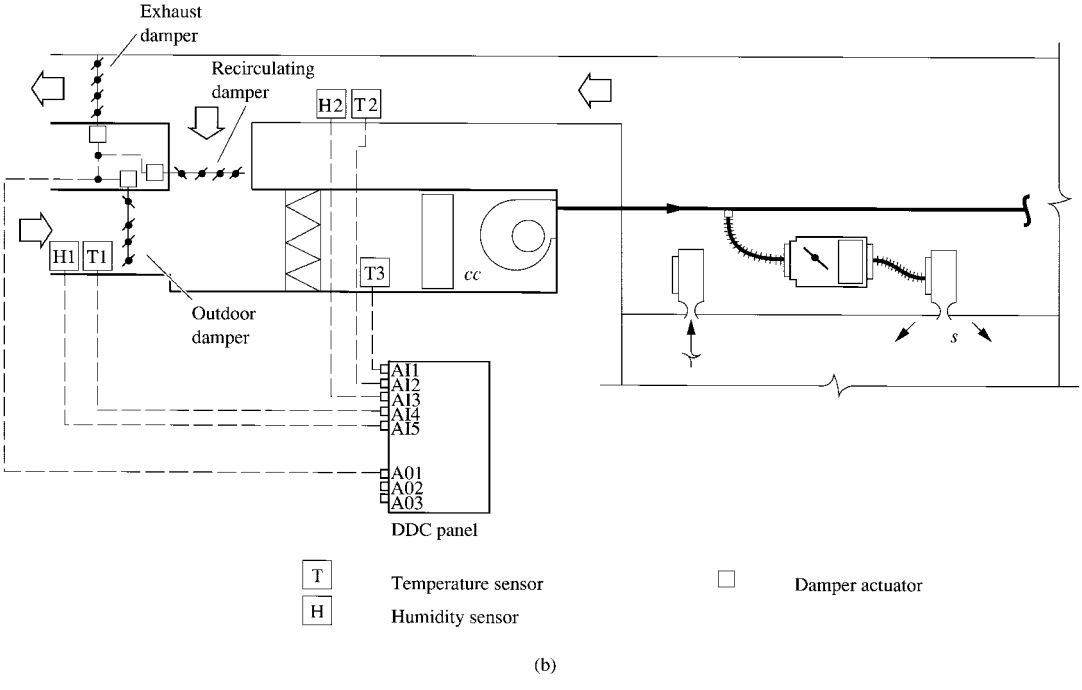
To achieve effective and energy-efficient operation when the outside weather changes from hot summer to cold winter, the year-round operation of a single-zone VAV air system described above can be divided into the following operating modes, corresponding to the various outside weather regions on the psychrometric chart, as shown in Fig. 21.2:

Region I: Refrigeration and Evaporative Cooling. The lower left boundary line of region I is the (dashed) enthalpy line that passes through the state point *ru* of recirculating air entering the mixing box. When the outside weather at point *o* is inside region I, the enthalpy of the outside air is higher than that of the recirculating air, that is, $h_o \geq h_{ru}$. In these instances, it is more energy-efficient to condition (cool and dehumidify) the recirculating air than the outside air. As such, the quantity of outdoor air mixed with the recirculating air will be the minimum needed to meet the required outdoor ventilation rate in the conditioned space.

In region I, refrigeration or evaporative cooling or both are required to cool and dehumidify or to cool and humidify the mixture of outdoor and recirculating air to point *cc* in order to maintain the predetermined space condition *r*.



(a)



(b)

FIGURE 21.2 Enthalpy economizer control: (*a*) outside weather regions; (*b*) enthalpy economizer.

For region I, the operating characteristics of the VAV system include the following:

- The outdoor damper is opened to such a degree that only that minimum required amount of outdoor air is extracted.
- The recirculating damper is fully opened.
- Supply air is cooled and dehumidified, or cooled and humidified to point *cc*.
- A refrigeration system, evaporative coolers, or both are operating.

Region II: Free Cooling and Refrigeration/Evaporative Cooling. Region II is enclosed by four boundary lines: the enthalpy line through point *ru* at the upper right, part of the saturation curve, the 55°F (12.8°C) temperature line, and the *cc*-*ru* line.

In region II, the enthalpy of outdoor air is lower than that of recirculating air, that is, $h_o < h_{ru}$. Less energy will be used in conditioning if 100 percent of outside air is cooled and dehumidified instead of a mixture outdoor air and recirculating air. Both free cooling by the outside air and refrigeration/indirect evaporative cooling (to be discussed in Chap. 27) are required to bring the 100 percent outside air down to 55°F (12.8°C) with a relative humidity of 95 percent.

The operating characteristics of this VAV system when outdoor air is inside region II include the following:

- The outdoor damper and the exhaust damper or relief damper both are fully opened.
- The recirculating damper is closed.
- Outdoor air is cooled and dehumidified to point *cc*. Point *cc* is at a temperature of 55°F (12.8°C) and a humidity ratio of 0.0088 lb/lb (kg/kg).
- Refrigeration system/and indirect evaporative cooler are operating.

Region III: Free Cooling and Evaporative Cooling/Refrigeration. Region III is bounded by the line *cc*-*ru*, 55°F (12.8°C) temperature lines, and enthalpy lines passing through points *ru* and *cc′* for $w_{cc'} = 0.005$ lb/lb (kg/kg), as shown in Fig. 21.2. For outdoor air inside this region, $h_o < h_{ru}$, the humidity ratio of outdoor air w_o is approximately less than 0.009 lb/lb (kg/kg). Indirect and direct evaporative cooling plus necessary refrigeration is required to maintain a space temperature of 75°F (23.8°C) and a space relative humidity greater than or equal to 30 percent.

When outdoor temperature $T_o \leq 78$°F (25.5°C), it is energy-efficient to extract 100 percent outdoor air (to make use of free cooling) and to use direct and indirect evaporative cooling plus necessary refrigeration to cool and humidify the outdoor air to a temperature of 55°F (12.8°C) with a humidity ratio $w_{cc'} \geq 0.005$ lb/lb (kg/kg). When $T_o > 78$°F (25.5°C), it is energy-efficient to extract minimum outdoor air and mix it with recirculating air. The mixture is then evaporatively cooled indirectly-directly with neccessary refrigeration to $T_{cc} = 55$°F (12.8°C) and $w_{cc'} = 0.005$ lb/lb (kg/kg).

The operating characteristics of the single-zone VAV system when outdoor air is inside region III and when 100 percent outdoor air is extracted are as follows:

- The outdoor air damper and the exhaust or relief air damper both are fully opened.
- The recirculating damper is closed.
- Evaporative coolers are used for evaporative cooling and humidifying.
- Necessary refrigeration is required.

When only a minimum of outdoor air is mixed with recirculating air, then

- The outdoor air damper is opened to minimum.
- The recirculating damper is fully opened.
- Evaporative coolers are used for evaporative cooling and humidifying.
- Sometimes refrigeration may be required.

Region IV: Free Cooling, Humidification, and Heating. Region IV is enclosed by part of the saturation curve, the 55°F (12.8°C) temperature line, and the enthalpy line passing through point cc' for $w_{cc'} = 0.005$ lb/lb (kg/kg), as shown in Fig. 21.2. For outdoor air state points in region IV, either $T_o <$ 55°F (12.8°C) and $h_o > h_{cc'}$ or $T_o <$ 78°F (25.5°C) and $h_o \leq h_{cc'}$, when $w_{cc'} = 0.005$ lb/lb (kg/kg).

In modulation of outdoor and recirculating dampers, or in locations with very cold winter, heating may be required to provide a mixed air temperature of 55°F (12.8°C) when the volume flow of outdoor air is equal to the minimum requirement. Free cooling of outdoor air will be utilized. Optimum humidifying methods should be adopted to maintain a space relative humidity of preferably not less than 25 percent.

In locations where the outdoor temperature may drop below 32°F (0°C) in winter, remedies should be provided to protect water from freezing.

The operating characteristics of this VAV system include the following:

- Outdoor, exhaust, and recirculating air dampers are modulated.
- Heating, humidification, or cooling and humidification may be required.
- The refrigeration system is not operating.

Year-round operation of a multizone VAV system having perimeter and interior zones, is similar to that for an interior zone VAV system with the exception that heating may be required in regions II, III, and IV. For a multizone VAV system, the space air temperature can be considered as the average value of multiple zones.

Economizer Cycle and Economizers

An economizer cycle is an air conditioning cycle that utilizes the free cooling capacity of outdoor air either directly or to cool the condenser water in a cooling tower (or an evaporative cooler), and then to cool the air indirectly, instead of using refrigeration to provide cooling/dehumidification so as to maintain a required space temperature. The component and devices used in the operation of an economizer cycle are collectively called an economizer, and the type of control used to operate the economizer cycle effectively and energy-efficiently is called economizer control.

There are two types of economizers:

1. *Air or air-side economizers*. These bring in up to 100 percent outdoor air to utilize its free cooling capacity, rather than use refrigeration or evaporative cooling, to offset the space cooling load when the enthalpy of the outdoor air is lower than the enthalpy of the recirculating air, or the outdoor air temperature is lower than a preset air temperature or a set point which is a function of outdoor climate.

An air economizer consists of outdoor, exhaust, relief, and recirculating ducts and dampers in the AHU or PU, as well as a control system to operate them (see Fig. 21.2*b*). Air economizer control can be subdivided into enthalpy-based differential enthalpy, fixed enthalpy, and electronic enthalpy economizer controls and also temperature-based fixed dry-bulb and differential dry-bulb economizer controls.

2. *Water or water-side economizers*. These economizers use the outdoor air to cool the condenser water or cooling water in the cooling tower or evaporative cooler first, and then to cool the the mixture of outdoor and recirculating air through a precooling coil. A water economizer consists mainly of a cooling tower (or an evaporative cooler), a water precooling coil in the AHU or PU, a circulating pump to circulate cooling water, and the associated control system.

Differential Enthalpy, Electronic Enthalpy, and Fixed Enthalpy Economizer Control

The operation of an enthalpy-based economizer control system is based on the comparison of air enthalpies or the comparison of air enthalpies and air temperatures. In a differential enthalpy

economizer, the on-and-off air economizer is based on the comparison of the enthalpy of the outdoor air h_o, in Btu/lb (kJ/kg), with the enthalpy of the recirulating air h_{ru}, in Btu/lb (kJ/kg). In cooling mode operation, if $h_o \leq h_{ru}$, the differential enthalpy economizer is turned on. If $h_o > h_{ru}$, the differential enthalpy economizer is shut off. Enthalpies of outdoor air and recirculating air can be calculated from their measured dry-bulb and wet-bulb temperatures, or from their measured dew points.

The sequence of operations of a typical differential enthapy economizer control system based on measured dry- and wet-bulb temperature of the outdoor and recirculating air is as follows:

1. The outdoor air temperature T_o is sensed by sensor T1, and the relative humidity of outdoor air φ_o is sensed by humidity sensor H1. Both signals are sent to a DDC controller, where calculations are performed to combine the signals into an outdoor air enthalpy h_o signal. This signal is compared with the recirculating air enthalpy h_{ru}, which is the combined signal from temperature T2 and humidity sensor H2, at the DDC controller.

2. If $h_o > h_{ru}$, the DDC controller sends signals to actuate the damper actuators so that the outdoor air and exhaust air dampers will be at their minimum opening positions, as determined by the manual position switch. Meanwhile, the recirculating damper is fully opened.

3. If $h_o \leq h_{ru}$, the DDC controller sends signals to actuate the damper actuators to fully open the outdoor air and exhaust air dampers; at the same time, the recirculating damper will be closed.

4. Temperature sensor T3 senses the mixed air temperature of outdoor and recirculating air T_m for airflow leaving the mixing box. This signal is compared with the set point $T_{m,s} = 55°F$ (12.8°C) in the DDC controller. When the outdoor air temperature $T_o \leq 55°F$ (12.8°C), under the condition that the opening of the outdoor damper is equal to or greater than its minimum opening position, the DDC controller then actuates the damper actuator to modulate the opening of the outdoor, exhaust, and recirculating air dampers to maintain a predetermined mixed air temperature of $T_m =$ 55°F (12.8°C). When $T_o \leq 55°F$ (12.8°C), the refrigeration compressors or chillers that serve this air system can be shut down.

In an electronic enthalpy economizer control, outdoor air is compared with a set point *A* which can be represented on the psychrometric chart by a curve that goes through a point at approximately 75°F (23.9°C) and 40 percent relative humidity. This curve is nearly parallel to dry-bulb lines at low humidity level and nearly parallel to enthalpy lines at high humidity levels. In a fixed enthalpy economizer control, outdoor air is often compared with a fixed enthalpy h_{oA} of 75°F (23.9°C) and 50 percent relative humidity with an enthalpy of 28 Btu/lb (65 kJ/kg) at sea level. In cooling mode operation, if $h_o > 28$ Btu/lb (65 kJ/kg) at sea level, the fixed enthalpy economizer shall be shut off. If $h \leq 28$ Btu/lb (65 kJ/kg), the fixed enthalpy economizer shall be turned on.

Fixed Dry-Bulb and Differential Dry-Bulb Economizer Control

There are two temperature-based economizer controls: fixed dry bulb and differential dry bulb. In a fixed dry-bulb economizer control, outdoor air temperature T_o is compared with a fixed dry-bulb temperature T_{db}. ASHRAE/IESNA Standard 90.1-1999 recommends the following fixed dry bulbs:

Climate	Fixed dry bulb	High-limit shutoff
Dry	75 to 78°F (23.9 to 25.6°C)	$T_o > 75$ to 78°F
Intermediate	70°F (21.1°C)	$T_o > 70$°F
Humid	65°F (18.3°C)	$T_o > 65$°F

From the psychrometric chart, the enthalpies of 75°F, 50 percent relative humidity, 70°F, 64 percent relative humidity, and 65°F, 85 percent relative humidity are all near to the value 28 Btu/lb (65 kJ/kg). In a differential dry-bulb economizer control, outdoor air temperature T_o is compared with the fixed recirculating temperature T_{ru}. In cooling mode operation, if $T_o \leq T_{ru}$, the differential

dry-bulb economizer control is turned on. If $T_o > T_{ru}$, the differential dry-bulb economizer control is shut off. For a space temperature $T_r = 75°F$ (23.9°C) in summer, T_{ru} is usually equal to 75 + 3 = 78°F (25.6°C) after absorbing the lighting sensible heat in the return ceiling plenum.

Theoretically speaking, it is more energy-efficient to compare the enthalpy of outdoor air with that of recirculating air during air economizer control. However, comparing enthalpies necessitates having both temperature and humidity sensors or temperature and dew point sensors. In actual practice, humidity sensors may demonstrate considerable errors (sometimes up to 10 percent) and have extensive maintenance requirements. Dew point sensors are delicate and expensive and cause maintenance difficulties. Therefore, it is simpler and more convenient to use only temperature sensors and to compare the outdoor air temperature T_o with the recirculating temperature T_{ru} (or a predetermined set point) instead of sensing and comparing enthalpies. This method of control is called temperature economizer control. Outside weather regions using temperature-based economizer control are shown in Fig. 21.3*a* and are similar to the outside weather regions for enthalpy economizer control.

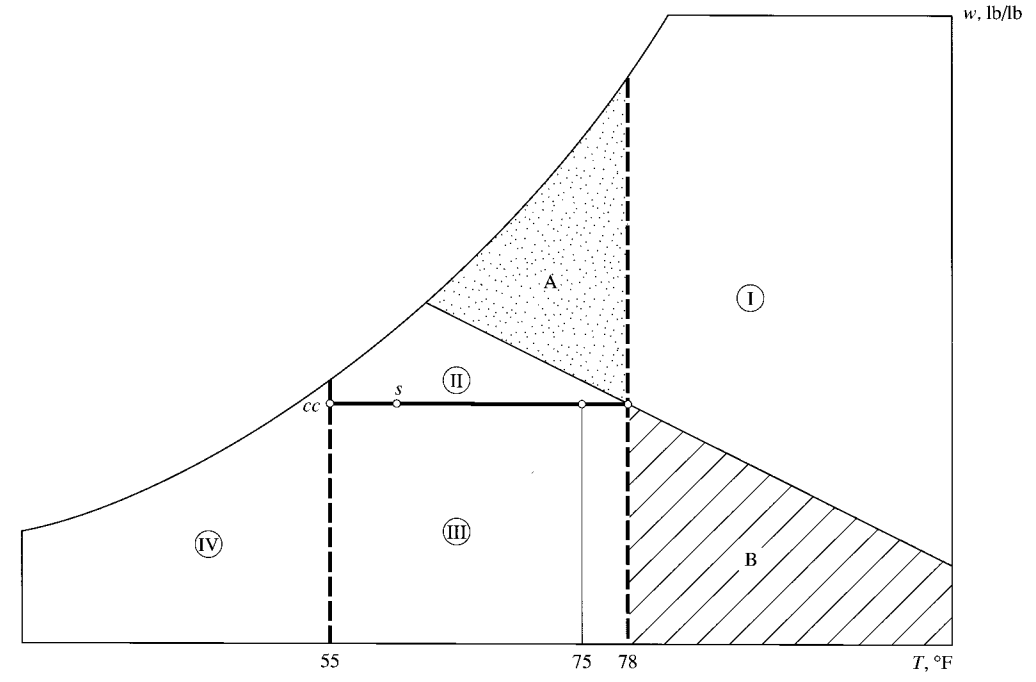
The sequence of operations of a typical differential dry-bulb economizer control system is as follows:

1. The outdoor air temperature T_o is sensed by temperature sensor T1, as shown in Fig. 21.3*b*. Then T_o is compared with the recirculating temperature T_{ru} sensed by temperature sensor T2. If $T_o > T_{ru}$, the DDC controller sends signals to damper actuators so that the outdoor and exhaust air dampers will be at their minimum opening, as determined by the manual positioning switch. The recirculating damper should be fully opened.
2. If $T_o \leq T_{ru}$, the outdoor and exhaust air dampers should be fully opened and the recirculating damper should be closed.
3. Temperature sensor T3 senses the mixed air temperature T_m after the mixing box. When the outdoor air temperature $T_o \leq 55°F$ (12.8°C), under the condition that the volume flow of outdoor air $\dot{V}_o$ equal or be greater than the minimum requirement, the DDC controller modulates the outdoor, exhaust, and recirculating air dampers to maintain a nearly constant mixed air temperature of 55°F (12.8°C).
4. When $T_o \leq 55°F$ (12.8°C), the refrigeration system used to serve this single-zone VAV air system is shut off.

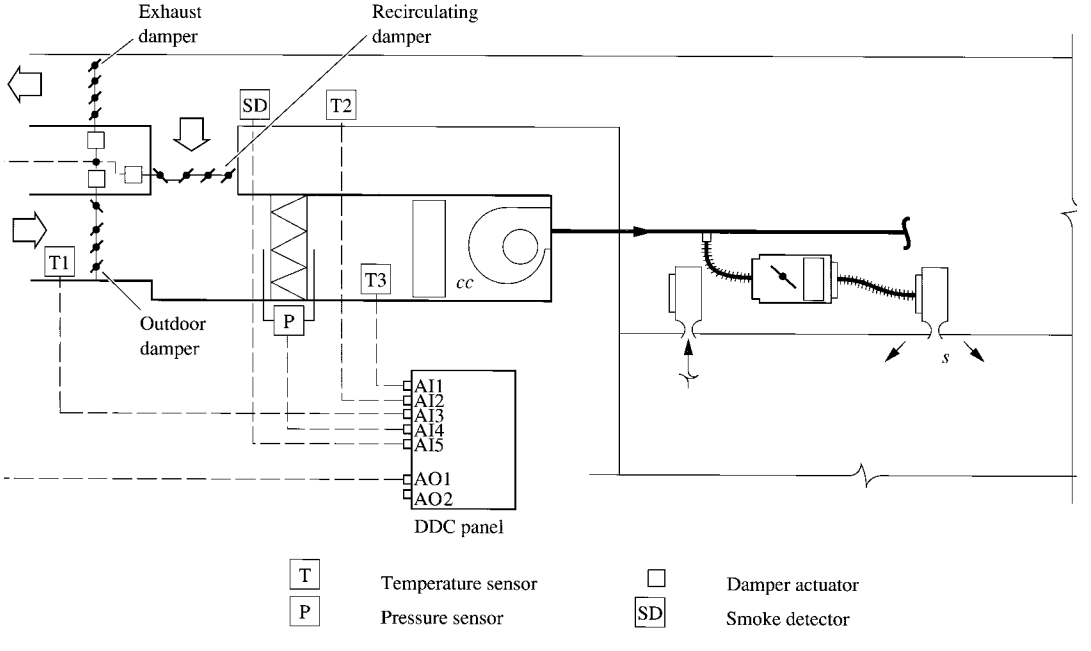
In a fixed dry-bulb economizer control, a single dry-bulb temperature set point T_{db}, may be used as an indicator of when to change from using the minimum amount of outdoor air to using 100 percent outdoor air. The changeover is determined from a comparison of T_o with T_{db}. The proper magnitude of T_{db} depends on the outside climate and the characteristics of the air system as recommended by ASHRAE/IESNA Standard 90.1-1999. According to Wacker (1989), an investigation is required to determine the optimum T_{db}.

Comparison of Enthalpy-Based and Temperature-Based Economizer Controls

In addition to the instrumentation and maintenance problems of enthalpy economizer control, enthalpy-based and temperature-based economizer controls are differentiated by the location of the status points of outside weather in the dotted area *A* or shaded area *B* shown in Fig. 21.3*a*. When the outside weather is inside the dotted area *A*, differential-enthalpy economizer control using minimum outdoor ventilation air recirculation mode often consumes less energy to cool and dehumidify the mixture of outdoor and recirculating air than with the 100 percent outdoor air mode. When the outside weather falls in the shaded area *B*, a differential dry-bulb economizer control using minimum outdoor ventilation air recirculating mode may sensibly cool the mixture at a smaller enthalpy differential than a 100 percent outdoor air intake in a differential enthalpy economizer control.



(a)



(b)

FIGURE 21.3 Temperature economizer control: (*a*) outside weather regions; (*b*) temperature economizer.

Consider a single-zone VAV system that serves a building located in Phoenix, Arizona. Assume that the outdoor temperature is 85°F (29.4°C), the humidity ratio is 0.006 lb/lb (kg/kg), and the conditioned air leaves the coil at a temperature of 55°F (12.8°C), with a relative humidity of 95 percent. The recirculating air enters the AHU or PU at a temperature of 78°F (25.5°C) and a humidity ratio of 0.0094 lb/lb (kg/kg). If a differential enthalpy economizer control is used, the state point of outdoor air o_h falls in region III, and if a differential dry-bulb economizer control is used, the state point of outdoor air o_T falls in region I. The properties of moist air are as follows:

	Differential enthalpy		Differential dry bulb			
	Outdoor	Off-coil	Recirculating	Outdoor	Mixture	Off-coil
Temperature, °F	85	55	78	85	79.5	55
Relative humidity, %	24	65	45	24	40	90
Humidity ratio, lb/lb	0.006	0.006	0.0094	0.006	0.0086	0.0086
Enthalpy, Btu/lb	27.0	19.8			28.6	22.4
Enthalpy difference, Btu/lb	7.2				6.2	
Amount of outdoor air, %	100				20	

A differential dry-bulb economizer control is required to cool the mixture of outdoor and recirculating air down to an enthalpy difference of 6.2 Btu/lb (14.4 kJ/kg) instead of 7.2 Btu/lb (16.7 kJ/kg) using a differential enthalpy economizer control.

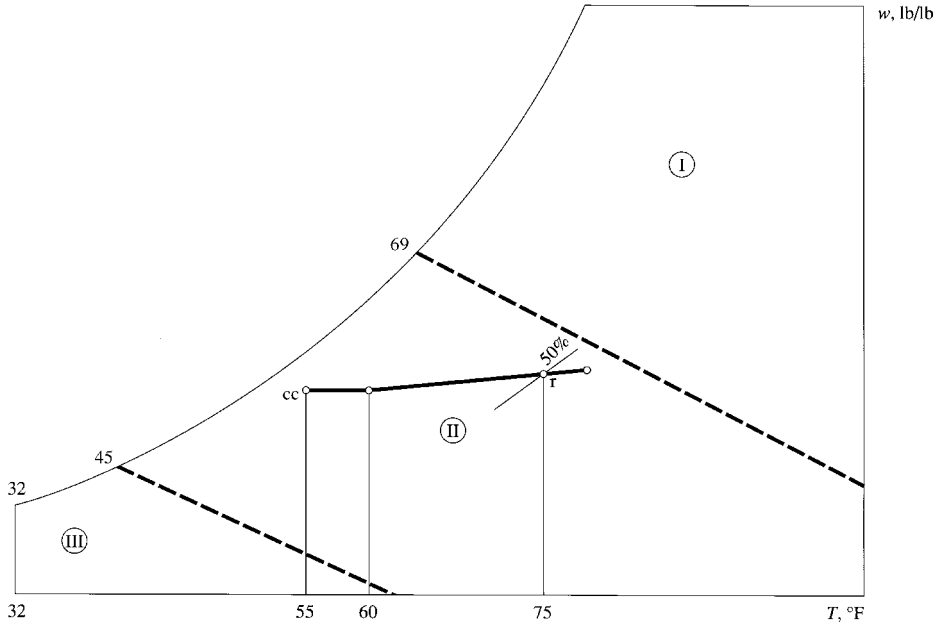
Therefore, consider the following:

1. For locations having hot and humid climates where many status points of outdoor air fall in the dotted area *A* in Fig. 21.3*a*, a differential enthalpy economizer control is recommended.
2. For most locations having moderate outdoor humidity ratios, according to Spitler et al. (1987), the energy savings offered by using a differential enthalpy economizer control instead of a temperature-based economizer are small. The two types of economizer control may differ by only about 5 to 10 percent. A fixed dry-bulb economizer control is recommended.
3. For locations having dry climates, a fixed dry-bulb of economizer control with a temperature setting of 75 to 78°F (23.9 to 25.1°C) sometimes even saves more cooling energy than a differential enthalpy economizer control. A fixed dry-bulb economizer control is strongly recommended.
4. For locations with humid climates, where the space requires a lower relative humidity, a fixed dry-bulb economizer control using a set point T_{db} a few degrees Farenheit lower than space temperature sometimes is beneficial.
5. A survey including the simulation and comparison of various air economizer control alternatives is often beneficial.

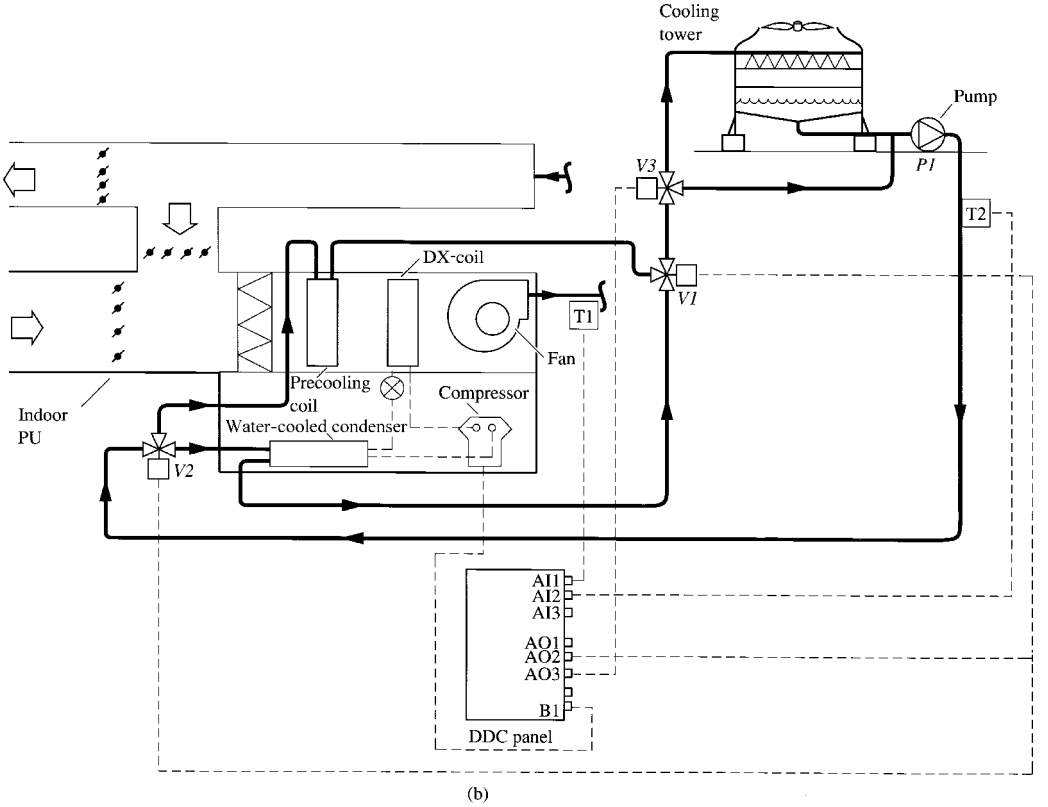
Water Economizer and Control

Consider a typical water economizer that uses a precooling coil in either an AHU or PU, as shown in Fig. 21.4*b*, and has the following operating characteristics:

- The cooling tower approach is assumed to be 5°F (2.8°C).
- The mixing temperature of outdoor air and recirculating air T_m, is 78°F (25.5°C).
- The minimum temperature difference between T_m and the condenser water temperature $T_{con,w}$ is 4°F (2.2°C).
- When the outdoor wet-bulb temperature $T'_o \leq 40$°F (4.4°C), the temperature of the condenser water used to replace the entire refrigeration through a heat exchanger is less than 45°F (7.2°C).



(a)



(b)

FIGURE 21.4 Water economizer control: (*a*) outside weather regions; (*b*) water economizer of an indoor PU.

- The outdoor wet-bulb temperature $T'_{o,we}$ below which the water economizer can be operated effectively is

$$T'_{o,we} = 78 - 5 - 4 = 69°\text{F (20.6°C)}$$

During the operation of a water economizer, the outside weather can be grouped into three regions based on the outdoor wet-bulb temperature, as shown in Fig. 21.4*a*:

Region I: The water economizer cannot be operated effectively.

Region II: The water economizer is operating effectively and replaces part of the refrigeration.

Region III: The water economizer replaces the refrigeration up to 100 percent.

The dividing line between regions I and II is the wet-bulb $T'_{o,we}$ line. The dividing line between regions II and III is the wet-bulb $T'_{r,of}$ line, where $T'_{r,of}$ represents the outdoor wet-bulb temperature below which the refrigeration plant serving the air system installed with a water economizer can be shut down.

In locations where the outdoor climate is cold, and when the temperature discharged from the cooling tower drops below 35°F (1.7° C), the water economizer should be shut down for freeze protection.

Comparison of Air and Water Economizers

The economizer cycle is a popular means of saving energy in the year-round operation of air conditioning systems. The amount of energy savings, compared with air systems for which the economizer cycle is not adopted, ranges from 15 to 40 percent of total energy use in various locations.

The advantages of air economizers over water economizers include the following:

1. Because of the intake of a large amount of outdoor air, the indoor air quality is improved.
2. The energy input to an air conditioning system that uses an air economizer is approximately 20 percent less than that for a water economizer.
3. The cooling tower requires less makeup water, fewer water treatments, and reduced maintenance.

However, the disadvantages of air economizers are as follows:

1. For an AHU or PU located in the interior core fan room or machinery room and used to serve several floors in a high-rise building, the large vertical shafts used to transport the outdoor air and exhaust air often occupy a lot of valuable rental space.
2. When outdoor weather suddenly changes, the space air may also produce corresponding fluctuations. Building pressurization is not as stable as with a water economizer.
3. During the winter season, the intake of drier outdoor air always results in a lower space relative humidity than is experienced using recirculating air. Humidification may be required.
4. For packaged units, the first cost of an air economizer is about 25 to 40 percent of the installation cost of the packaged unit. For a water economizer, the first cost is about 10 percent of the installation cost of the PU.
5. Coil freeze protection is more critical in locations where the outdoor temperature drops below 32°F (0°C).

ASHRAE / IESNA Standard 90.1-1999 Economizer Control Specifications

For the use of free cooling of outdoor air to reduce refrigeration, to improve indoor air quality, and to save energy use in HVAC&R systems ASHRAE/IESNA Standard 90.1-1999 specified the following:

Installation of Economizers. Each air system that has a fan and cooling capacity equal to or greater than that listed below shall include an air or water economizer and meet the following requirements:

No. of hours between 8 a.m and 4 p.m. with $55°F < T_{db} < 69°F$	$T_{wb} < 69°F$, Btu/h	$69°F \leq T_{wb} \leq 73°F$, Btu/h	$T_{wb} > 73°F$, Btu/h
200 – 599	135,000	N.R.	N.R.
600 – 799	65,000	135,000	N.R.
800 – 999	65,000	135,000	135,000
1000 – 1199	65,000	65,000	135,000
> 1199	65,000	65,000	65,000

Here, N. R. indicates "not required," T_{db} the dry-bulb temperature, and T_{wb} the wet-bulb temperature. Exceptions include:

- Air system used for the purpose of outdoor ventilation air cleaning to meet ASHRAE Standard 62 requirement.
- Where more than 25 percent of the supply air is used for space humidification at above 35°F (1.7°C) dew point to satisfy process needs.
- Air systems that include a condenser heat recovery system.
- Air systems that serve residential spaces, or serve spaces whose sensible cooling load is rather small.
- Air systems expected to operate less than 20 hours per week.
- Where the use of outdoor air will affect supermarket open refrigerated display cases.
- When the cooling efficiency of the packaged unit or heat pump meet the following increasing requirements:

System Size, kBtu/h Q_{rc}	Mandatory Minimum EER	Cooling Degree Days (CDD50)			
		3601–5400	5401–7200	7201–9000	9001–10,800
$65 \leq Q_{rc} < 135$	10.1	12.1	11.6	11.1	10.7
$135 \leq Q_{rc} \leq 240$	9.3	11.3	10.8	10.4	9.9
$240 \leq Q_{rc} < 760$	9.0	10.9	10.5	10.0	9.6

For other packaged systems, when mandatory minimum EERs varies from 10.3 to 9.0, the corresponding minimum increased cooling efficiencies (EER) required are from 12.5 to 9.9. For details, refer to ASHRAE/IESNA Standard 90.1-1999.

Design and controls of economizers shall be such that economizer operation does not increase the building heating energy use during normal operation except allowable reheating, recooling, and mixing by Standard 90.1-1999 (refer to Sec. 20.20).

Air Economizers. Standard 90.1-1999 specifies the following for air economizers:

- *Capacity control*. Air economizers shall be capable of modulating outdoor air and return air dampers to provide up to 100 percent of outdoor air for cooling. Air economizers shall be capable of automatically reducing outdoor air intake to the required minimum outdoor ventilation air specified by ASHRAE Standard 62 when outdoor air intake is no longer reducing cooling energy usage. Operators of air economizers using various types of temperature-based or enthalpy-based economizer controls shall be aware that for dry climates (shaded area *B* in Fig. 21.3*a*) minimum outdoor ventilation air (required by ASHRAE Standard 62) recirculating mode often needs less cooling energy than 100 percent outdoor air economizer operation. The 100 percent air economizer operation shall be shut off. Similarly, for humid climates (dotted area *A* in Fig. 21.3*a*) in which minimum

outdoor ventilation air recirculating mode often needs less cooling energy than 100 percent outdoor air economizer control and the air economizer operation shall also be shut off.

- *Sequence control*. Economizer dampers shall be in sequence with the refrigeration and shall not be controlled only by mixed air temperature except for single-zone air systems.
- *Damper leakage*. Both recirculating and outdoor air dampers shall have a maximum leakage rate of 20 cfm/ft^2 (100 L/s · m^2) at a pressure difference of 4 in. WC (1000 Pa).
- *Relief of excess outdoor air*. Means to relieve excess outdoor air during air economizer operation shall be provided to prevent overpressurization of the building. The relief outlet shall be located to avoid recirculation into the building.

Water Economizers. Water economizers shall be capable of cooling supply air through indirect evaporation and providing up to 100 percent of the system cooling coil load (precooling coil) at an outdoor air temperature of 50°F (10°C) dry bulb/45°F (7.2°C) wet bulb and below [except when the dehumidification requirement cannot be met at these outdoor dry- and wet-bulb temperatures and the 100 percent system cooling (precooling) coil load must be satisfied at outdoor air temperatures 45°F (7.2°C) dry bulb/40°F (4.4°C) wet bulb].

Precooling coils and water-to-water heat exchangers as components of the water economizer shall either have a water-side pressure drop of less than 15 ft (44.7 kPa), or using a secondary water loop so that the pressure drop of the coil or heat exchanger is not taken care of by the circulating pumps when the system is in normal cooling (noneconomizer) mode.

Air systems with water cooling and humidification to maintain space humidity at greater than 35°F (1.7°C) dew point temperature shall use a water economizer if an economizer is required because the air economizer needs to extract outdoor air at lower temperature and humidity ratios and thus increases the humidifying load.

Integrated Economizers Control. Economizers shall be integrated with refrigeration and can provide partial cooling and even additional refrigeration if required to meet the remaining cooling coil load, except the following:

- Direct-expansion systems that control and reduce the required quantity of outdoor air to prevent coil frosting at the lowest step of compressor unloading if this lowest step is no greater than 25 percent of the total system capacity.
- Individual packaged unit has a capacity less than 65,000 Btu/h (19 kW) and uses nonintegrated economizer controls.
- System in locations with less than 800 average hours annually between 8 a.m. and 4 p.m. when the ambient (outdoor) dry-bulb temperatures are between 55°F (12.8°C) and 69°F (20.6°C) inclusive.

Air Conditioning Cycle and System Calculations

The air conditioning cycle of a single-zone VAV system during cooling mode operation at summer design conditions and heating mode operation at winter design conditions without space humidity control can be plotted on the psychrometric chart as shown in Fig. 21.1*b* During summer cooling mode part-load operation, the state point of outdoor air *o* varies, and the sensible ratio of the space conditioning line SHR_{sp} usually becomes smaller than at design load. If the zone temperature T_r is maintained at the same predetermined limit as in design load, the zone relative humidity will be slightly greater than that in design load if SHR_{sp} becomes smaller. During winter heating mode part-load operation, if the sensible heat ratio of space conditioning line SHR_{sp} becomes smaller because of the reduced volume flow, the zone relative humidity will also be greater than that in design load.

From Eq. (20.69), the design supply volume flow rate $\dot{V}_s$, in cfm [m^3/(60 s)], can be calculated as

$$\dot{V}_s = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_s)} \tag{21.1}$$

where Q_{rs} = zone sensible cooling load, Btu/h (W), and T_r, T_s = zone and supply air temperature, °F (°C). In Eq. (21.1), [m^3/(60 s)] means that in SI units, the $\dot{V}_s$ unit should be $^1/_{60} \times m^3/s$.

From Eq. (20.48*b*), the design cooling coil load Q_{cc}, in Btu/h (W), can be calculated as

$$Q_{cc} = 60\dot{V}_s\rho_s(h_m - h_{cc}) \tag{21.2}$$

where h_m = enthalpy of the mixture of outdoor and recirculating air, Btu/lb (J/kg), and h_{cc} = enthalpy of conditioned air leaving the cooling coil, Btu/lb (J/kg). From Eq. (20.85), the design heating coil load Q_{ch}, in Btu/h (W), which offsets the space heating load or sometimes maintains a setback temperature during unoccupied periods to reduce warm-up load next morning and prevent freezing, can be calculated as

$$Q_{ch} = 60\dot{V}_s\rho_s c_{pa}(T_{hc} - T_{en}) \tag{21.3}$$

where T_{hc}, T_{en} = air temperature leaving and entering heating coil, °F (°C).

Zone Temperature Control—Sequence of Operations

Consider a single-zone VAV system using an AHU with a water cooling coil, a water heating coil, and a temperature economizer (air economizer), as shown in Fig. 21.1. The sequence of operations of this VAV system to maintain a preset zone temperature T_r is as follows:

1. When the AHU is in the off position, the outdoor and exhaust dampers are closed, the two-way valves of the cooling and heating coils are also closed, and supply and return fan motors are off.

2. When the AHU is in the on position, zone temperature sensor T1 senses the zone temperature T_r, sends a signal to the DDC controller, and compares it with the set point in summer of 75°F (23.9°F) in the controller. If $T_r > 75$°F (23.9°C), the DDC controller calls for cooling—cooling mode operation. At the same time, the outdoor temperature T_o is measured by the outdoor temperature sensor T2. If $T_o > 75$°F (23.9°C) (a temperature economizer set point), the DDC controller opens the outdoor and exhaust dampers to a minimum opening position, and fully opens the recirculating damper. The DDC controller also starts supply and return fan motors with speed control via variable speed drives.

3. The discharge air temperature T_{dis} is sensed by temperature sensor T4. During cooling mode operation, if the off-coil temperature T_{cc} is 55°F (12.8°C), the set point of $T_{dis} = 55 + 2$ (fan power) = 57°F (13.9°C). When $T_{dis} > 57$°F (13.9°C) is sensed by T4, the DDC controller opens and modulates the two-way valve of the water cooling coil so as to maintain $T_{dis} = 57$°F (13.9°C).

4. If $T_o \leq 75$°F (23.9°C), using the free cooling of outdoor air is always beneficial in order to save energy. The DDC controller fully opens the outdoor and exhaust dampers and, at the same time, closes the recirculating damper. After the air economizer is energized, if T_{dis} is still higher than 57°F (13.9°C), the DDC unit controller will open and modulate the two-way valve of the water cooling coil to maintain $T_{dis} = 57$°F (13.9°C).

5. Zone temperature T_r is sensed by a temperature sensor T1. This signal is compared with the zone temperature set point $T_{r,s}$, such as 75°F (23.9°C) in the DDC controller. If T_r drops below or rises above 75°F (23.9°C), an output from the DDC controller actuates and modulates the supply fan speed, and thus the supply volume flow rate through an adjustable-frequency variable-speed drive, to maintain $T_r = 75$°F (23.9°C). For single-zone VAV system serving a large conditioned space, several temperature sensors can be located in different areas. Either the mean value or the highest or lowest temperature can be taken as sensed input to the DDC controller.

6. A pressure sensor P1 is used to sense the zone pressure Δp_r in the conditioned space. If $\Delta p_r >$ 0.03 in. WC (7.5 Pa), the DDC controller starts the return fan and modulates its fan speed to maintain a zone pressure $\Delta p_r \leq 0.03$ in. WC (7.5 Pa).

7. When $T_o \leq 55°F$ (12.8°C) and the system is operated in cooling mode operation, the DDC controller

- Modulates the outdoor and recirculating damper to maintain a mixing temperature of 55°F (12.8°C) and $T_{dis} = 57°F$ (13.9°C) with a volume flow of outdoor air not less than the ventilation requirement
- Modulates the fan speed to maintain a zone temperature of 75°F (23.9°C)
- Closes the two-way valve of the water cooling coil

8. If the zone temperature sensed by sensor T1 is $72°F \leq T_r < 74°F$ ($22.2°C \leq T_r < 23.3°C$), the system is operated in dead-band operation. Dead-band operation is neither a cooling nor heating mode operation. In dead-band mode, the DDC controller

- Keeps both of the two-way valves of the cooling and heating coils closed
- Opens the outdoor damper to its minimum opening position and fully opens the recirculating damper to provide necessary outdoor ventilation air for the occupants

9. If the zone temperature T_r drops below 72°F (22.2°C), the DDC controller calls for heating, (heating mode operation). Also if $T_o < 55°F$ (12.8°C), then

- The outdoor damper is opened to its minimum opening position to extract only the necessary amount of outdoor ventilation air.
- The DDC controller opens and modulates the two-way valve of the heating coil to maintain a discharge air temperature during heating mode not higher than 15°F (8.3°C) above the space temperature, say $72 + 15 = 87°F$ (30.6°C).

For the zone temperature control in heating mode operation, a reverse-acting relay is energized so that the control action is reversed when the cold air supply is changed over to warm air supply. The DDC controller also modulates the supply fan speed to maintain a zone temperature $T_r = 72°F$ (22.2°C).

During the above zone controls for a single-zone VAV system, to determine whether the DDC controller calls for cooling or heating, either a zone temperature set point $T_{r,s}$ or a specific outdoor temperature set point $T_{o,s}$ can be used as the comparing criterion. Zone temperature set point $T_{r,s}$ directly relates to the zone load and the actual zone conditions and is recommended. Because of the high internal load, a zone may still need a cold air supply at a rather low outdoor temperature.

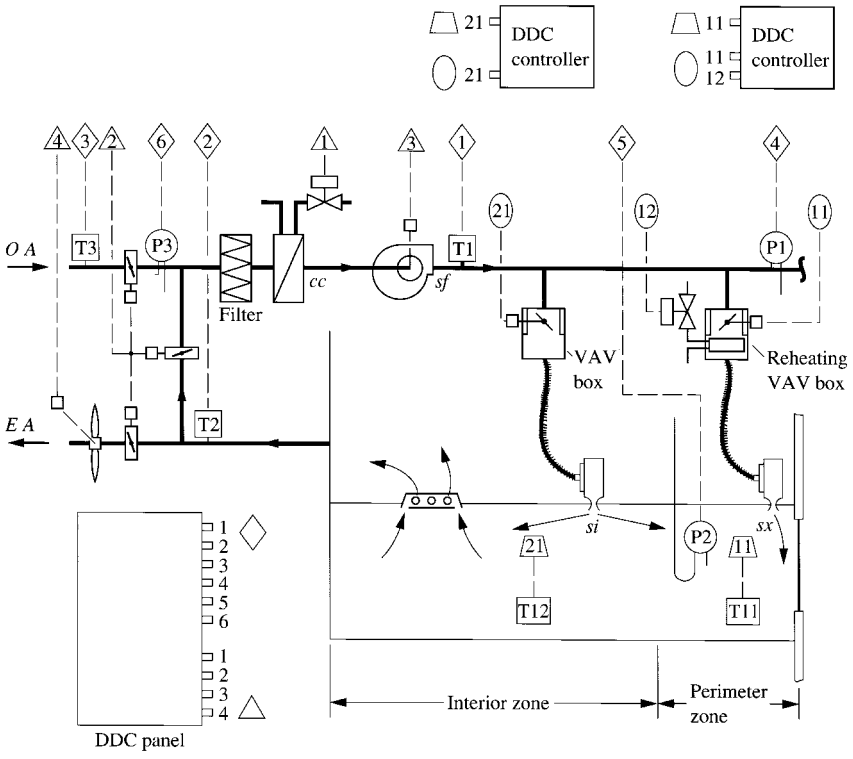
When a cooling mode is automatically changed to heating mode or vice versa, the width of the dead band greater than 3°F (1.7°C) is not appropriate for thermal comfort. When two separate set points are used and heating and cooling mode operations are not automatically changed over, only then can a dead-band width of 5°F (2.8°C) or sometimes greater be used.

To save energy as well as to increase the zone supply volume flow rate, the discharge air temperature is often reset to a higher value during cooling mode part-load operation, such as from 57 to 60°F or to 63°F (13.9 to 15.6°C or to 17.2°C).

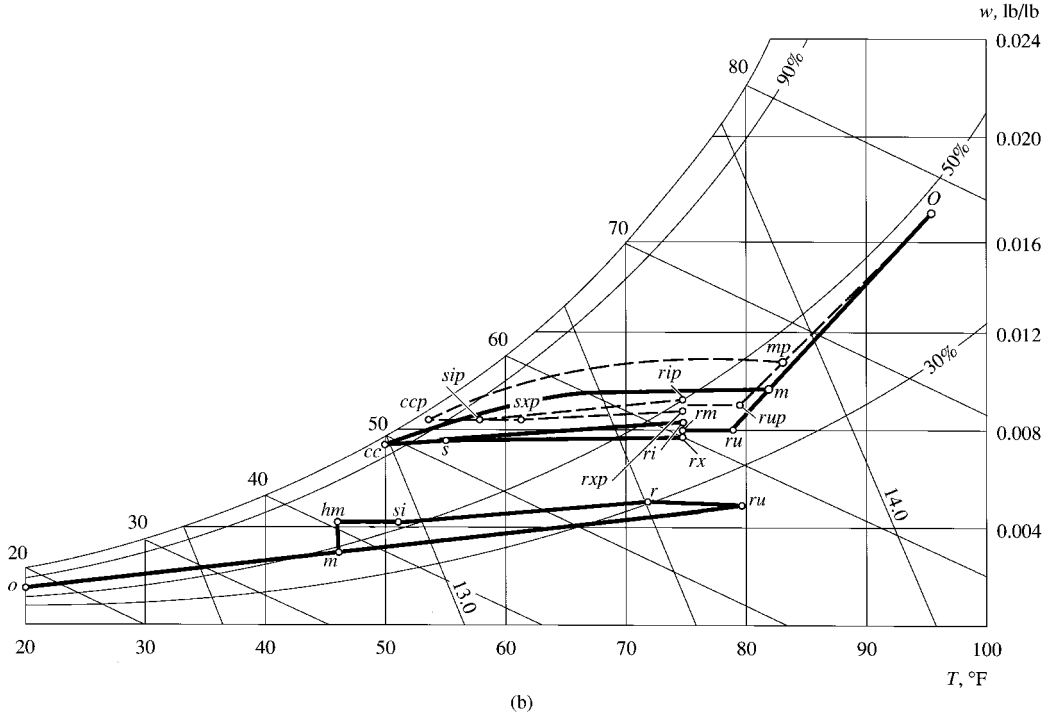
21.3 VAV COOLING, VAV REHEAT, AND PERIMETER HEATING VAV SYSTEMS

VAV Cooling Systems

A VAV cooling system is a multizone air system with VAV boxes to modulate the cold supply air volume flow rate to match the variation of zone load to maintain a preset zone temperature, as shown by the air system that serves the interior zone in Fig. 21.5*a*. VAV cooling systems with VAV boxes only are widely used to serve the interior zone in commercial and industrial buildings.



(a)



(b)

FIGURE 21.5 A VAV reheat system: (*a*) schematic diagram; (*b*) air conditioning cycle.

VAV Reheat Systems

A VAV reheat system is a multizone VAV system with either VAV boxes or reheating VAV boxes to modulate its cold supply volume flow rate for each of the control zones, to match the variation of the zone load to maintain a preset zone temperature. For the interior zone in a building, VAV boxes are used (VAV cooling); and for the perimeter zone in a building, reheating VAV boxes are used (VAV reheat) in which a reheating coil is added to a cooling VAV box, as shown in Fig. 21.5.

To minimize energy consumption, as covered in Sec. 20.20, ASHRAE/IESNA Standard 90.1-1999 specifies the following criteria: VAV reheat systems shall be capable of operating in sequence the supply of heating and cooling energy to the zone to minimize simultaneous cooling and heating except as follows:

1. 300 cfm (142 L/s), or a zone peak flow rate total of no more than 10 percent of the total fan system flow rate

2. The minimum required to meet ventilation requirements

3. 0.4 cfm/ft^2 (2.03 L/s·m^2) of zone conditioned floor area

VAV reheating is a simple and effective means of maintaining preset space temperature with a lower relative humidity at cooling mode part-load operation and winter heating mode. For details refer to Standard 90.1-1999.

System Description

The mixture of outdoor and recirculating air is conditioned either in an AHU or in a PU. Both contain a supply fan, a return or relief fan, cooling coils, and medium-efficiency filters and are distributed to various control zones via supply ducts, runouts, reheating VAV boxes, VAV boxes, and outlets.

During summer cooling mode operation, both the reheating VAV boxes in the perimeter zone and the VAV boxes in the interior zone modulate the zone supply volume flow rate to maintain a predetermined zone temperature T_m. When the space sensible load in the perimeter zone drops during summer cooling mode part-load operation, the VAV box closes its opening until its minimum setting is reached. If the zone temperature T_m drops further and falls below a certain limit, the reheating coil is energized and reheats the supply air to maintain a predetermined zone temperature.

During winter heating mode operation, when one of the zone temperatures in the perimeter zone falls below a predetermined limit, the reheating coil in the reheating VAV box in this control zone is energized to heat the air to offset the space heating load to maintain a preset zone temperature.

For a control zone in the interior zone, the zone sensible cooling load seldom drops below 30 percent of its peak load while the space is occupied, except during morning warm-ups and weekends. Therefore, reheating VAV boxes are not used in the interior zone. In the interior zone, zone temperature T_m is also maintained by modulating the supply volume flow rate of the VAV box in each control zone.

If an AHU or PU serves both the perimeter and the interior zone, space air from the perimeter and interior zone is often returned to the same AHU or PU in the fan room through return slots, ceiling plenum, and connecting return ducts.

Perimeter Heating VAV System

A perimeter heating VAV system consists of a hot water perimeter heating system to offset the winter transmission and infiltration loss in the perimeter zone and a VAV cooling system with VAV

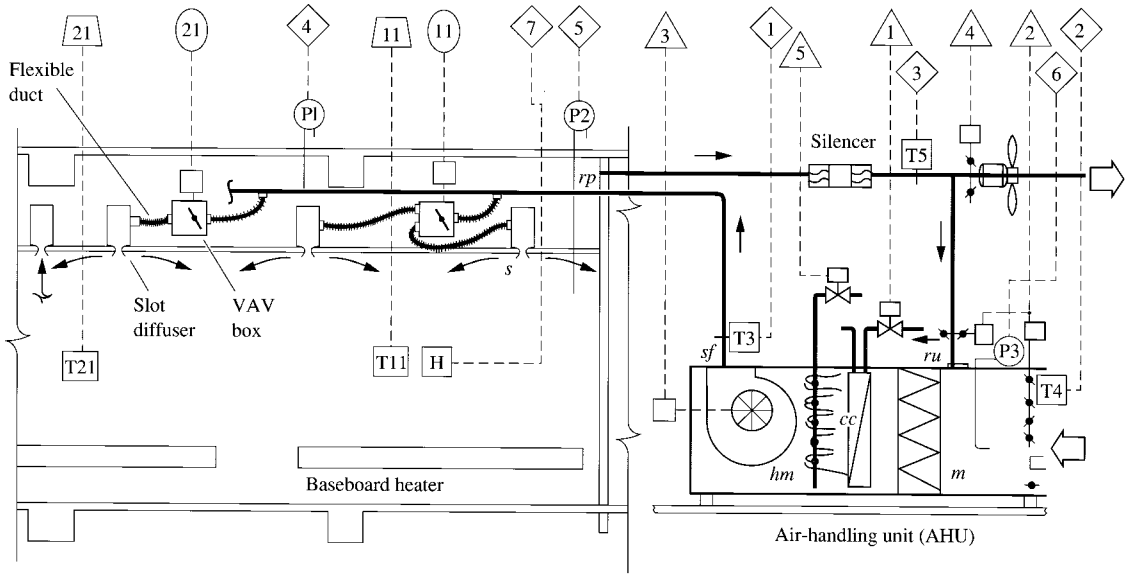


FIGURE 21.6 A perimeter heating VAV system.

boxes to provide cooling and outdoor ventilation air for both perimeter and interior zones, as shown in Fig. 21.6. A perimeter heating system consists of hydronic hot water finned-tube baseboard heaters with a zone control system. The baseboard heaters must be carefully zoned and controlled in coordination with the VAV boxes. Each side of the building should have at least a separate control zone to match the variation of the solar load and the infiltration loss as the wind direction shifts. Perimeter heating systems are suitable for commercial buildings in locations with long, cold winters.

During the early development of VAV systems, air skin VAV systems were used in office buildings. An air skin VAV system uses a constant-volume variable supply temperature air skin system to offset the transmission loss or transmission gain in the perimeter zone; it also has a VAV cooling system with VAV boxes to provide cooling and outdoor air for the perimeter and interior zones. The supply temperature of the air skin system is reset according to the outdoor temperature. If several air skin systems are not used for various zones of different sides of a building, field surveys show that during a cold day in winter, rooms facing south are sufficiently warmed by the incoming solar radiation. The additional warm air supply from the constant-volume air skin system raises the zone temperature to an unacceptably high level. Simultaneous cooling from the VAV boxes must be provided to offset the warm air supply, which results in a waste of energy.

VAV Box

A variable-air-volume box or a cooling VAV box is a terminal device in which the supply volume flow rate is modulated by varying the opening of the air passage by means of a single-blade butterfly damper, a multiblade damper, or an air valve, as shown in Fig. 21.7. A VAV box may have a single outlet or multiple round outlets.

A single-blade damper VAV box, as shown in Fig. 21.7*a* and *b*, has a simple construction and is simple to operate. A typical damper closes at an angle 30° from vertical and rotates in counterclockwise direction to an angle of 60° in the fully open position.

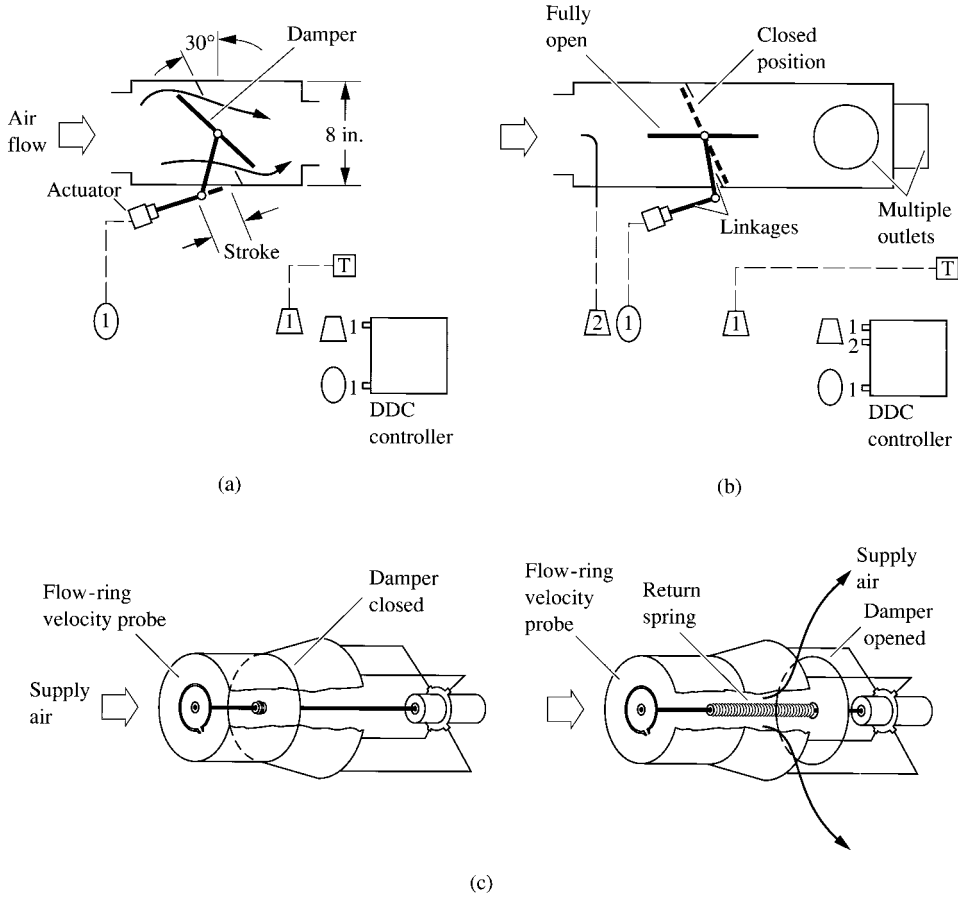


FIGURE 21.7 VAV boxes: (*a*) single-blade, pressure-dependent; (*b*) single-blade, pressure-independent; (*c*) air valve.

An air valve, as shown in Fig. 21.7*c*, is a piston damper moving horizontally inside a hollow cylinder. The opening of the air passage can be adjusted. The main advantage of an air valve is its almost linear relationship between the modulated air volume and the displacement of the piston damper.

Control Mechanisms. The control mechanism of a VAV box can be classified as pneumatic or direct digital control. In a pneumatic control VAV box, the zone temperature is sensed by a pneumatic thermostat, and the rotation of the single-blade damper is actuated by pneumatic power. When the thermostat senses an increase in zone temperature, the increase of compressed air pressure on a diaphragm moves the actuator at a stroke of 1 to 1.5 in. (25 to 38 mm) and rotates the damper through the linkages, as shown in Fig. 21.7*b*. Therefore, the air passage opens wider. The spring pushes the actuator back when the air pressure decreases.

In a DDC VAV box, the temperature sensor sends a signal to a DDC terminal controller. It actuates the motorized operator, moves the actuator to a certain displacement, and opens the single-blade damper wider. The damper is closed by either spring force or the reverse rotation of the motorized operator. Usually, a PI control mode is used by such a DDC terminal controller.

The pneumatic or direct digital control of an air valve is similar to that of a single-blade damper. A pneumatic control VAV box has a faster response actuator and a lower initial cost than those of a DDC VAV box. However, pneumatic systems require more maintenance to provide clean, dry compressed air. More importantly, DDC VAV boxes provide many sophisticated control functions and can communicate with other DDC controllers and the central computer.

There is also a type of self-powered or system-powered VAV box control. It requires a higher supply air pressure to actuate the single-blade damper or air valve according to the input signal from the temperature sensor. A self-powered VAV box also has complicated control mechanisms, so its applications are very limited.

Influence of Duct Static Pressure. VAV boxes can be classified as pressure-dependent or pressure-independent. In a pressure-dependent VAV box, the variation of the duct static pressure at the inlet of the VAV box caused by the opening and closing of the dampers connected to the same main supply duct influences the modulation of its supply volume flow rate. When the pressure at the inlet varies, the airflow often oscillates. Pressure-dependent VAV boxes are least expensive. They are used when the duct static pressure is more stable and in places where there is no need for maximum and minimum limit control.

A pressure-independent VAV box modulates its supply volume flow rate regardless of the variation of duct static pressure at its inlet. A typical pressure-independent VAV box is shown in Fig. 21.7*c*. The temperature sensor resets the set point of the sensed velocity, and the controller actuates the damper to a wider or a narrower position based on the sensed signal from the velocity probe and the reset set point. Even if the static pressure at the VAV box inlet varies from 0.5 to 3 in. WG (125 to 750 Pag), the volume flow is maintained according to the required value called for by the temperature sensor and the controller. Pressure-independent VAV boxes are widely used in VAV systems.

The sizes of the VAV boxes made by one manufacturer range from 04 to 20, with a corresponding volume flow rate from 225 to about 4000 cfm (106 to 1887 L/s). The pressure drop of these VAV boxes at nominal maximum volume flow rate when the damper is fully open usually varies from 0.2 to 0.5 in. WC (50 to 125 Pa).

To provide the required amount of outdoor air to the conditioned space for occupants, a VAV box usually reduces its volume flow to a minimum setting, such as 30 percent of its design flow, while the space is occupied.

Reheating VAV Box

A reheating VAV box is a VAV box with a reheating coil, as shown in Fig. 21.8. The reheating coil is usually a two- to three-row hot water heating coil. If an electric coil is installed, it should be a duct-mounted coil at least 4 ft (1.22 m) downstream from the VAV box. Federal and local safety codes must be followed if an electric coil is installed. To prevent excessive pressure drop at the reheating coil, the face velocity of the reheating coil is usually between 200 and 400 fpm (1 and 2 m/s). For each row of the reheating coil, the additional pressure drop is usually between 0.1 and 0.2 in. WC (25 and 50 Pa).

In the perimeter zone, a two-way slot diffuser should be located about 1 ft (0.3 m) from the window. Air should be discharged downward from one of the slots in a direction toward the window, as described in Sec. 18.6. Air is discharged horizontally away from the window from another slot.

Sound Power Level of a VAV Box

The sound power level of a VAV box depends mainly on the following:

- Volume flow of supply air for a specific size box, cfm (L/s)
- Difference in static pressure Δp_s across the VAV box, in in. WC (Pa), to provide a specific volume flow, in cfm (L/s)
- The configuration of the VAV box, flexible ducts, and diffusers

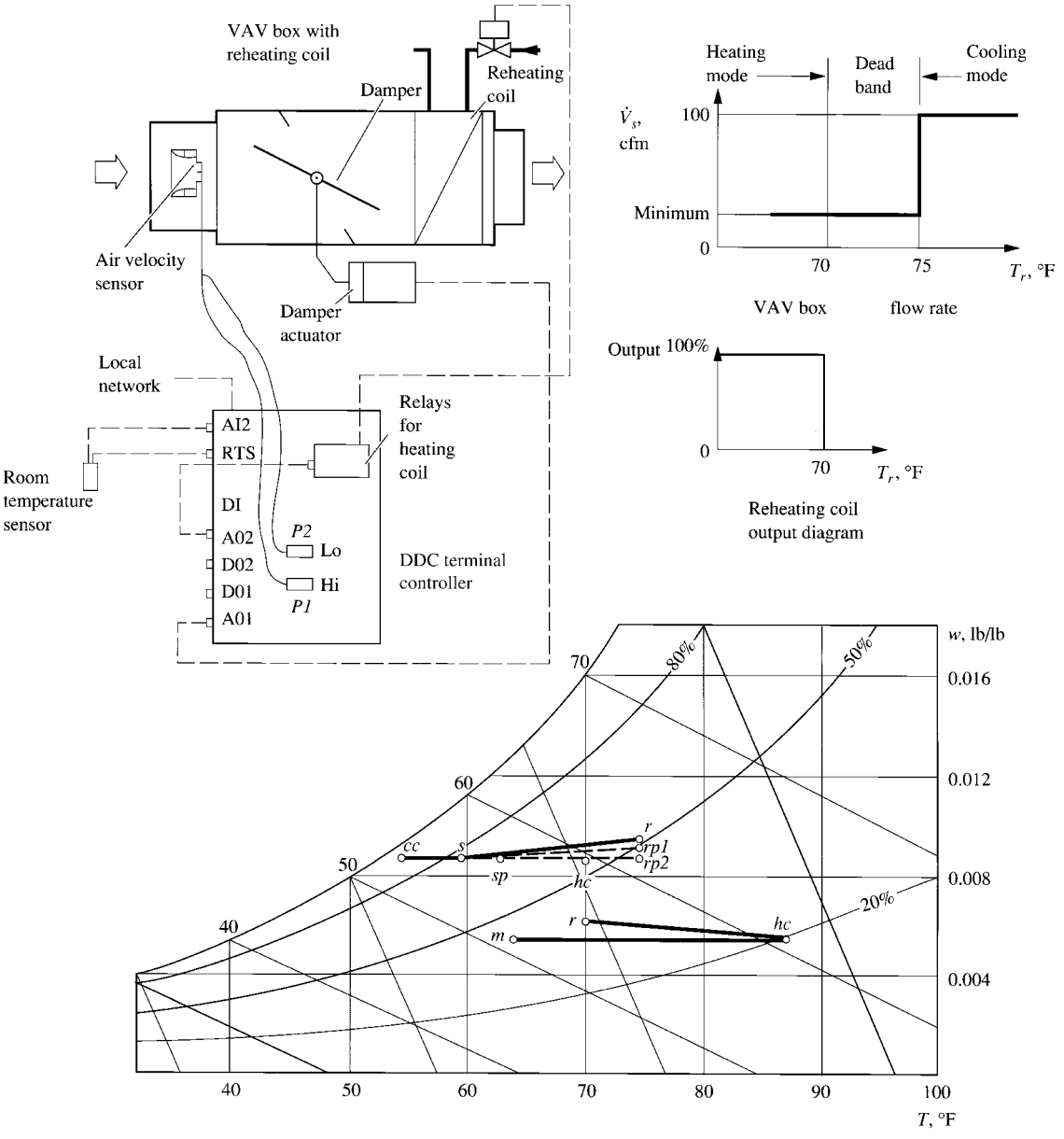


FIGURE 21.8 DDC for a reheating VAV box.

The greater Δp_s, the larger the static pressure at the box inlet. The smaller the damper opening, the higher the sound power level of the VAV box.

Noise generated by the VAV box can be transmitted to occupants in the conditioned space by a duct-borne path through flexible ducts and diffusers, or a radiated path from its casing through the ceiling plenum.

Evaluating noise levels from VAV boxes on an occupied zone has always been difficult because of the many variables that affect the results. The ADC and ARI have jointly developed Industry

Standard 885, *Acoustical Level Estimation Procedure*, as described in Chap. 19, to provide a reliable method to predict NC or RC in an occupied space.

In Smith (1989), the sound pressure levels of two VAV boxes in a 29-ft × 20-ft (8.8-m × 6.1 m) room with an 8-ft 7-in. (2.6-m) ceiling were estimated and measured. One VAV box was operated at 500 cfm (236 L/s) at 4.0 in. WG (1000 Pa) inlet static pressure and supplied 250 cfm (118 L/s) to each of two diffusers through flexible ducts. The actual sound pressure level measured in the occupied space was NC-41. Another VAV box was operated at 600 cfm (283 L/s) with 3.0 in. WG (750 Pa) inlet static pressure and supplied 300 cfm (142 L/s) to each of two diffusers in the same room. The actual sound pressure level measured was about NC-41. The measured sound pressure levels were about 2 to 3 dB higher or lower than the estimated duct-borne and radiated paths according to Industry Standard 885. If the sound power level at mid-frequency 2*k* and 4*k* Hz can be further attenuated by flexible duct and the slot diffuser, the sound pressure level in the occupied space may drop to NC-35. The sound power level of a VAV box listed in a manufacturer's catalog must be tested in a certified laboratory.

VAV Reheat Zone Temperature Control and Sequence of Operations

The year-round control of zone temperature served by a reheating VAV VV1 box in the perimeter zone of a building served by a VAV reheat system can be divided into cooling mode, dead-band mode, and heating mode operations.

1. On-off control is in the off position. When an AHU or a PU that serves this perimeter zone is shut off, the single-blade damper of the VAV box VV1 and the two-way valve V1 of the reheating coil both are closed.

2. On-off control is in the on position. When the zone temperature T_{rx1} sensed by the zone sensor exceeds a set point, such as 75°F (23.9°C), the DDC system controller calls for cooling; the AHU or PU and the reheating VAV box are both in cooling mode operation. The DDC terminal controller opens the single-blade damper VV1 of the reheating VAV box to maximum opening position. The two-way valve of the reheating coil is still closed.

3. If the zone temperature T_{rx1p} at cooling mode part-load operation drops below 75°F (23.9°C) because of a reduction in the zone sensible cooling load Q_{rsx1p}, the DDC terminal controller receives a corresponding analog input from the zone sensor. This signal is compared with the set point 75°F (23.9°C). If $T_{rx1p} < 75$°F, (23.9°C), the DDC terminal controller sends an output to the damper actuator and starts to reduce the opening of the flow passage of the reheating VAV box. The total pressure loss across the VAV box is then increased, and so the volume flow rate of the conditioned air supply to this control zone $\dot{V}_{sx1p}$ is reduced accordingly. If the reduced $\dot{V}_{sx1p}$ is balanced with the zone sensible cooling load, T_{rx1p} will remain at 75°F (23.9°C). The DDC terminal controller then modulates the opening of the single-blade damper VV1 with a PI control mode to maintain a constant 75°F zone set point, as shown in Fig. 21.8.

4. Further drop in Q_{rsx1p} causes damper VV1 to close more and more until it reaches the minimum setting which is necessary to supply the required amount of outdoor ventilation air for this zone, such as 30 percent of the zone peak flow. After the damper reaches the minimum setting, if Q_{rsx1p} further drops and causes T_{rx1p} to drop below 75.0°F (23.3°C), the reheating VAV box will operate in dead-band mode.

5. When the zone temperature drops within the range 72.0°F $< T_{rx1p} \leq$ 75.0°F (22.2°C $< T_{rx1p} \leq$ 23.9°C), the reheating box and zone control operate in dead-band mode. In dead-band mode, the DDC terminal controller actuates as follows:

- The damper in the reheating VAV boxes closes to the minimum setting, and the minimum cold primary air supply $\dot{V}_{sx1,\min}$ is at a value, such as 30 percent of the zone peak flow, to provide outdoor ventilation air for the control zone.
- The reheating coil is deenergized.
- The space relative humidity will be slightly lower than that at the design condition.

In dead-band mode, the reheating box will reenter the cooling mode operation only if $T \geq 75°F$ (23.9°C).

6. When the zone temperature $T_{rx1} \leq 72.0°F$ (22.2°C), the reheating box and the associated zone control are in heating mode operation. During heating mode operation, the DDC terminal controller actuates the reverse action relay so that the direct-acting mode will be changed to reverse-acting mode.

The volume flow rate of the cold primary air supplied from the reheating box in heating mode $\dot{V}_{s,cxn}$ is still maintained in a minimum setting of 30 percent of zone peak volume flow and will remain constant to provide cold primary air for ventilation requirement.

The DDC terminal controller opens and modulates the opening of the two-way valve of the reheating coil as well as the mass flow rate of hot water flowing through the coil to maintain a preset zone temperature $T_{rx1} = 72°F$ (22.2°C). The zone relative humidity may be at a value between 25 and 50 percent depending on the zone internal latent loads, outdoor air humidity ratio, and percentage of outdoor air intake.

At winter design conditions, the warm supply air temperature from the reheating box (after the cold primary air is heated in the reheating coil) should not exceed 87°F (30.6°C), in order to reduce the buoyancy effect of the warm supply air.

The dead band in a year-round zone-temperature control for a perimeter zone served by a VAV reheat system is different from the dead band in a dual-thermostat year-round zone temperature control system served by a constant-volume single-zone system, as discussed in Sec. 20.19. In a VAV reheat system,

- The cooling mode is automatically changed over to heating mode operation and vice versa.
- The occupants may suffer the zone temperature variation from cooling mode to dead-band mode, and then to heating mode and vice versa, continuously within a time period of several hours.

The width of the dead-band mode in such a circumstance is preferably limited to 1 to 3°F (0.6 to 1.7°C).

Stability of Zone Control Using VAV Boxes and Reheating VAV Boxes

VAV boxes including reheating VAV boxes using DDC terminal controllers can provide the following functional controls:

- Zone temperature control by modulating the damper or air valve
- Zone temperature reset based on outdoor temperature or zone temperature
- Maximum and minimum settings of airflow for each zone
- Morning warm-up and cool-down control

A normally closed DDC VAV box can shut off its supply volume flow when the served control zone is not occupied. This is especially useful for partially occupied floors after normal working hours.

Air flowing through VAV boxes is self-balanced along the supply main duct. However, airflow imbalance does affect the modulation range of VAV boxes. A well-designed supply duct system with minimized imbalance is recommended.

One characteristic of VAV boxes that directly affects their performance is control stability. Dean and Ratzenberger (1985) described the primary differences between an efficient and inefficient design of a VAV box and its control. An excellent design minimizes the open-loop gain and improves the overall system stability. Gain is defined as the real part of the logarithm of the transfer function of the control system. The primary factors that affect the open-loop gain and thus the stability of a VAV box are as follows:

- *Actuator size*. In an efficient design, the stroke of the actuator is small [typically 1.0 in. (25 mm)], and in a pneumatic system, the active area of the actuator is typically only 2 in.^{2} (1290 mm^2)

instead of 8 in.^{2} (5160 mm^2). A larger actuator needs more compressed air to drive a full-range stroke, so the gain is greater.

- *Damper size*. Gain is directly proportional to the size of the damper as well as the size of the VAV box.
- *Duct pressure*. A lower excess pressure Δp_{exc}, in in. WC (Pa), during maximum and minimum airflow affects open-loop gain. Excess pressure is defined as the amount of pressure above that required to drive a nominal flow through a fully open damper.

According to Dean and Ratzenberger (1985), the typical measured performance of 26 efficient B-size VAV boxes and 27 inefficient C-size VAV boxes is shown below:

	B size	C size
Maximum flow, cfm	442	448
Corresponding pressure $p_{s,max}$, in. WG.	1.07	1.40
Minimum flow, cfm	64	66
Corresponding pressure, $p_{s,min}$, in. WG	1.24	1.59
Excess pressure, in. WC	0.08	0.62

For a step function response due to a step change in internal loads, such as turning lights on or off, the deviation from set point in a pneumatic control system with a large actuator lasts about 40 min. In a DDC system, it lasts only about 15 min.

Based on their analysis, Dean and Ratzenberger (1985) recommended the following:

- A VAV system using pneumatic control VAV boxes should have relatively low main duct pressures and relatively small VAV boxes and actuators.
- VAV boxes using DDC systems have excellent operating characteristics, although their actuators are a little slower than pneumatic controls.
- Pressure-independent VAV boxes with velocity resetting improve the control stability of poorly designed pneumatic controls, make an unstable system more stable, and improve the performance of a poorly designed supply duct system.

Case Study: A VAV Reheat System

Consider a multizone VAV reheat system using an AHU to serve a typical floor of 20,000 ft^2 (1858 m^2) of conditioned area in a high-rise office building that is divided nearly equally into perimeter and interior zones. Reheating VAV boxes are used for summer cooling and winter heating in perimeter zones. VAV boxes are used for summer and winter cooling in the interior zone when conditioned space is occupied. The perimeter and interior zones may each be divided into several to more than 20 control zones. The summer and winter design conditions are shown below:

	Perimeter	Interior
Summer indoor temperature, °F	75	75
Space relative humidity, percent	43	45
Summer outdoor temperature, °F	95	95
Summer outdoor wet-bulb temperature, °F	78	78
Winter indoor temperature, °F	72	72
Winter outdoor temperature, °F	20	20
Winter indoor relative humidity, percent	30	30

	Perimeter	Interior
Winter outdoor humidity ratio, lb/lb	0.0017	0.0017
Summer block cooling load, Btu/h	300,000	160,700
Summer sensible cooling load, Btu/h	285,500	135,000
Winter heating linear density, Btu/h·ft	150	—
Winter interior zone latent load, Btu/h	13,300	13,300
Outdoor air required, cfm	1,920	1,920
Summer supply system heat gain, °F	5	5
Summer return system heat gain, °F	3	3

For the perimeter zone in a typical floor of this office building, the average space cooling load density at summer design condition is 30 Btu/h·ft^2 (94.6 W/m^2 or 8.8 W/ft^2). The sensible heat ratio of the space conditioning line SHR_s for the perimeter zone at design load is usually about 0.95.

For the interior zone of a typical floor of this office building, the summer design load density is 15.5 Btu/h·ft^2 (4.5 W/ft^2 or 48.4 W/m^2) depending on the equipment load density. Occupant density is assumed to be 150 ft^2/person (14 m^2/person). The sensible heat ratio of the space conditioning line SHR_s for the interior zone may vary from 0.80 to 0.90.

Conditioned Air Off-Coil and Supply Temperature Differential

For a supply temperature differential of 20°F (11.1°C), if the zone supply temperature T_s = 55°F (12.8°C), for a typical AHU with a supply fan power temperature rise of 2°F (1.1°C) and a supply duct temperature rise of 3°F (1.7°C) at summer design conditions, the temperature of conditioned air leaving the cooling coil T_{cc} should be 55 − (2 + 3) = 50°F (10°C). If the relative humidity of conditioned air off the coil φ_{cc} is 95 percent, and h_{cc} = 20 Btu/lb (46.5 kJ/kg), then, from the psychrometric chart shown in Fig. 21.5*b*, the relative humidity of the supply air φ_s is 81 percent.

Psychrometric analysis shows that to maintain a zone temperature of 75°F (23.9°C) with an off-coil temperature of T_{cc} = 50°F (10°C) and φ_{cc} = 95 percent, a supply temperature T_s = 55°F (12.8°C), and a sensible heat ratio of space conditioning line in the perimeter zone SHR_{sx} of 0.95, or of interior zone SHR_{si} of 0.84, and a supply temperature differential ΔT_s = 20°F (11.1°C), the relative humidity of the perimeter zone must be φ_{rx} = 43 percent and the relative humidity of the interior zone φ_{ri} = 45 percent, as shown in Fig. 21.5*b*.

System Volume Flow Rate and Coil Load

For any control zone in the perimeter and interior zones, the VAV box, reheating VAV box, flexible ducts, and slot diffusers should be sized according to the zone peak supply volume flow rate. The zone peak supply volume flow rate of any control zone served by a VAV box or a reheating VAV box $\dot{V}_{sn}$, in cfm [m^3/(60 s)], at summer design conditions can be calculated as

$$\dot{V}_{sn} = \frac{Q_{rsn}}{60\rho_s c_{pa}(T_r - T_s)} \tag{21.4}$$

where Q_{rsn} = zone sensible cooling load at summer design conditions, Btu/h (W). In a multizone VAV system, the total supply volume flow rate for the perimeter zone $\dot{V}_{sx}$, in cfm [m^3/(60 s)], at summer design load is the maximum possible coincident total supply volume flow rate of various control zones at the same instant, or the block supply volume flow rate, which corresponds to the block load of the perimeter zone Q_{rsx}, in Btu/h (W). The block supply volume flow rate in the

perimeter zone at summer design conditions can be calculated as

$$\begin{aligned}\dot{V}_{sx} &= \frac{Q_{rsx}}{60\rho_s c_{pa}(T_{rx} - T_s)} \\ &= \frac{285{,}500}{60 \times 0.075 \times 0.243(75 - 55)} = 13{,}054 \text{ cfm } (6160 \text{ L/s})\end{aligned} \tag{21.5}$$

Similarly, the total block supply volume flow rate of the interior zone at summer design condition $\dot{V}_{si}$, in cfm [m/(60 s)], is

$$\begin{aligned}\dot{V}_{si} &= \frac{Q_{rsi}}{60\rho_s c_{pa}(T_{ri} - T_s)} \\ &= \frac{135{,}000}{60 \times 0.075 \times 0.243\ (75 - 55)} = 6173 \text{ cfm } (2913 \text{ L/s})\end{aligned} \tag{21.6}$$

Assume that Q_{rsi} and $\dot{V}_{si}$ remain constant during occupied hours. The total supply volume flow rate for the AHU at summer design conditions is calculated as

$$\dot{V}_s = \dot{V}_{sx} + \dot{V}_{si} = 13{,}054 + 6173 = 19{,}227 \text{ cfm } (9073 \text{ L/s}) \tag{21.7}$$

Trunks or main ducts that serve multizones should accommodate the maximum possible airflow through these ducts at a given time. A diversity factor can be multiplied by the sum of individual peak volume flows to estimate the possible maximum flow.

From the psychrometric chart, the weighted mean space relative humidity for perimeter and interior zones $\varphi_{rm} = 43.5$ percent, and the weighted mean humidity ratio $w_{rm} = 0.0079$ lb/lb (kg/kg). Because the temperature of recirculating air entering the AHU $T_{ru} = 75 + 3 = 78°F$, and $w_{ru} = w_{rm} = 0.0079$ lb/lb, point *ru* can be plotted on the psychrometric chart. Draw line *ru-o*.

$$\frac{ru\text{-}m}{ru\text{-}o} = \frac{3840}{19{,}227} = 0.20$$

so point *m* can then be determined: $T_m = 80.5°F$ (26.9°C), and $h_m = 30.2$ Btu/lb (70.2 kJ/kg). The cooling coil load Q_{cc} in the AHU which is also a block load, in Btu/h (W), is

$$\begin{aligned}Q_{cc} &= 60\dot{V}_s\rho_s(h_m - h_{cc}) \\ &= 60 \times 19{,}227 \times 0.075(30.2 - 20) = 882{,}519 \text{ Btu/h } (258{,}578 \text{ W})\end{aligned} \tag{21.8}$$

Summer Cooling Mode Part-Load Operation

During summer cooling mode part-load operation, if the sensible cooling load in the perimeter zone is reduced to about 50 percent of its design load and the sensible cooling load of the interior zone is reduced to 85 percent of its design load, the total volume flow rate of the supply fan may decrease to about two-thirds of its design value. It is assumed that the supply fan power heat gain and the volume flow are both reduced to 0.65 of the design values. The temperature rise at the supply fan at part load is still about 2°F (1.1°C). If the chilled water temperature entering the coil T_{we} is reset from 45°F (7.2°C) at design load to 48°F (8.9°C) at about two-thirds part-load operation, the discharge air temperature increases to $T_{dis} = 50 + 2 + 48 - 45 = 55°F$ (12.8°C). The temperature rise from duct heat gain increases because of the reduction of the supply volume flow rate. Because the increase of this temperature rise is proportional to the reduction of the supply volume flow rate, the supply air temperature for the perimeter zone at part load can be

calculated as

$$T_{sxp} = 50 + 2 + 3 + \frac{3}{0.5} = 61°\text{F (16.1°C)}$$

And the supply temperature for the interior zone at part load is

$$T_{sip} = 50 + 2 + 3 = \frac{3}{0.85} = 58.5°\text{F (14.7°C)}$$

The supply volume flow rate for the perimeter zone at part load is

$$\dot{V}_{sxp} = \frac{0.5 \times 285{,}500}{60 \times 0.075 \times 0.243(75 - 61)} = 9325 \text{ cfm (4400 L/s)}$$

The supply volume flow rate for the interior zone at part load is

$$\dot{V}_{sip} = \frac{0.85 \times 35{,}000}{60 \times 0.075 \times 0.243(75 - 58.5)} = 6359 \text{ cfm (3001 L/s)}$$

At summer cooling mode part-load operation, the total supply volume flow rate is

$$\dot{V}_{sp} = \dot{V}_{sxp} + \dot{V}_{sip} = 9325 + 6359 = 15{,}684 \text{ cfm (7401 L/s)}$$

Assume the following conditions:

- The sensible ratio of the perimeter zone at part-load SHR_{sxp} drops to 0.91.
- The SHR_s for the interior zone at part-load SHR_{sip} remains the same.
- The temperature rise of the return system heat gain from the heat released by the light troffer in the ceiling plenum $\Delta T_{\text{ret}p}$, in °F (°C), is proportional to the air volume flow; that is, $\Delta T_{\text{ret}p} = 3/(15{,}684/19{,}227) = 3.7°\text{F (2.1°C)}$.

From the psychrometric chart, $T_{rup} = 75 + 3.7 = 78.7°\text{F (25.9°C)}$.

If, at summer cooling mode part-load operation, the required amount of outdoor air remains the same as at design load, the ratio of outdoor air to supply air at summer cooling mode part-load operation is therefore

$$\frac{rup\text{-}mp}{rup\text{-}o} = \frac{3840}{15{,}684} = 0.25$$

If the dry- and wet-bulb temperatures of outdoor air are the same as those at design conditions, then from the psychrometric chart $T_{mp} = 83°\text{F (28.3°C)}$, and $h_{mp} = 31.9$ Btu/lb (74.2 kJ/kg). As T_{cc} is raised to 53°F (11.7°C) because of the 3°F (1.7°C) reset in T_{we}, the enthalpy of conditioned air off the coil at 53°F, 95 percent relative humidity is 21.6 Btu/lb (50.2 kJ/kg), and the coil load at summer cooling mode part-load operation is

$$\begin{aligned} Q_{ccp} &= 60\dot{V}_{sp}\rho_s(h_{mp} - h_{cc}) \\ &= 60 \times 15{,}684 \times 0.075(31.9 - 21.6) = 726{,}953 \text{ Btu/h (212,997 W)} \end{aligned} \qquad (21.9)$$

Winter Reheating in Perimeter Zone

If there is only one AHU or PU to serve both the perimeter and interior zones of a typical floor in a VAV reheat system, the air discharged from the AHU or PU at winter design conditions using an air economizer cycle must fulfill the cold air supply requirement of the interior zone, typically at 52°F

(11.1°C), as shown in Fig. 21.5*b*. If the temperature rise from supply duct heat gain is 2°F (1.1°C), the temperature of supply air entering the reheating coil in a reheating VAV box T_{en} is 54°F (12.2°C).

One of the primary considerations in the design of warm air supply VAV systems (including VAV reheat system) at winter design conditions is the control of stratification when warm air is supplied from overhead slot and ceiling diffusers in the perimeter zone. As discussed in Sec. 18.6, it is recommended that the supply air temperature differential $T_r - T_s$ not exceed 15°F (8.3°C), to prevent excessive buoyancy effects. A terminal velocity of 150 fpm (0.75 m/s) is required at the 5-ft (1.5-m) level to offset the cold draft from the external window. An even distribution of the volume flow rate of supply air between the warm airstream blowing downward toward the window glass and the warm airstream horizontally discharged toward the partition wall across from the window is preferable.

At a warm air supply volume flow rate equal to 30 percent of the summer peak volume flow rate of 1.5 cfm/ft^2 (9.14 L/s·m^2), the space heating load per ft^2 of floor area that can be offset by warm air supply in the perimeter zone during winter design conditions is

$$\frac{q_{rh}}{A} = 0.3 \times 1.5 \times 60 \times 0.075 \times 0.243 \times 15 = 7.4 \text{ Btu/h·ft}^2 \text{ (23.3 W/m}^2\text{)}$$

If the depth of the perimeter zone is 15 ft (4.6 m), the heating load linear density $q_{h,\text{ft}}$ is

$$q_{h,\text{ft}} = 7.4 \times 15 = 111 \text{ Btu/h·ft (107 W/m)}$$

The maximum reheating coil load for any control zone in the perimeter zone to offset the zone heating load at winter design conditions Q_{chxn}, in Btu/h (W), can be calculated as

$$\begin{aligned} Q_{chxn} &= Q_{rhxn} + Q_{venxn} \\ &= Q_{rhxn} + 60\dot{V}_{s,\,cxn}\rho_s c_{pa}(T_{rxn} - T_{en,\,n}) \end{aligned} \tag{21.10}$$

where Q_{rhxn} = space heating load for control zone n in perimeter zone, Btu/h (W)
Q_{venxn} = ventilation load due to a 30 percent of peak volume flow of cold supply air, Btu/h (W)
$\dot{V}_{s,cxn}$ = volume flow rate of cold primary air discharged to control zone from AHU of VAV reheating system, cfm [m3/(60 s)]
$T_{en,n}$, T_{rxn} = cold primary air temperature and zone temperature of control zone n, °F (°C)

For a summer peak volume flow rate of 1.5 cfm/ft^2 (9.14 L/s·m^2) in the perimeter zone, at 30 percent of peak volume flow that means $0.3 \times 1.5 = 0.45$ cfm/ft^2 (2.3 L/s·m^2) of cold supply air.

If the zone heating load linear density $q_{h,\text{ft}}$ is 111 Btu/h·ft (107 W/m) and the zone heating load density is 7.4 Btu/h·ft^2, the reheating coil load for 1 ft^2 (0.0929 m^2) of floor area at winter design conditions can be calculated as

$$\begin{aligned} \frac{Q_{chn}}{A} &= 7.4 + 60 \times 0.045 \times 0.075 \times 0.243(72 - 54) = 7.4 + 8.9 \\ &= 16.3 \text{ Btu/h·ft}^2 \text{ (51.4 W/m}^2\text{)} \end{aligned}$$

If the AHU that serves the perimeter zone in a VAV reheat system is separated from the interior zone, and a minimum outdoor air intake of 15 percent of the peak supply volume flow rate can be set up for the perimeter zone during the winter heating mode operation, as shown in the air conditioning cycle in Fig. 21.1*b*, the reheating coil load is then

$$\begin{aligned} Q_{chn} &= 7.4 + 60 \times 0.45 \times 0.075 \times 0.243(72 - 64) = 7.4 + 3.9 \\ &= 11.3 \text{ Btu/h·ft}^2 \text{ (35.6 W/m}^2\text{)} \end{aligned}$$

A saving of about $8.9 - 3.9 = 5.0$ Btu/h·ft^2 (15.7 W/m^2) is possible.

For the winter heating mode operation in the perimeter zone of a VAV reheat system, the following is recommended:

- If there is only one AHU or PU to serve both the perimeter and interior zones in a VAV reheat system and if the zone heating load linear density $q_{h,\text{ft}} > 120$ Btu/h·ft (115 W/m), then the warm air supply volume flow rate from the reheating box in heating mode operation (at cold primary air minimum setting) should be proportionally increased to prevent excessive buoyancy effects.
- It is more energy-efficient to use a perimeter heating VAV system; a dual-fan, dual-duct VAV system; or a fan-powered VAV system; or else the AHU or PU that serves the perimeter zone is separated from another AHU or PU serving the interior zone.

Winter Cooling Mode Operation in Interior Zone

At winter design conditions, the outdoor humidity ratio $w_o = 0.0017$ lb/lb (kg/kg), and the space sensible cooling load in the interior zone may still equal that of summer design conditions. Therefore, the supply temperature differential for the interior zone is $T_r - T_s = 20$°F, or $T_s = 72 - 20 = 52$°F (11.1°C). The supply volume flow rate for the interior zone is still $\dot{V}_{si} = 6173$ cfm (2913 L/s).

If the same AHU or PU is used for both perimeter and interior zones, a minimum supply volume flow rate of cold primary air $\dot{V}_{sx,c}$ that equals 0.3 of the peak supply volume flow rate (including the required outdoor ventilation) is supplied to the perimeter zone during winter heating mode and can be calculated as

$$\dot{V}_{sx,c} = 0.3 \times 13{,}054 = 3916 \text{ cfm } (1848 \text{ L/s})$$

The difference in humidity ratio between supply and zone air is

$$w_r - w_s = \frac{Q_{rl}}{60\dot{V}_s\rho_s h_{fg,\,o}}$$

$$= \frac{13{,}300 + 13{,}300}{60 \times (6173 + 3916) \times 0.075 \times 1061} = 0.00055 \text{ lb/lb (kg/kg)}$$

where w_r, w_s = humidity ratio of zone and supply air, lb/lb (kg/kg). The temperature rise from the return system heat gain at winter design conditions is then

$$\Delta T_{\text{ret},w} = \frac{3}{(6173 + 3916)/19{,}227} = 5.7\text{°F } (3.2\text{°C})$$

and

$$T_{ru} = T_r + \Delta T_{\text{ret},w} = 72 + 5.7 + 2 = 79.7\text{°F } (26.5\text{°C})$$

If the temperature rise from the supply fan power is still about 2°F (1.1°C) and the temperature rise from supply duct heat gain is 3°F (1.7°C) for the interior zone, then

$$T_m = T_s - (2 + 3) = 52 - (2 + 3) = 47\text{°F } (7.2\text{°C})$$

and

$$\frac{T_{ru} - T_m}{T_{ru} - T_o} = \frac{79.7 - 47}{79.7 - 20}$$

$$= \frac{\dot{V}_o}{\dot{V}_s} = 0.55$$

From the psychrometric chart, at a temperature $T_r = 72$°F (22.2°C) and $\varphi_r = 30$ percent, $w_r = 0.005$ lb/lb (kg/kg). Draw line *r-s* from point *r* with $\text{SHR}_s = 0.84$, which intersects the

72 − 20 = 52°F (10.0°C) constant-temperature line at point s, and $w_s = 0.005 - 0.00055 = 0.00445$ lb/lb (kg/kg), as shown in Fig. 21.5*b*. Because $T_{ru} = 79.7$°F (26.5°C) and $w_{ru} = w_r = 0.005$ lb/lb (0.005 kg/kg), draw a line *o-ru*. Because *om/o-ru* = 0.55, $w_m = 0.0032$ lb/lb (kg/kg).

To maintain a space relative humidity of 30 percent at winter design conditions, a steam humidifier is required with a humidifying capacity $\dot{m}_s$, in lb/h (g/s), calculated as follows:

$$\dot{m}_s = 60\dot{V}_s\rho_s(w_s - w_m)$$
$$= 60(6713 + 3916)(0.075)(0.00445 - 0.0032) = 56.8 \text{ lb/h } (0.0072 \text{ kg/s})$$

From the above analysis, when a VAV reheat system is used in an office building, these primary problems may arise:

- Outdoor ventilation air supply becomes insufficient during part-load operation. This is discussed in Chaps. 22 and 23.
- If an air economizer cycle is used for free cooling in the interior zone during winter cooling mode operation, for locations where the outdoor humidity ratio w_o drops below 0.003 lb/lb (kg/kg), the zone relative humidity may fall below 25 percent, or even 20 percent. A steam humidifier may be used, as shown in Fig. 21.5.
- If the same AHU or PU serves both perimeter and interior zones, free cooling is used for the interior zone, and the mixed air in the perimeter zone contains the same percentage of outdoor air as in the interior zone, then a humidifier may be required for the perimeter zone otherwise the space relative humidity may even drop to 20 percent.

When a steam humidifier is installed in an AHU or PU, it is important that its humidifier capacity be strictly controlled so that wet surfaces will not occur inside the AHU, PU, and supply ducts, which may cause indoor air quality problems.

21.4 DUAL-DUCT VAV SYSTEMS

System Description

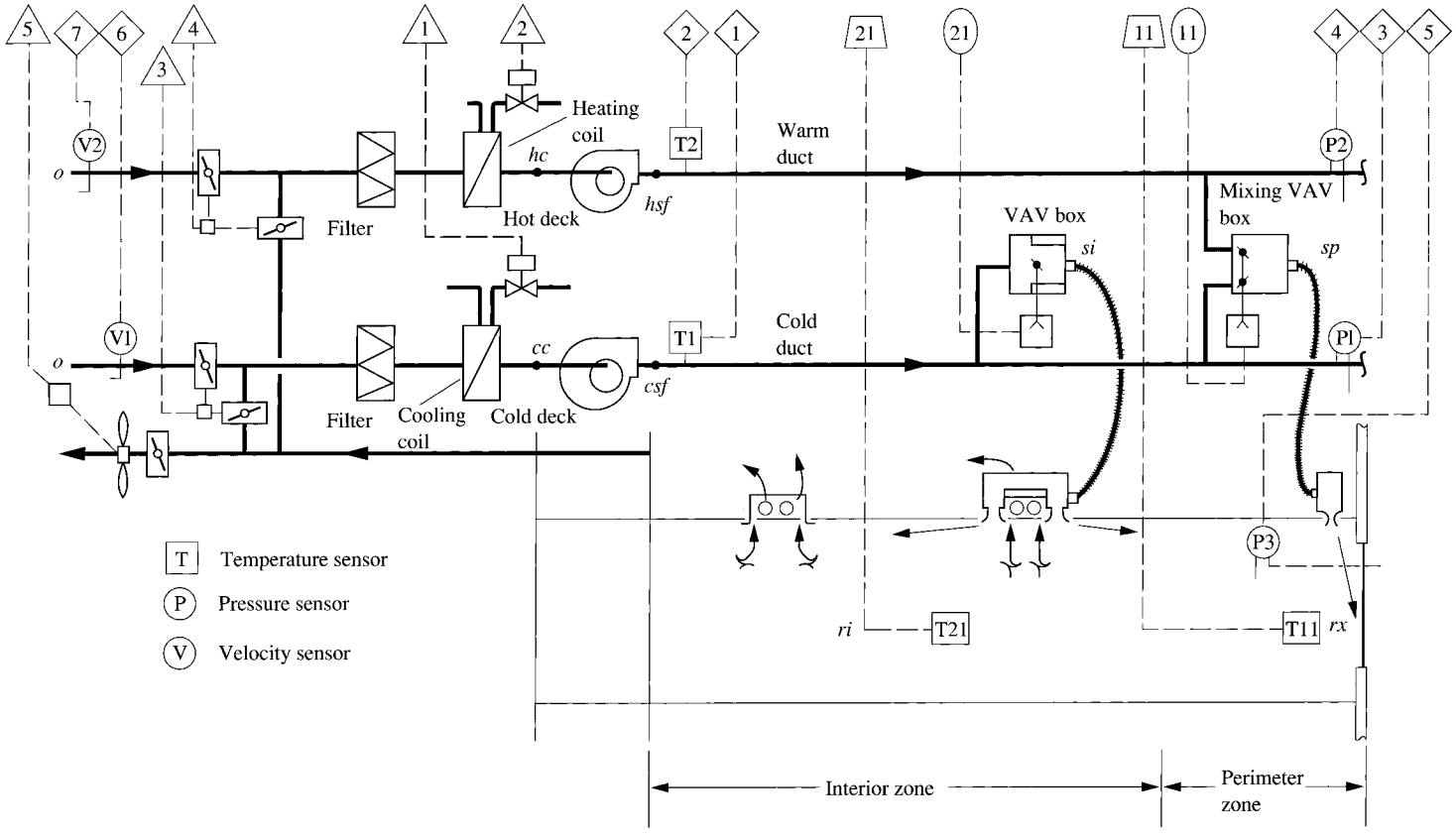
A dual-duct VAV system uses two supply air ducts—a warm air duct and a cold air duct—to supply both warm air and cold air to various control zones in the perimeter zone. Warm air and cold air are mixed in the mixing VAV box in order to maintain a predetermined zone temperature, as shown in Fig. 21.9*a*. For control zones in the interior zone, only a cold air duct is used to supply cold air to offset the year-round zone cooling load through VAV boxes.

A dual-duct VAV system can be a built-up system with a cold deck and a hot deck, or it can be a combination of two air-handling units. Packaged units are seldom used for dual-duct VAV systems because of their more complicated configuration and controls.

A dual-duct VAV system may condition and supply air separately to perimeter zones and interior zones. It is more flexible because the supply air temperature and the variation of supply volume flow rate can be matched better according to the zone load changes and outdoor ventilation air requirements. It is also more energy-efficient than other VAV systems because of

- Transfer of internal heat gains from the interior zone to the perimeter zone through the recirculating air
- Different outdoor air mixing ratio that can be used for interior and perimeter zones for free cooling

According to Warden (1996), a dual-duct VAV system is less expensive than a perimeter heating VAV system and is also less expensive than a VAV hot-water reheat system depending upon the design and configurations.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Warm duct
P2
Filter
Hot deck
hsf
VAV box
si
Mixing VAV box
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o
T1
Cold duct
Pl
cc
csf
Filter
Cooling coil
Cold deck
P3
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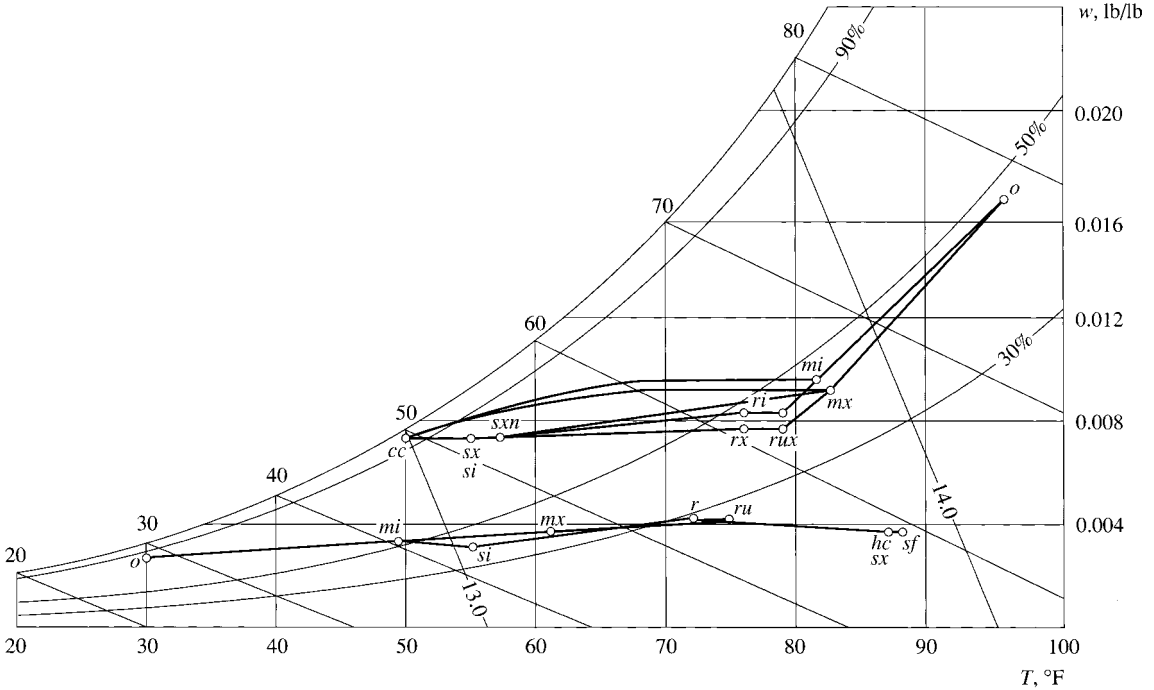
T21
T11
rx
T Temperature sensor
P Pressure sensor
V Velocity sensor
Interior zone
Perimeter zone

(a)



(b)

FIGURE 21.9 A dual-fan dual-duct VAV system: (*a*) schematic diagram; (*b*) air conditioning cycle.

It is possible to use the warm air duct as a cold air duct in summer cooling mode operation so that there is a warm/cold duct and a cold duct. Such an approach may reduce the size of the cold duct and the warm air leakages. However, changeover from cold air supply to warm air supply often produces many problems if the warm/cold duct serves control zones with different orientations in the perimeter zone. This approach is not advisable.

Number of Supply Fans

If an air economizer cycle is used, a relief fan or a return fan must be installed to relieve the space air when 100 percent outdoor air is used for free cooling. If the ceiling plenum is used as the return plenum, and if each typical floor in a multistory building is served by one or two air-handling units, then the pressure drop of the return system is often less than 0.6 in. WC (150 Pa) and a relief fan is a better choice than a return fan. The comparison between a return fan and a relief fan is made in Chap. 22.

A dual-duct VAV system may be installed with a single supply fan that delivers to both warm and cold ducts, or two supply fans, one for the warm duct and one for the cold duct. Although a single supply fan is simpler and lower in initial cost, and the air-handling unit can be a factory-made multizone unit rather than a field-assembled built-up AHU, a dual-fan design has the following advantages:

- It allows the use of different ratios of outdoor air to supply air for the cold duct and the warm duct.
- The warm air supply fan is operated only when it is required, such as during the warm-up period, or in spring, fall, and winter when the transmission and infiltration loss is greater than the internal and solar loads in any control zone in the perimeter zone.
- A dual-fan design simplifies control and saves energy. Different control schemes are possible for cold and warm air temperature resets, as well as for cold and warm air duct static pressure controls.

Dual-fan dual-duct VAV systems are widely used in current dual-duct VAV systems design.

Mixing VAV Box

Figure 21.10 shows a pressure-independent mixing VAV box using a terminal DDC controller. This box consists of two separate, equal-size air passages arranged in parallel — one for warm air and the other for cold air. Each has a single-blade inlet mixing damper. These two air passages are then combined together, and the mixture of warm and cold air is discharged through a discharge volume damper, and then to the diffusers through a flexible duct, single and multiple outlets, as shown in Fig. 21.10*a*. The DDC terminal controller controls the inlet mixing dampers directly according to the signal from the temperature sensor and controls the discharge volume damper from the signal sent by the airflow sensor. The temperature sensor also resets the airflow set point in the DDC controller. As in the VAV box, a pressure-independent mixing VAV box improves system stability and performance and is widely used.

A currently available DDC mixing VAV box has the following controls: zone temperature and reset, minimum and maximum limit, minimum outdoor air, and warm-up and cool-down controls. Most DDC mixing VAV boxes use proportional plus integral (PI) control mode.

The year-round operation of a mixing VAV box can be divided into cooling mode, mixing mode, and heating mode operations, as shown in Fig. 21.10*b*. The cooling mode has the greatest supply volume flow rate, and the cold air supply volume flow rate $\dot{V}_{sc}$, in cfm (L/s), reduces from 100 percent peak supply volume flow rate at summer design conditions down to the minimum setting. In the mixing mode operation, the reduction of the cold air supply volume flow rate exactly equals the increase of the warm air supply volume flow rate; i.e., the total supply volume flow rate remains constant and is equal to the minimum setting. In the heating mode operation, the maximum warm

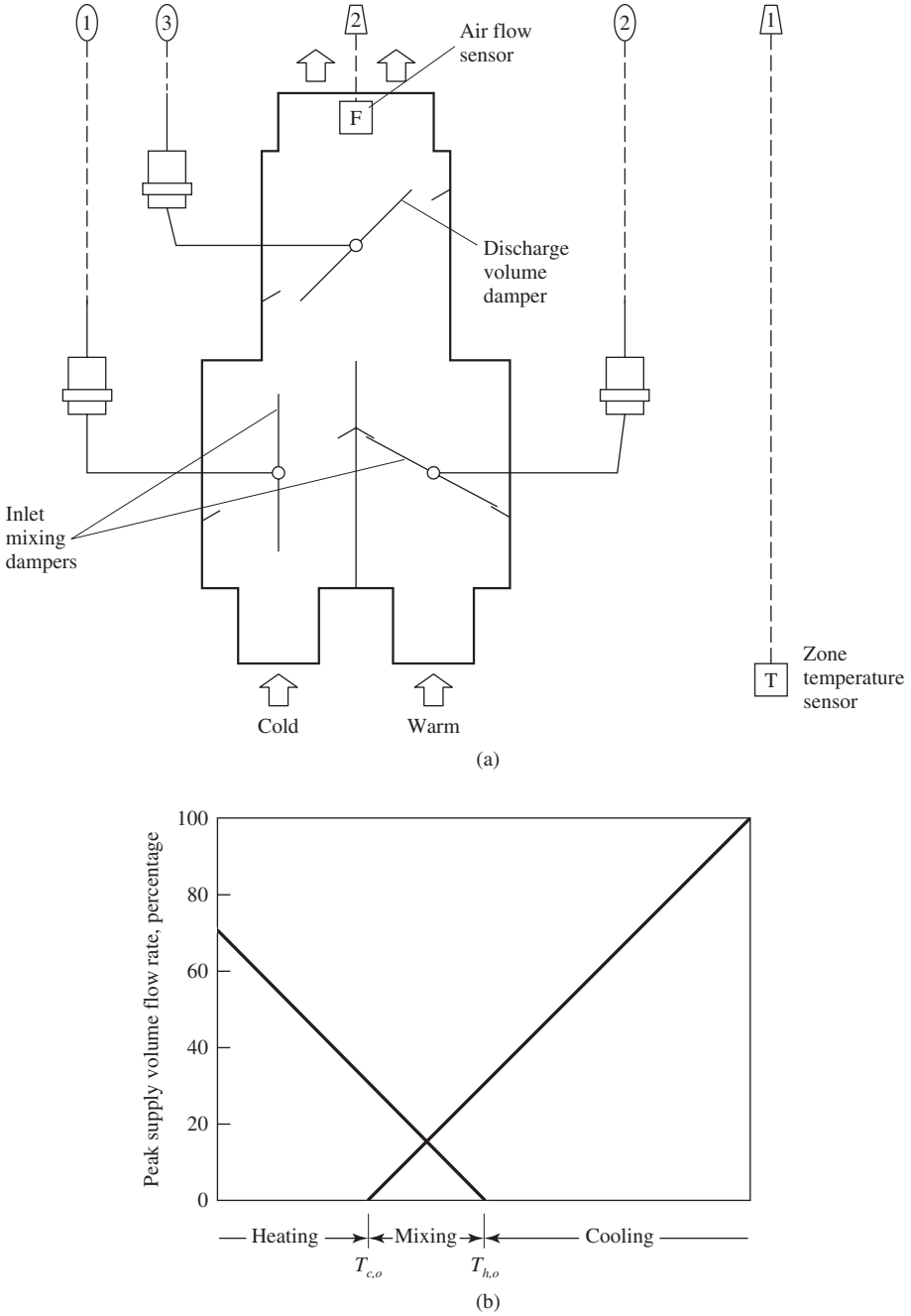


FIGURE 21.10 A pressure-independent mixing VAV box: (*a*) schematic diagram; (*b*) cooling, mixing, and heating mode.

air supply volume flow rate $\dot{V}_{sw}$, in cfm (L/s), may be varied between 50 and 100 percent of the peak cold air supply volume flow rate, depending on the heating load linear density $q_{\text{h,ft}}$, in Btu/h·ft (W/m), in the perimeter zone, so that the maximum zone supply air temperature differential during heating mode operation does not exceed 15°F (8.3°C). The warm air supply volume flow rate reduces as the zone heating load decreases until it is equal to zero in the mixing mode operation.

As in a VAV reheat system, the minimum setting of zone supply volume flow from the mixing VAV box in a dual-duct VAV system should follow the guidelines for VAV systems specified in ASHRAE/IESNA Standard 90.1-1999.

Mixing Mode Operation

In Fig. 21.10*b*, the mixing mode starts at an outdoor air temperature $T_{h,o}$ when the warm air supply reduces to zero and ends at $T_{c,o}$ when the cold air supply drops to zero, both in °F (°C). When outdoor temperature $T_o = T_{h,o}$, the cold air supply volume flow $\dot{V}_{scx1,p}$, in cfm (L/S) is reduced to a mixing-mode setting, such as, 30 percent of the cold peak volume flow rate. Similarly, when $T_o = T_{c,o}$, the warm air supply is also reduced to mixing-mode setting. Consider a mixing VAV box VXn serving a control zone *rxn* in the perimeter zone; its peak cold air supply volume flow rate is 13,054/10,000 = 1.3 cfm/ft^2 (6.6 L/s·m^2), excluding the influence of warm air leakage. If the maximum warm air supply volume flow rate is equal to 70 percent of the peak cold air supply volume flow rate, also when the outdoor air temperature $T_o = T_{h,o}$, if the cold deck discharge air temperature $T_{c,\text{dis}}$ has been reset to 58°F (14.4°C), for a duct heat gain equal to 1°F (0.6°C), the zone sensible cooling load is then

$$Q_{rsxn} = 0.3 \times 1.3 \times 60 \times 0.0075 \times 0.243(75 - 59) = 6.82 \text{ Btu/h·ft}^2 \text{ (21.5 W/m}^2\text{)}$$

When outdoor air temperature $T_o = T_{c,o}$, the control zone *rxn* that the mixing VAV box serves has the following characteristics:

- There is only warm air supply except cold air leakage.
- Zone temperature should be reset to a value for heating mode operation, such as 72°F (22.2°C).
- Heating from hot water, electric heater or gas heater is usually required.

When $T_o = T_{c,o}$, if the hot deck discharge air temperature $T_{h,\text{dis}} = 80$°F (26.7°C), the warm air supply can offset a zone heating load

$$Q_{rhxn} = 0.3 \times 1.3 \times 60 \times 0.075 \times 0.243(80 - 72) = 3.4 \text{ Btu/h·ft}^2 \text{ (10.7 W/m}^2\text{)}$$

For a dual-duct VAV system with a mixing VAV box, there may or may not be a dead band between the changeover from the cooling mode to heating mode through mixing mode operation, as shown in Fig. 21.10*b*.

Zone Controls and Sequence of Operations of Dual-Fan Dual-Duct VAV System

Consider a mixing VAV box VX1 and a VAV box VI1 in a dual-fan dual-duct VAV system, as shown in Fig. 21.9, that serves the perimeter zone and interior zone of a typical floor in an office building with a summer indoor temperature of 75°F (23.9°C) and a winter indoor temperature of 72°F (22.2°C). The following are the zone controls and the sequence of operations of this dual-fan dual-duct VAV system:

1. When time-of-day clock signals the AHUs of the dual-fan dual-duct VAV system to "off," the mixing VAV box VX1 and the VAV box VI1 are closed.

2. When the clock signals the AHU in the on position, the zone temperature sensor T1 measures the zone temperature T_{rxn} in the perimeter zone. If T_{rxn} exceeds a predetermined limit, say,

77°F (24.4°C), the AHUs and the mixing VAV box VX1 and VAV box VI1 are in cooling mode operation. The DDC terminal controller signals VX1 and VI1 to peak airflow supply to the served control zones, and the DDC system controller signals the AHU to reset discharge air temperature to maximum cooling set point, say, 52°F (11.1°C).

3. If the outdoor air temperature T_o exceeds a temperature economizer set point, that is, $T_o >$ 75°F (23.9°C), the DDC system controller closes the outdoor and exhaust dampers to a minimum opening to extract a minimum amount of outdoor ventilation air, fully opens the recirculating damper in both hot and cold decks, and opens and modulates the two-way valve of the water cooling coil in the cold deck to maintain a required discharge air temperature $T_{\text{dis.}}$

At summer design conditions, the refrigeration capacity needed to produce chilled water is 100 percent full capacity. At the same time, the zone temperature sensors sense the zone temperature T_{rx1} and T_{ri1}, and the airflow sensors measure the airflow at the discharge outlet of the mixing VAV box in the perimeter zone and the VAV box in the interior zone. In the control zone X1 of the perimeter zone, the DDC terminal controller modulates and controls the inlet mixing dampers according to the sensed temperature signal and the reset signal from the airflow sensor to maintain a required zone temperature of 75°F (23.9°C). The cold air supply volume flow rate is gradually reduced as the zone sensible cooling load decreases.

In the control zone I1 of the interior zone, the DDC terminal controller modulates the damper opening of the VAV box VI1 to maintain a zone temperature of 75°F (23.9°C). The cold primary air supplied from the VAV box is gradually reduced as the zone sensible cooling load decreases at cooling mode part-load operation. In the cold deck, the discharge air temperature $T_{c,\text{dis}} = 52°\text{F}$ (11.1°C); and in the hot deck, the discharge air temperature $T_{h,\text{dis}} = 81°\text{F}$, which is the mixing temperature of recirculating air (85 percent of airflow) at 78°F (25.6°C) and an outdoor air temperature (15 percent of airflow) of 95°F (35°C) at summer design conditions. The heating coil in the hot deck is not energized.

4. When $50°\text{F} < T_o \leq 75°\text{F}$ (10.0°C $< \text{T}_o \leq$ 23.9°C), the DDC system controller fully opens the outdoor and exhaust dampers and closes the recirulating damper to extract 100 percent outdoor air for free cooling in the cold deck. Refrigeration is at a capacity significantly less than the full-load design capacity and is still required to maintain a cold deck discharge air temperature. In the hot deck, the outdoor damper remains in the minimum opening position, and the recirculaing damper is fully opened to provide a warm air supply.

In the mixing VAV box, the terminal DDC controller modulates the inlet mixing dampers and the discharge volume damper according to the sensed zone temperature and airflow, to maintain a required zone temperature of 75°F (23.9°C). The volume flow rate of cold air supply further reduces as the space cooling load decreases.

The cold-deck discharge air temperature remains at 52°F (11.1°C) in order to meet the requirement of a 15 to 17°F (8.3 to 9.4°C) supply temperature differential in the interior zone. However, the cold supply air temperature from the mixing box may be 3 to 5°F (1.7 to 2.8°C) higher because of the duct and fan power heat gains at part load. The hot deck discharge air temperature may be varied from 80 to 76°F (26.1 to 24.4°C) because of the variation of the outdoor air temperature. The heating coil in the hot deck is not energized.

5. If $T_o = T_{h,o} \leq 50°\text{F}$ (10.0°C), the DDC system controller positions the outdoor and recirulating dampers in both the cold deck and hot deck, first, to provide an outdoor air intake at a value not less than the minimum ventilation requirment and, second, to maintain a mixing temperature $T_m = 50°\text{F}$ (10°C) if it is possible. The DDC system controller also closes the two-way valve of the water cooling coil in the cold deck of the AHU, so that no refrigeration is required for cooling mode operation.

At $T_o = T_{h,o}$, when the cold air supply in the mixing VAV box $\dot{V}_{s,cx1}$ is reduced to mixing mode setting 30 percent of the peak cold air supply volume flow rate $\dot{V}_{scx1,d}$, the mixing VAV box is in mixing mode operation. The mixing operation ends at the warm air supply $\dot{V}_{shx1} = 0.3\dot{V}_{shx1,d}$. In mixing mode, the DDC terminal controller for control zone X1 in the perimeter zone

- Gradually resets the zone temperature from $T_{rx1} = 75°\text{F}$ (23.9°C) at $\dot{V}_{s,cx1} = 0.3\dot{V}_{scx1,d}$ ($T_o = T_{h,o}$) to $T_{rx1} = 72°\text{F}$ (22.2°C) at $\dot{V}_{shx1} = 0.3\dot{V}_{scx1,d}$ ($T_o = T_{c,o}$)

- Modulates the discharge volume damper of the mixing VAV box so that the total supply volume flow rate is maintained at mixing mode setting 30 percent of the peak cold air supply $\dot{V}_{scx1,d}$ during the mixing mode operation

The cold deck discharge air temperature $T_{c,\text{dis}}$ is still 52°F (11.1°C). If the cold air supply in the perimeter zone drops nearly to its minimum setting (30 percent of the peak supply volume flow rate), the duct and fan power heat gain may have a temperature rise of 10°F (5.6°C) which raises the cold supply air temperature in the perimeter zone to about 60°F (15.6°C).

If the hot-deck discharge air temperature $T_{h,\text{dis}}$ cannot be maintained at a value 5°F (2.8°C) higher than the zone temperature T_{rxn}, the DDC system controller opens the two-way valve of the hot water coil and maintains a $T_{h,\text{dis}} = T_{rxn} + 5$. And $T_{h,\text{dis}}$ increases as T_o falls.

The relationship between the outdoor air temperature T_o and $T_{h,o}$ as well as T_o and $T_{c,o}$ depends on the characteristics of the zone load and the construction of the building shell.

6. When the outdoor air temperature $T_o < T_{c,o}$, the mixing VAV box is in heating mode operation. In heating mode operation, the DDC system controller positions the outdoor and recirculating dampers to provide required minimum ventilation as well as to maintain a mixture temperature of 50°F (10°C) if possible. The cold deck discharge air temperature remains at 52°F (11.1°C). Also the two-way valve of the water cooling coil remains closed.

If the duct heat loss is about 2°F (1.1°C), the hot deck discharge air temperature is equal to $T_{h,\text{dis}} = 72 + 15 + 2 = 89$°F (31.7°C). The DDC system controller modulates the two-way valve of the hot water coil in the AHU to maintain $T_{h,\text{dis}} = 89$°F (31.7°C).

In the mixing box X1 in the perimeter zone, the DDC terminal controller modulates the inlet mixing dampers and the discharge volume damper according to the sensed zone temperature and airflow signal, to maintain a preset zone temperature of 72°F (22.2°C). As the zone heating load increases, the warm air supply volume flow rate increases accordingly to a maximum value at winter design conditions with a warm air supply temperature differential not exceeding 15°F (8.3°C).

Discharge Air Temperature Control

In general, the discharge air temperature from the cold deck must meet the year-round cooling requirements for the interior zone. An air economizer cycle using free cooling should be applied to the cold deck whenever possible, to meet the outdoor air requirements and to minimize refrigeration in order to save energy. In the hot deck, internal heat gains carried by the recirculating air should be fully utilized.

Cold deck discharge air temperature $T_{c,\text{dis}}$ is often reset according to zone demands at part load or outdoor air temperature by means of a DDC unit controller. Hot deck discharge air temperature $T_{h,\text{dis}}$ is often reset by outdoor temperature during winter heating mode.

Figure 21.11 shows the cold and warm deck discharge air temperature from the AHU of a typical dual-fan dual-duct VAV system at various outdoor temperatures. The cold deck discharge air temperature $T_{c,\text{dis}}$ is maintained at 52°F at all outdoor air temperatures, in order to meet high zone cooling loads in the interior zone. However, when the cold air supply volume flow rate in the perimeter zone reduces to about 30 percent of its peak volume flow during part-load operation, the temperature rise because of the fan power and duct heat gain may increase to 10°F (5.6°C), which results in a supply air temperature of cold air up to 60°F (15.6°C).

When the outdoor air temperature T_o drops from 70 to 60°F (21.1 to 15.6°C), the mixing of minimum outdoor air and recirculating air lowers the hot deck discharge air temperature $T_{h,\text{dis}}$ from 80 to 76°F (26.7 to 24.4°C). If $T_o < 50$°F (10.0°C), heating may be required in the perimeter zone, so $T_{h,\text{dis}}$ increases as T_o falls. If the duct heat loss is 2°F (1.1°C), the warm air supply temperature differential $T_{h,\text{dis}} - 2 - T_{rxn}$ should not exceed 15°F (8.3°C) even at winter design conditions, in order to avoid stratification.

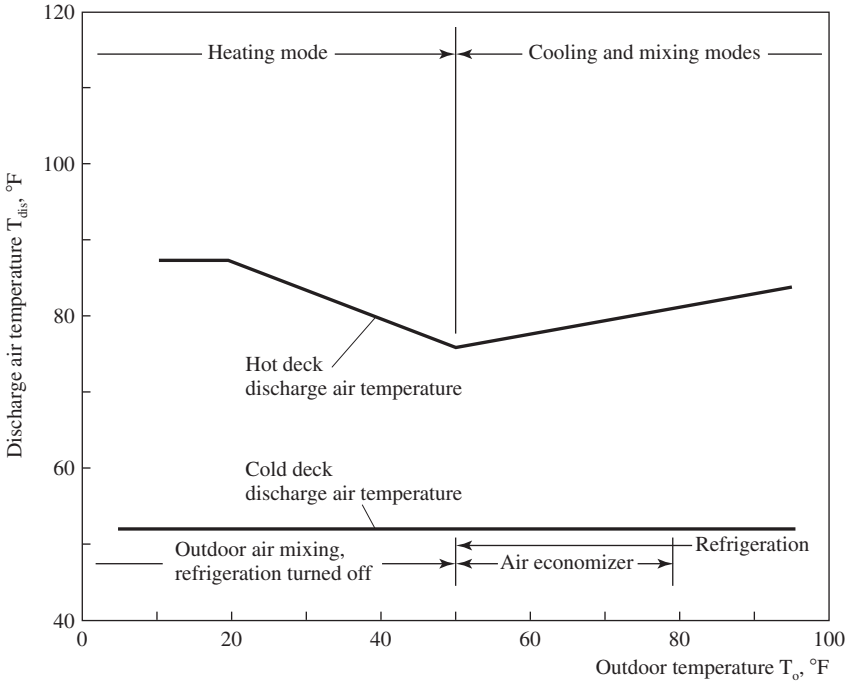


FIGURE 21.11 Warm and cold deck discharge air temperature in a dual-fan dual-duct VAV system.

Zone Supply Volume Flow Rate

For each control zone in the perimeter zone served by a mixing VAV box, an air leakage of 0.03 to 0.07 of its peak supply volume flow should be considered for the shut off damper. If the difference in air density between warm and cold air is ignored, the additional supply volume flow rate $\dot{V}_{lk}$ required to compensate for the air leakage when the warm air damper is shut, in cfm (L/s), can be calculated as

$$\dot{V}_{lk} = 0.05\dot{V}_{sxn}\frac{T_{s,h} - T_{rxn}}{T_{rxn} - T_{s,c}} \tag{21.11}$$

where $T_{s,h}$, $T_{s,c}$ = temperature of warm and cold air supply at mixing VAV box, °F (°C)
T_{rxn} = zone temperature, °F (°C)

As in a VAV reheat system, for any control zone in either the perimeter or interior zones, the mixing VAV box, VAV box, flexible ducts, and slot or ceiling diffusers should be sized according to the peak supply volume flow rate. If there is no air leakage, the control zone peak supply volume flow rate $\dot{V}_{rxn}$, in cfm [m^3/(60 s)], is

$$\dot{V}_{rxn} = \frac{Q_{rsxn}}{60\rho_s c_{pa}(T_{rxn} - T_{s,c})} \tag{21.12}$$

where Q_{rsxn} = control zone maximum sensible cooling load, Btu/h (W). The control zone peak supply volume flow rate for cold air in the perimeter zone at summer design conditions is

$$\dot{V}_{cxn} = \dot{V}_{rxn} + \dot{V}_{\text{lk}} \tag{21.13}$$

For any control zone in the perimeter zone, the peak supply volume flow rate $\dot{V}_{sxn}$ delivered to the conditioned space from the mixing VAV box, including 5 percent air leakage from the warm air damper at summer design conditions, in cfm (L/s), is therefore

$$\dot{V}_{sxn} = \frac{\dot{V}_{cxn}}{0.95} \tag{21.14}$$

Because only cold air from the VAV box is supplied to the interior zone, for any control zone in the interior zone, the peak supply volume flow rate $\dot{V}_{sin}$ at summer design conditions, in cfm (L/s), is then

$$\dot{V}_{sin} = \dot{V}_{rin} \tag{21.15}$$

Peak supply volume flow rate for control zone n, $\dot{V}_{sin}$, can be calculated by Eq. (21.12).

Case Study: A Dual-Fan Dual-Duct VAV System

Consider a dual-fan dual-duct VAV system serving a typical 20,000-ft^2 floor in an office building with the same operating parameters as the VAV reheat system described in Sec. 21.3, except the winter outdoor design temperature is 30°F (− 1.1°C) and the outdoor humidity ratio is 0.003 lb/lb (kg/kg). As in the VAV reheat system, the maximum supply volume flow rate for the interior zone at summer design conditions based on block load is

$$\dot{V}_{si} = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_{s,c})}$$

$$= \frac{135{,}000}{60 \times 0.075 \times 0.243(75 - 55)} = 6173 \text{ cfm } (2913 \text{ L/s})$$

For a control zone in the perimeter zone, if the temperature of the mixture of outdoor and recirculating air T_{mx} = 80.5°F (26.9°C), humidity ratio w_{mx} = 0.010 lb/lb (kg/kg), $T_{sx} = T_{s,c}$ = 55°F (12.8°C), and $\varphi_{sx} = \varphi_{s,c}$ = 80 percent, draw line *sx-mx* as shown in Fig. 21.9*b*.

For a warm air leakage of $0.05\dot{V}_{sxn}$ from the inlet damper, the state point of the mixture of cold air supply and warm air leakage *sxn* can be determined. From the psychrometric chart, T_{sxn} = 56.5°F (13.6°C), and w_{sxn} = 0.0083 lb/lb (kg/kg).

The volume flow rate of cold air supply to the perimeter zone at summer design conditions, based on the block load and including 5 percent air leakage from the warm air inlet damper, is therefore

$$\dot{V}_{scx} = \frac{285{,}500}{60 \times 0.075 \times 0.243(75 - 56.5)} = 14{,}113 \text{ cfm } (6660 \text{ L/s})$$

The total cold air supply volume flow rate to both the perimeter zone and the interior zone at summer design conditions is

$$\dot{V}_{s,c} = \dot{V}_{scx} + \dot{V}_{sci}$$

$$= 14{,}113 + 6713 = 20{,}826 \text{ cfm } (9828 \text{ L/s}) \tag{21.16}$$

The volume flow rate of warm air supply in the perimeter zone is usually expressed as a percentage of peak supply volume flow rate of cold air in the perimeter zone, usually between $0.5\dot{V}_{scx}$ and $1.0\dot{V}_{scx}$. It depends mainly on the space heating load linear density $q_{h,\text{ft}}$, in Btu/h · ft (W/m), and the

supply temperature differential $T_s - T_r$). If the volume flow rate of the warm air supply in the perimeter zone at winter design condition $\dot{V}_{shx}$, in cfm (L/s), is $0.7\dot{V}_{scx}$, then

$$\dot{V}_{shx} = 0.7 \times 14{,}113 = 9879 \text{ cfm } (4661 \text{ L/s})$$

Other parameters such as cooling and heating coil loads can be calculated as for VAV reheat systems.

Winter Heating and Winter Cooling Mode Operation

In a dual-fan dual-duct VAV system, the cold deck and the hot deck have their own outdoor and recirculating air supply. In the hot deck during winter heating mode operation, it is possible to have an outdoor air ratio of only about 22 percent and a mixing temperature of outdoor and recirculating air $T_{mx} = 62°F$ (16.7°C) for the perimeter zone, so that only necessary heating energy is provided for a hot deck discharge air temperature of 87°F (30.6°C), as shown in Fig. 21.9*b*. In the cold deck winter cooling mode operation, nearly 60 percent outdoor air is required to form a mixture of outdoor and recirculating air with a temperature T_{mi} of about 50°F (10.0°C) and a cold deck discharge air temperature of 52°F (11.1°C), so that outdoor air free cooling can be fully utilized to offset the zone cooling load in the interior zone in winter.

Part-Load Operation

Consider a typical control zone in the perimeter zone of this dual-fan, dual-duct VAV system whose operating parameters at summer design condition and summer part-load operation are as follows:

	Full load	Part load
Zone sensible cooling load, Btu/h	14,200	4,260
Leaving cooling-coil temperature, °F	50	50
Relative humidity, percent	95	95
Supply system heat gain, °F (°C)	5 (2.8)	10 (5.6)
Warm air supply temperature, °F (°C)	87 (30.6)	80 (26.7)

If the supply air temperature discharged from the mixing VAV box $T_{sxn} = T_{m2} = 56.5°F$ (13.6°C), the volume flow rate of conditioning air supplied to this control zone $\dot{V}_{sxn}$, in cfm (L/s), including 5 percent warm air leakage from the damper at summer design conditions, is therefore

$$\dot{V}_{sxn} = \frac{14{,}200}{60 \times 0.075 \times 0.243(75 - 56.5)} = 702 \text{ cfm } (331 \text{ L/s})$$

During part-load operation, as the zone sensible cooling load is reduced to 30 percent of the design load, $T_{sxnp} = 50 + 10 = 60°F$ (15.6°C). Assuming that the percentage of warm air leakage and the temperature rise due to this warm air leakage are the same at design condition, the supply temperature of the mixture of cold and warm air of control zone *xn* in the perimeter zone at part-load T_{sxnp} is now equal to $50 + 10 + 1.5 = 61.5°F$ (16.4°C).

The supply volume flow rate from the mixing VAV box in the perimeter zone at part-load can be calculated as

$$\dot{V}_{sxnp} = \frac{Q_{rsxnp}}{60\rho c_{pa}(T_{rxn} - T_{sxnp})} \tag{21.17}$$

$$= \frac{4260}{60 \times 0.075 \times 0.243(75 - 61.5)} = 289 \text{ cfm } (136 \text{ L/s})$$

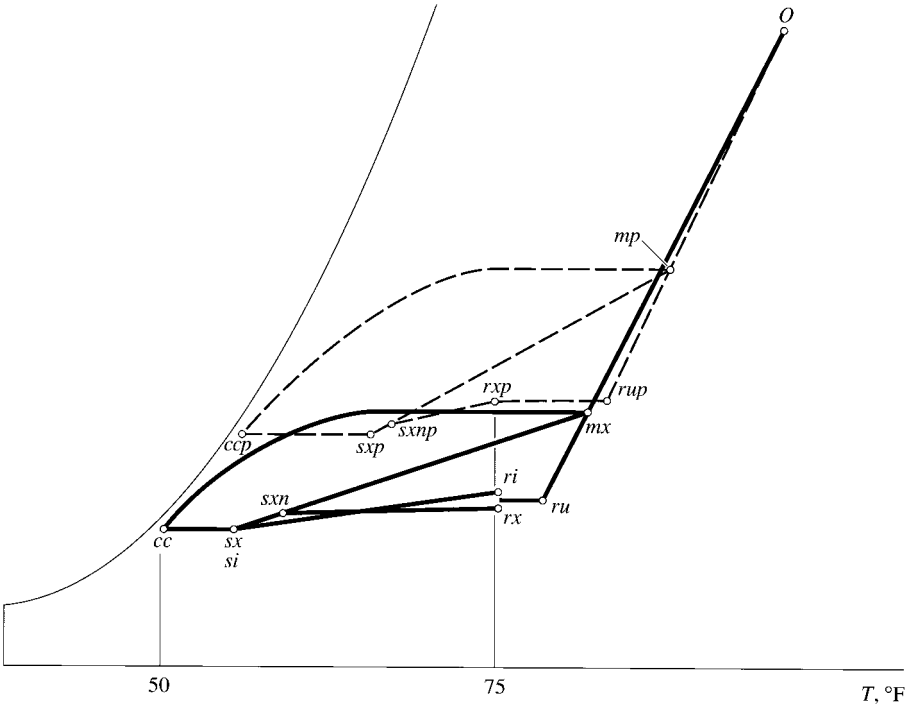


FIGURE 21.12 Air conditioning cycle of a typical control zone at summer part-load operation served by a mixing VAV box of a dual-fan dual-duct VAV system.

From Eq. (21.14), the volume flow rate of cold air supply at part load can be calculated as

$$\dot{V}_{s,cn} = 0.95 \times 289 = 275 \text{ cfm } (130 \text{ L/s})$$

Also the warm air supply is

$$\dot{V}_{s,hn} = 289 - 275 = 14 \text{ cfm } (6 \text{ L/s})$$

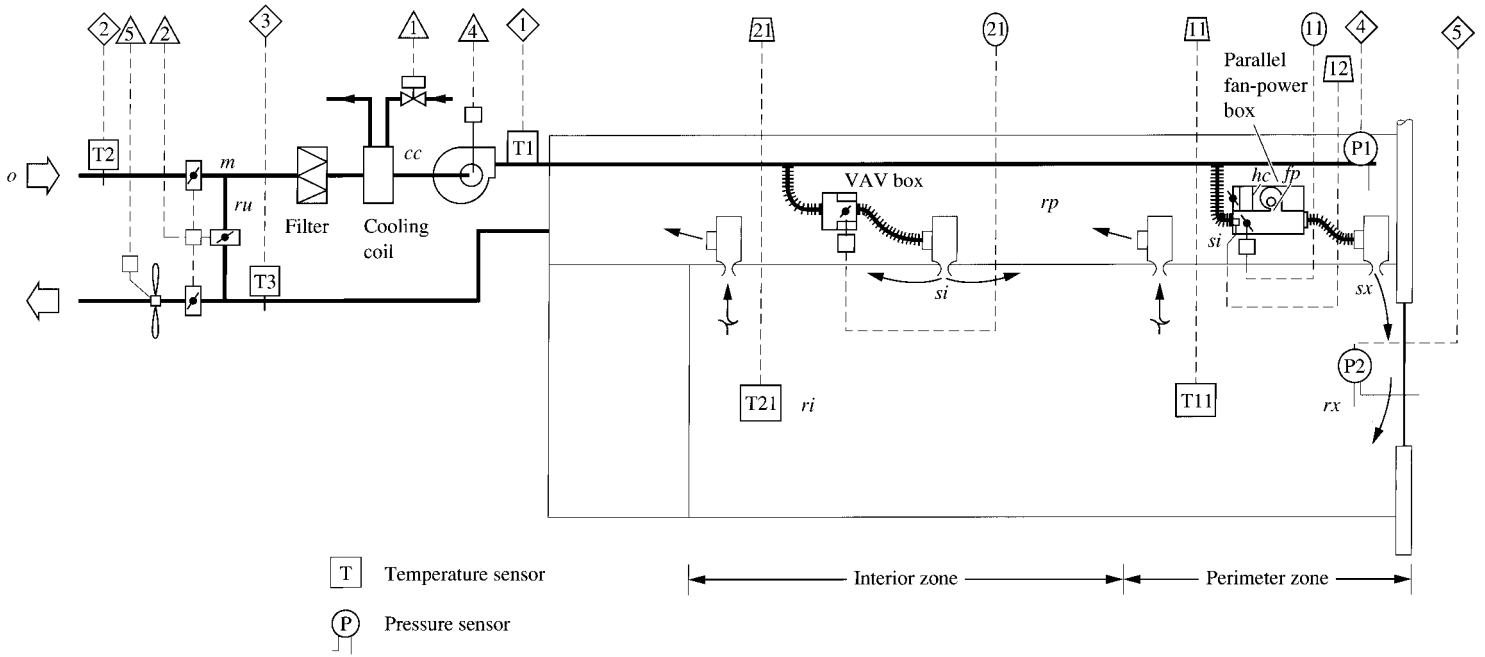
Figure 21.12 shows the air conditioning cycle of a typical control zone in the perimeter zone served by a dual-fan dual-duct VAV system at summer cooling mode part-load operation. From the psychrometric chart, zone relative humidity at summer part-load operation is higher than at full-load condition. This is mainly because of the increase of the cold deck discharge air temperature due to the duct heat gain and the possible chilled water reset at part load.

At winter heating mode part-load operation, the state point of supply air will move along the horizontal line *mx-hc* as shown in Fig. 21.9*b*, to maintain a predetermined zone temperature.

21.5 FAN-POWERED VAV SYSTEMS

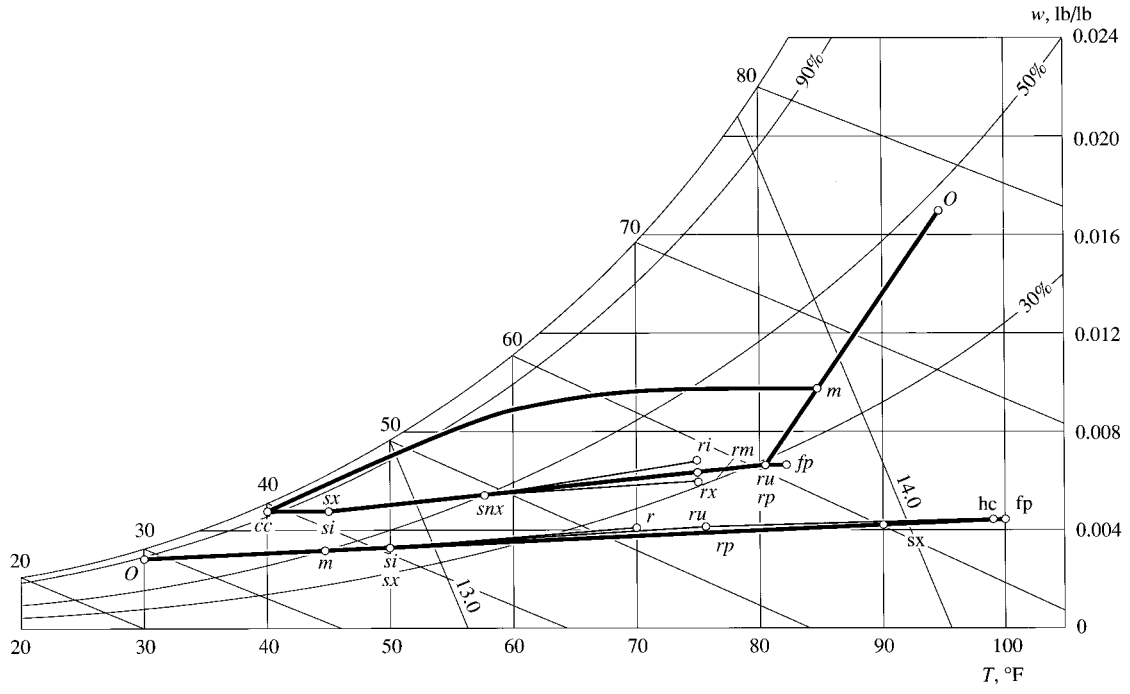
System Description

A fan-powered VAV system, as shown in Fig. 21.13, often employs fan-powered boxes in each control zone of the perimeter zone and VAV boxes in each control zone of the interior zone in a



(a)

FIGURE 21.13 A parallel fan-powered VAV system: (*a*) schematic diagram; (*b*) air conditioning cycle; (*c*) details of winter heating mode air conditioning cycle; (*d*) cold primary air and heating coil output diagram.



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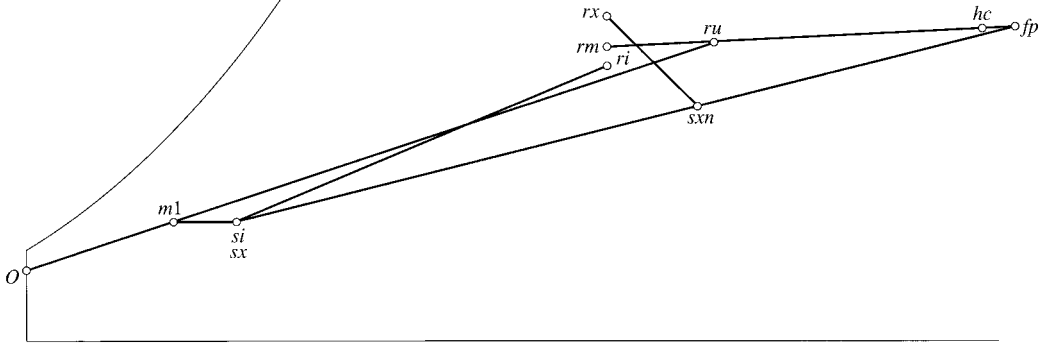


(c)

FIGURE 21.13 (*Continued*)

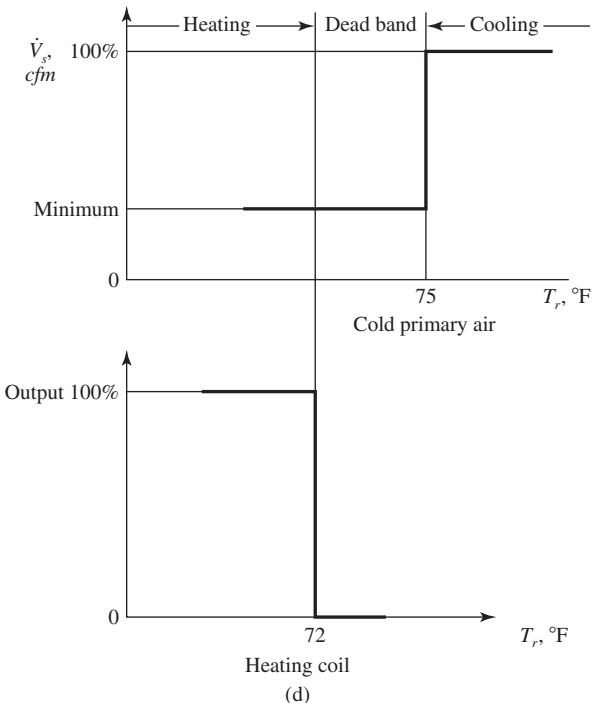


FIGURE 21.13 (*Continued*)

building. Both fan-powered boxes and VAV boxes can provide cooling for the perimeter and interior zones. The function and advantages of a fan-powered box include the following:

- It extracts warm recirculating air from the ceiling plenum to maintain a preset zone temperature when the volume flow of the cold primary air supplied from the fan-powered box has dropped to the minimum setting or other preset value at summer cooling part-load operation. This is the first step.
- If the zone temperature drops further in any of the control zones in the perimeter zone, a reheating coil in the fan-powered box is energized to maintain predetermined zone temperature during heating mode operation. This is the second step.
- It can increase the warm air supply volume flow rate to meet the requirement that the warm air supply temperature differential does not exceed 15°F (8.3°C), in order to prevent excessive buoyancy effects.
- It mixes the low-temperature primary air with warm plenum air to prevent the dump of a cold air jet and possible surface condensation in cold air distribution.

If the fan-powered box is used for mixing low-temperature primary air with warm plenum air in cold air distribution, fan-powered boxes can also be installed in the control zones of the interior zone. The disadvantages of using a fan-powered box instead of a VAV box include higher energy use, higher noise level, and greater initial cost and maintenance.

Fan-Powered VAV Boxes

A fan-powered VAV box, or simply a fan-powered box, consists of a small forward-curved centrifugal fan, a VAV box, a heating coil (hot water or electric heater), a low-efficiency disposable filter, dampers, an outer casing, and corresponding controls, as shown in Fig. 21.14. Fan-powered VAV boxes can be classified as series and parallel boxes according to their construction and operating characteristics.

Series Fan-Powered Box. In a series fan-powered box, a small centrifugal fan is connected in series with the cold primary airstream flowing through the VAV box, as shown in Fig. 21.14*a*. The volume flow of the cold primary air is varied at part-load operation, but after this air mixes with the induced plenum recirculating air, the fan delivers a continuously operated, nearly constant-volume flow rate of the mixture of cold primary air and plenum recirculating air. The mixture is then supplied to the control zones, either heated at the heating coil or without heating depending upon the zone load.

Because the airflow from the series fan-powered box is constant, it is easier to select ceiling diffusers to achieve good space air diffusion. In a series fan-powered box, cold primary air is thoroughly mixed with plenum recirculating air. The main disadvantage of a series fan-powered box is its larger fan, which consumes more energy than a parallel fan-powered box.

Parallel Fan-Powered Box. In a parallel fan-powered box, the induced recirculating plenum airstream discharged from the centrifugal fan is connected in parallel with the cold primary airstream discharged from the VAV box in the fan-powered box. These two airstreams are mixed either before or after the heating coil and then are supplied to the control zones.

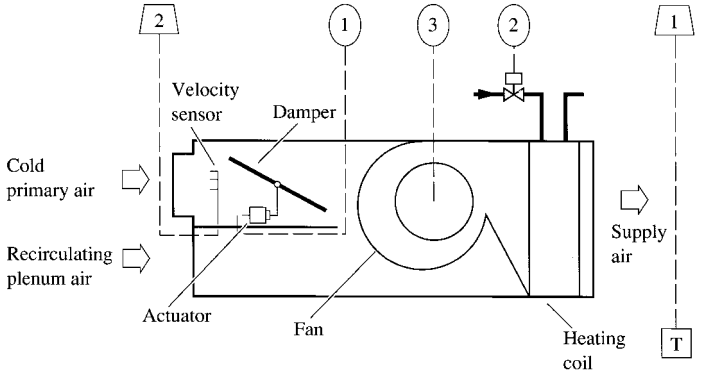
A parallel fan-powered box with a heating coil located upstream from the fan is called a draw-through parallel fan-powered box, as shown in Fig. 21.14*b*. When the heating coil is located downstream from the fan, the box is called a blow-through parallel fan-powered box, shown in Fig. 21.14*c*. The air temperature entering the heating coil of a draw-through unit is the temperature of recirculating plenum air, which is higher than that of the mixture of cold primary air and recirculating plenum air in a blow-through box.

A parallel fan-powered box operates only when there is a need to mix the warm plenum air with the cold primary air at summer cooling mode part-load operation, when the volume flow of the cold primary air is at its minimum setting or other preset value, or when the mixture of cold primary air and warm plenum air is heated at the heating coil to offset the zone winter heating load. When a parallel fan-powered box is used to mix 40 to 50°F (4.4 to 10.0°C) low-temperature primary air with warm plenum air in a cold air distribution system, it is also operated continuously during summer cooling mode.

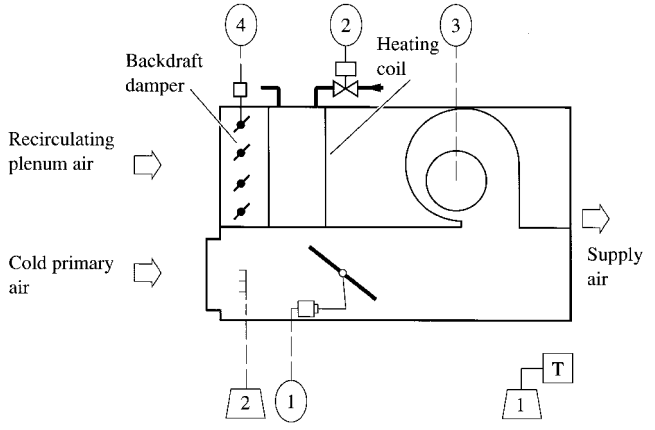
In a parallel fan-powered box when the cold primary airstream is reduced to its minimum setting during cooling mode part-load operation, the volume flow rate of the extracted plenum recirculating air plus the cold primary air is usually equal to 0.80 to 1.0 of the zone summer peak supply volume flow rate $\dot{V}_{sn,d}$. The fan in a parallel fan-powered box extracts only recirculating plenum air, and therefore it has a smaller volume flow rate and a lower fan total pressure than that in a series fan-powered box. Because of its smaller fan size, it consumes less fan power in cooling mode operation.

Current parallel fan-powered box designs also provide a good mixture of cold primary air and extracted plenum air. Some of the recent models use a variable-speed small parallel fan. It energizes as soon as the supply volume flow of the cold primary air drops at cooling mode part-load operation. The extracted recirculating plenum air gradually increases as the volume flow of cold primary air decreases up to a maximum limit between $0.4\dot{V}_{sn,d}$ and $0.5\dot{V}_{sn,d}$. Under this condition, the zone supply volume flow rate is kept nearly constant. Parallel fan-powered boxes are often used in cold air distribution because of their lower initial cost and lower fan energy consumption.

In a parallel fan-powered box, a backdraft damper should be installed upstream from the fan. This damper is normally closed to prevent the backward flow of cold primary air in case the fan is deenergized. When the fan is turned on, the backdraft damper is fully opened.



(a)



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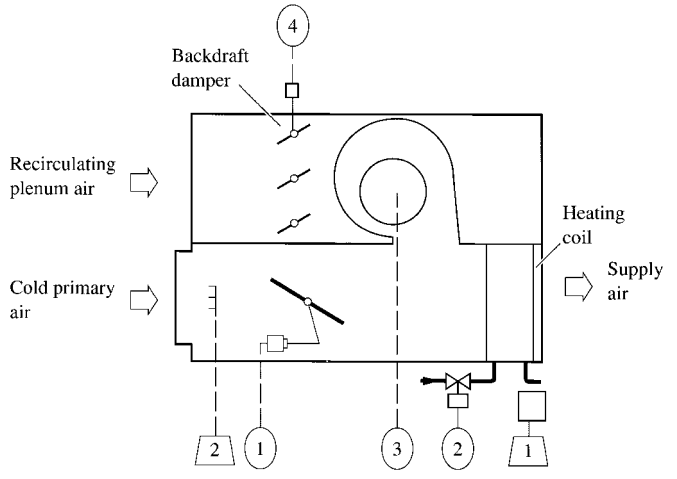

(c)

FIGURE 21.14 Fan-powered boxes: (*a*) series box; (*b*) parallel box, draw-through; (*c*) parallel box, blow-through.

The drawbacks of a parallel fan-powered box include the need for a careful balance of flow and total pressure of two airstreams, the difficulty in selection of the size of the box, and its more complicated controls.

The size and capacity of a fan-powered box is determined by the capacity of the fan, the VAV box, and the heating coil. The heating coil in a fan-powered box can be a hot water coil or an electric duct heater located downstream from the fan-powered box. Fan-powered boxes are usually built in sizes corresponding to the associated VAV box primary air volume flow rates from 300 to 3200 cfm (150 to 1600 L/s) and fan supply volume flow rates of 250 to 4000 cfm (125 to 2000 L/s). Most fan-powered boxes have a fan total pressure between 0.5 and 0.8 in. WC (125 and 200 Pa) and an external pressure loss of about 0.25 in. WC (63 Pa) to offset the pressure drop of downstream flexible ducts and diffusers. The pressure drop of the VAV box and the corresponding pressure drop of flexible ducts and diffusers due to cold primary air should be offset by the supply fan in the primary air AHU or PU.

The speed of the centrifugal fan in the fan-powered box can be adjusted to high, medium, or low or even variable-speed drive. Both radiated and duct-borne sound paths should be checked to determine whether they can meet the space air required NC criteria. Because the fan total pressure is higher in a fan-powered box than in a fan-coil unit, a temperature rise due to fan power heat gain of 0.75°F (0.4°C) should be considered in psychrometric analysis.

The VAV box a in fan-powered box can be either pressure-dependent or pressure-independent. Pressure-independent fan-powered boxes are widely used.

Fan Characteristics in Parallel Fan-Powered Boxes

In a fan-powered VAV system using a parallel fan-powered box, two fan-duct systems are connected in parallel: the cold primary airstream discharged from the supply fan F1 in the AHU or PU and the recirculating plenum airstream extracted by fan F2, as shown in Fig. 21.15. At the combining point *A*, the pressures of these two airstreams must be equal, and the volume flow rate is the sum of the volume flow rates of the cold primary air $\dot{V}_{sn,c}$ and the recirculated plenum air $\dot{V}_{sn,r}$.

For a control zone in the perimeter zone using a parallel fan-powered box at its zone peak supply volume flow rate with conventional air distribution $\dot{V}_{sn,d}$ (a supply air temperature of 55°F, or 12.8°C), in cfm (L/s), the distribution is as follows:

Zone supply volume flow rate of control zones at summer design conditions	$1.0\dot{V}_{sn,d}$
Zone supply volume flow of recirculating plenum air from parallel fan-powered box	0

According to Yu et al. (1993), the zone supply volume flow rates of control zones in the interior zone using a parallel fan-powered box with cold air distribution at summer design conditions are as follows:

Supply volume flow rate of 44°F (6.7°C) cold primary air, cfm (L/s)	$0.6\dot{V}_{sn,d}$
Supply volume flow rate from fan-powered box (recirculating plenum air), cfm (L/s)	$0.4\dot{V}_{sn,d}$

Zone supply volume flow rate for control zones in the perimeter zone at winter heating design conditions must meet the following requirements:

Cold primary air for ventilation	$0.3\dot{V}_{sn,d}$
Supply temperature differential $T_s - T_r$	$\leq$ 15°F (8.3°C)

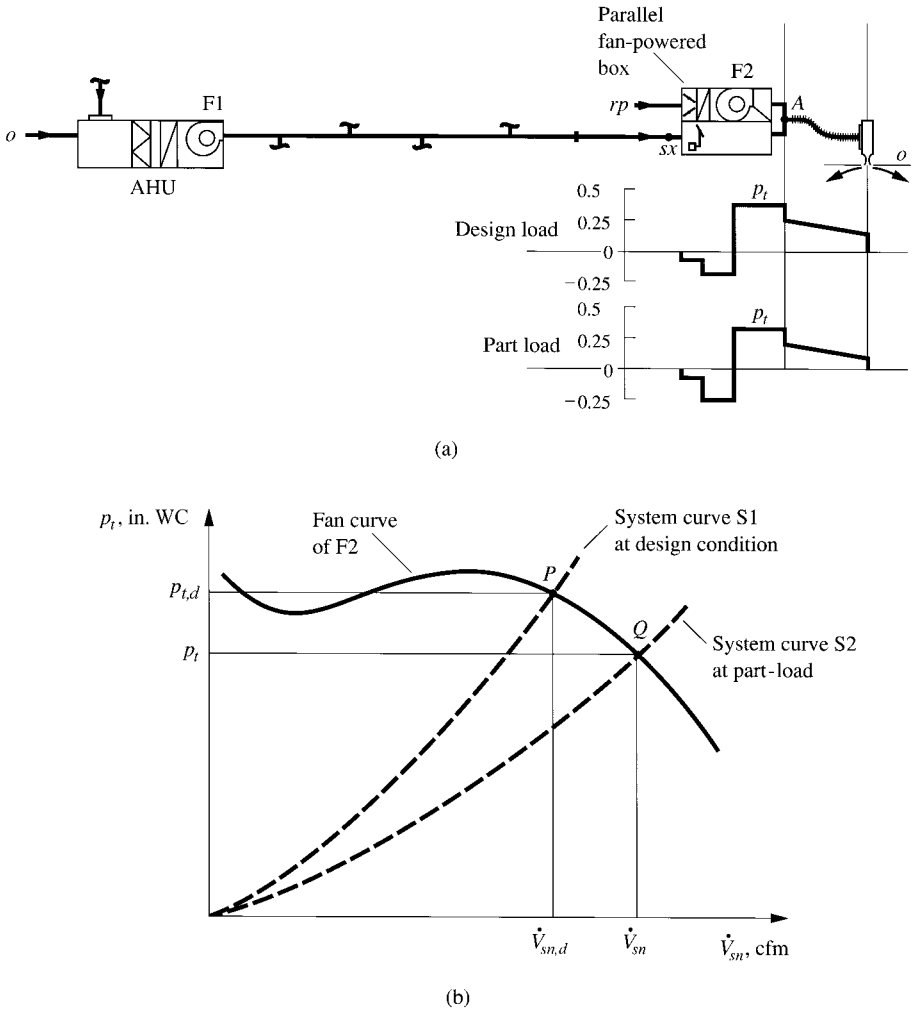

FIGURE 21.15 Fan operating characteristics of a parallel fan-powered VAV system: (*a*) pressure characteristics; (*b*) fan-duct system characteristics.

In a fan-powered VAV system, the fan total pressure of F2 in Fig. 21.15*a* is about 0.5 in. WC (125 Pa), and the pressure drop across the flexible duct and the slot diffuser may be around 0.25 in. WC (63 Pa).

During summer cooling mode operation, when the zone sensible cooling load is reduced to about one-half of the design load, the volume flow rate of the cold primary air may be reduced from $0.6\dot{V}_{sn,d}$ to $0.3\dot{V}_{sn,d}$. As the supply volume flow through the flexible duct and slot diffuser is reduced accordingly, the pressure drop between points *A* and *o* also decreases. The system pressure drop of the recirculating plenum air passage becomes smaller, and a new system curve S2 is formed, as shown in Fig. 21.15*b*. The operating point of fan-duct system F2 and S2 moves from point *P* to point *Q*. The result is a greater volume flow rate of the recirculating plenum air $\dot{V}_{sxn,r}$ than that at summer design conditions, and a smaller zone supply volume flow rate flowing through the flexible duct and slot diffusers.

Zone Control and Sequence of Operations of a Fan-Powered VAV System with Parallel Fan-Powered Box

Consider a control zone *rx1* in the perimeter zone served by a parallel fan-powered box FBX1 that mixes low-temperature cold primary air with warm plenum air, and a control zone *ri1* in the interior zone in which the cold primary air is directly supplied from the VAV box, as shown in Fig. 21.13. Because the temperature of cold primary air from the AHU that serves fan-powered box FBX1 and VAV box VI1 $T_{c,p} = 44°F$ (6.7°C), the summer indoor space relative humidity drops to about 40 percent. The summer indoor design temperature is still maintained at 75°F (23.9°C). At winter design conditions, the zone temperature is maintained at 72°F (22.2°C) with a relative humidity of 20 to 30 percent.

1. When the time-of-day clock signals the AHU to "off," the AHU and the fan-powered box FBX1 and VAV box VI1, as shown in Fig. 21.13*a*, are deenergized.

2. When the clock signals the AHU in the on position, if the zone temperature sensed by the temperature sensor T11, $T_{rxn} > 75°F$ (23.9°C), then the AHU and fan-powered box FBX1 and VAV box VI1 call for cooling. The DDC system controller then opens the two-way valve of the water cooling coil and resets the discharge air temperature from the AHU $T_{dis} = 44°F$ (6.7°C). The DDC terminal controllers also open the single-blade volume dampers in the fan-powered boxes and VAV boxes to the maximum cooling position.

If the outdoor temperature $T_o > 75°F$ (set point of the temperature economizer cycle), the DDC system controller also positions the outdoor damper to a minimum opening position and fully opens the recirculating damper.

3. In the perimeter zone, the control zone temperature T_{rx1} is sensed by sensor T11. The DDC terminal controller modulates the volume damper and thus the volume flow of cold primary air supply in the fan-powered box FBX1 to maintain a preset zone temperature of $T_{rx1} = 75°F$ (23.9°C) with a PI control mode. The fan in the fan-powered box FBX1 is not energized.

The control zone temperature in the interior zone T_{rin} is sensed by sensor T21. Another DDC terminal controller modulates the volume damper in the VAV box VI1 to maintain a preset zone temperature $T_{ri1} = 75°F$ (23.9°C).

4. When $52°F \le T_o \le 75°F$ ($11.1°C \le T_o \le 23.9°C$), the DDC system controller fully opens the outdoor air damper and closes the recirculating damper to use 100 percent outdoor air free cooling as the first-stage cooling in the AHU. Based on the measured discharge air temperature by sensor T1, the DDC system controller modulates the two-way valve of the water cooling coil as the second-stage cooling simultaneously to maintain a preset discharge air temperature $T_{dis} = 44°F$ (6.7°C).

When the outdoor temperature T_o drops from 65 to 52°F (18.3 to 11.1°C), T_{dis} is reset inversely proportional by the DDC system controller from 44 to 52°F (6.7 to 11.1°C). At the same time, the DDC terminal controllers modulate the volume damper as well as the volume flow of cold primary air supply to maintain a preset zone temperature of 75°F (23.9°C) in the control zones *rx1* and *ri1*. The fan in the fan-powered box FBX1 is not energized.

5. When $52°F \le T_o \le 75°F$ ($11.1°C \le T_o \le 23.9°C$) and the supply volume flow rate of the cold primary air from the fan-powered box FBX1 in the perimeter zone has dropped to $0.40\dot{V}_{sn,d}$, the fan in the fan-powered box FBX1 is energized in low-speed operation and extracts recirculating ceiling plenum air to the fan-powered box, and the supply volume flow rate of recirculating plenum air $\dot{V}_{sn,r}$ to the control zone *rx1* is roughly increased to about $0.40\dot{V}_{sn,d}$.

If the volume flow rate of the cold primary air supplied from the fan-powered box FBX1 has dropped to $0.35\dot{V}_{sn,d}$, the fan in the fan-powered box is energized in high-speed operation, and its supply volume flow of recirculating plenum air $\dot{V}_{sn,r}$ is further increased to $0.45\dot{V}_{sn,d}$.

6. When $52°F \le T_o \le 75°F$ ($11.1°C \le T_o \le 23.9°C$) and the supply volume flow rate of the cold primary air from the fan-powered box FBX1 has dropped to the minimum setting, that is, $0.3\dot{V}_{sn,d}$, if the control zone temperature T_{rx1} sensed by sensor T11 drops below 75°F (23.9°C), that is, $72°F \le T_{rx1} \le 75°F$ ($22.2°C \le T_{rx1} \le 23.9°C$), then the fan-powered box FBX1 is in dead-band operation, as shown in Fig. 21.13*d*. In dead-band operation

- The supply volume flow rate of the cold primary air is at the minimum setting.
- The fan in the fan-powered box FBX1 is still energized.
- The heating coil is not energized.

7. When $T_o < 75°F$ (23.9°C) and the supply volume flow rate of cold primary air from the fan-powered box FBX1 has dropped to the minimum setting, if the control zone temperature T_{rx1} sensed by sensor T11 drops below 72°F (22.2°C), the fan-powered box is in heating mode operation. In heating mode operation, the extracted recirculating plenum air is heated at the heating coil, flows through the centrifugal fan, mixes with the cold primary air, and is then supplied to the control zone, as shown in Fig. 21.13*b*. The DDC terminal controller opens and modulates the mass flow rate of hot water entering the heating coil to maintain a control zone temperature $T = 72°F$ (22.2°C) in reverse-acting mode.

The operation of the fan in a DDC fan-powered box can be arranged in different alternatives.

Supply Volume Flow Rate

According to Eq. (20.69), at summer design conditions, the volume flow rate of cold primary air supplied to a control zone with conventional air distribution $\dot{V}_{sn,d}$ in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sn,d} = \frac{Q_{rsn}}{60\rho_s c_{pa}(T_r - T_s)}$$

where Q_{rsn} = zone peak sensible cooling load, Btu/h (W) and

$$T_r - T_s = 20°F\ (11.1°C)$$

At summer design conditions, the volume flow rate of cold primary air supplied to a control zone with cold air distribution $\dot{V}_{sn,c\,\mathrm{dis}}$ in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sn,c\,\mathrm{dis}} = K\dot{V}_{sn,d} = \frac{Q_{rsn}}{60\rho_s c_{pa}(T_r - T_{s,c\,\mathrm{dis}})}$$

where Q_{rsn} = zone peak sensible cooling load, Btu/h (W)
K = a factor
$T_r - T_{s,c\,\mathrm{dis}} = 32°F\ (17.8°C)$

At winter design conditions, the volume flow rate of the mixture of cold primary air and recirculating plenum air $\dot{V}_{sn,h}$, in cfm [$m^3/(60\ s)$], supplied to a control zone in the perimeter zone can be calculated as

$$\dot{V}_{sn,h} = \frac{Q_{rnh}}{60\rho_s c_{pa}(T_{s,h} - T_r)} \tag{21.18}$$

where Q_{rnh} = zone heating load, Btu/h (W)
$T_{s,h}$ = supply temperature at winter design conditions, °F (°C)

Warm air supply temperature differential $T_{s,h} - T_r$ should not exceed 15°F (8.3°C) in order to prevent buoyancy effects.

Supply volume flow rate of a control zone in the perimeter zone served by a fan-powered VAV system with cold air distribution is shown in Fig. 21.16. In Fig. 21.16: $\dot{V}_{sn,c}$, $\dot{V}_{sn,r}$ indicate supply volume flow rate of cold primary air and recirculated plenum air to the perimeter control zone, in cfm (L/s), and $T_{s,n}$ the supply air temperature to the perimeter control zone *xn*, in °F (°C).

As in the VAV reheat system, the cooling and heating coil load can be similarly calculated by Eqs. (21.8), (21.9), and (21.10).

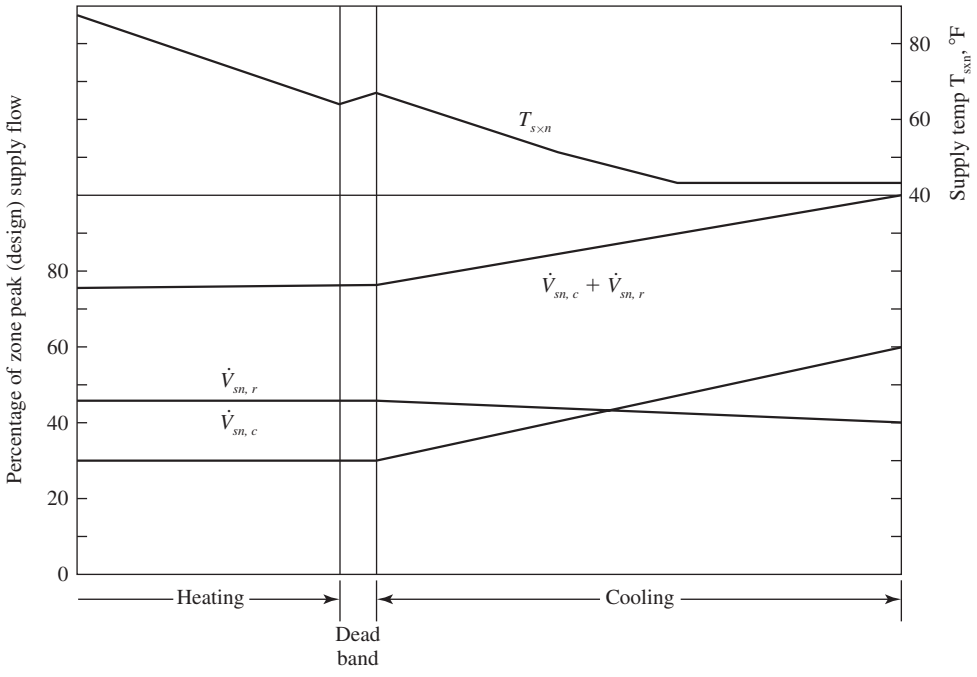

FIGURE 21.16 Supply volume flow rate of a control zone in the perimeter zone served by a fan-powered VAV system.

Fan Energy Use

In a fan-powered VAV system, there are two kinds of fan energy use: the supply fan energy use in an AHU or a PU and the energy use of the small centrifugal fans in the fan-powered boxes. Typical supply fan efficiency is about 60 percent, and the motor efficiency is around 90 percent, which gives a combined fan and motor efficiency of about 55 percent.

According to Elleson (1993), the efficiency of fans in fan-powered boxes from four manufacturers ranged from 16 to 35 percent, and the motor efficiency ranged from 45 to 55 percent. Assuming a fan efficiency of 30 percent and a motor efficiency of 50 percent, the combined fan and motor efficiency for the fan in the fan-powered box is 15 percent.

Elleson (1993) has collected detailed data for the 1988 cooling season from the fourth floor of a 12-story building in southern California using a fan-powered VAV system (parallel fan-powered boxes). The results are as follows:

	55°F (12.8°C) Supply air			45°F (7.2°C) Supply air		
	Supply fan	Fan-powered boxes	Total	Supply fan	Fan-powered boxes	Total
Continuous operation, kWh	11,720	7,000	18,720	7,794	7,000	14,794
Intermittent operation, kWh	11,720	1,861	13,581	7,794	2,140	9,934
No fan-powered boxes, kWh	11,720	0	11,720	8,417	0	8,417

Intermittent operation indicates that fans in parallel fan-powered boxes operated only when the cold primary airflow fell below 50 percent of design flow.

If series fan-powered boxes and parallel fan-powered boxes are both operated continuously, the energy use of the supply fan plus the fans in the parallel fan-powered boxes consume less than one-half of the kWh used by the supply fan plus the fans in the series fan-powered boxes in 45°F (7.2°C) cold air distribution.

Design Considerations

- Because there are two mixings (the mixing of outdoor air and recirculating air in the AHU or PU and the mixing of cold primary air and plenum air in the fan-powered box), adequate supply of outdoor air is critical in a fan-powered VAV system. This is discussed in Chap. 23.
- In a VAV system using cold air distribution, fan-powered boxes can be installed in both the perimeter and interior zones, or in the perimeter zone only, for lower initial cost. Cold primary air is supplied directly to the interior zone by high-induction ceiling diffusers, as shown in Fig. 18.10*a* and *b*.
- The mixing ratio of cold primary air to zone supply air and the reset of the cold primary air supply temperature for the perimeter zone are different from those of the interior zone. Different AHUs or PUs are preferable for the perimeter zone and interior zone separately. An analysis of energy savings and investment costs is often necessary.
- In a series fan-powered box, the supply fan in the AHU or PU offsets the system pressure drop up to the inlet of the VAV box. The fan in the fan-powered box offsets the pressure drop of the VAV box, elbows (if any), downstream heating coil, flexible duct, and diffuser.

  In a parallel fan-powered box, the supply fan in the AHU or PU offsets the pressure drop of the VAV box, flexible duct, diffuser, and straight-through mixing loss of the cold primary airstream in the fan-powered box.
- At summer and winter design conditions, the ratio of the volume flow rate of cold primary air supplied to the fan-powered box to the zone supply volume flow rate $\dot{V}_{sn,c}/\dot{V}_{sn,d}$ and the ratio of the volume flow rate of warm recirculating plenum air to the zone supply volume flow rate $\dot{V}_{s,\text{rec}}/\dot{V}_{sn,d}$ depend on the following factors: (1) zone sensible cooling load and heating load, (2) zone supply temperature differential at summer and winter design conditions $T_{rn} - T_{sn,d}$, or $T_{sn,d} - T_{rn}$, and (3) outdoor air requirements.

  As mentioned previously, for cold air distribution $\dot{V}_{sn,c}/\dot{V}_{sn,d}$ is about 0.6 for a fan-powered box. The winter heating in the perimeter zone often needs greater supply volume flow to match the requirement of the warm air supply temperature differential not exceeding 15°F (8.3°C); $\dot{V}_{sn,h}/\dot{V}_{sn,d}$ for cold air distribution is about 0.7.
- The supply main ducts, branch ducts, cold air passage in the VAV box, flexible ducts, and diffusers must be well insulated especially for cold air distribution. Cold primary air and recirculated plenum air should be mixed in a perpendicular flow direction to prevent downstream temperature stratification.
- A low zone relative humidity and zone dew point are desirable to prevent possible surface condensation during cold air distribution. When cold primary air at 44°F (6.7°C) is supplied directly to the conditioned space through high induction diffusers, the surface of the induction diffusers can be assumed to be 3°F (1.7°C) higher than the supply air; or the dew point of zone air should be lower than 44 + 3 = 47°F (8.3°C), that is, a zone temperature of 75°F (23.9°C) and a space relative humidity $\varphi_r < 37$ percent.
- At the start of cool-down period in a hot and humid summer, direct supply of 40 to 50°F (4.4 to 10.0°C) cold primary air to the fan-powered boxes may cause condensation because of the higher dew point of stagnant air in the ceiling plenum before the cool-down period. The AHU or PU should supply 55°F (12.8°C) air to the fan-powered boxes first, to reduce moisture and lower the dew point.
- If humidification is required to maintain an acceptable zone relative humidity in winter, a humidifier can be installed in the AHU or PU.

- During the unoccupied period in winter, the AHU or PU can be shut off while the fan-powered boxes provide the necessary heating.
- Fan-powered boxes installed inside the ceiling plenum must meet local safety code requirements.

21.6 COMPARISON BETWEEN VARIOUS VAV SYSTEMS

VAV systems are used in applications where space load is varied in order to maintain a required indoor environment and to save fan energy. Single-zone VAV systems are mainly used for single-zone conditioned spaces such as arenas, indoor stadiums, assembly halls, and many factories.

VAV reheat systems using reheating VAV boxes are simple and effective for perimeter heating in winter and with VAV boxes provide cooling only in the interior zone in a building. Simultaneous cooling and heating may occur in reheat systems at low zone cooling loads. The VAV reheat system becomes energy-inefficient for locations where the winter heating load linear density exceeds 120 Btu/h·ft (115 W/m).

Perimeter heating VAV systems are used for buildings that need heating for their perimeter zones in locations with cold and long winters.

Dual-fan dual-duct VAV systems provide satisfactory indoor environmental control for multizone high-rise buildings and health care facilities with flexible operating characteristics. If the shutoff air leakage in the mixing VAV box can be minimized, it is often an appropriate choice for more demanding applications.

Fan-powered VAV systems provide low cooling load compensation and winter heating from a two-stage sequence operation: intake warm plenum air and winter heating. The small centrifugal fan in the fan-powered box also provides the mixing of cold primary air with the plenum air. Fan-powered VAV systems are widely used in buildings where cold air distribution is needed, such as office buildings and schools.

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CHAPTER 22

AIR SYSTEMS: VAV SYSTEMS—FAN COMBINATION, SYSTEM PRESSURE, AND SMOKE CONTROL

22.1 RETURN AND EXHAUST SYSTEMS 22.2
- Types of Return and Exhaust Systems 22.2
- Return Ceiling Plenum 22.2
- Low-Level Return Systems and Enclosed Parking Garage Ventilation 22.2
- Exhaust Hoods 22.3

22.2 FAN COMBINATIONS AND OPERATING MODES 22.4
- Fan Combinations 22.4
- Operating Modes 22.4

22.3 SYSTEM PRESSURE DIAGRAM 22.5
- System Pressure and Duct Static Pressure Control 22.5
- Fan Characteristics 22.7
- Mixing-Exhaust Section and Conditioned Space 22.8

22.4 SUPPLY FAN AND EXHAUST FAN COMBINATION 22.8
- System Characteristics 22.8
- Operating Characteristics 22.9
- Recirculation Mode and Design Volume Flow 22.9
- Recirculation Mode, 50 Percent Design Flow Rate 22.12
- Air Economizer Cycle 22.13
- Warm-Up and Cool-Down Mode 22.13
- Pressure Variation at the Mixing Box 22.13

22.5 SUPPLY FAN AND RELIEF FAN COMBINATION 22.14
- Recirculation Mode 22.14
- Air Economizer Mode and Design Volume Flow Rate 22.14
- Air Economizer Mode, 50 Percent Design Flow 22.17
- Warm-Up and Cool-Down Mode 22.17
- Design Considerations and Controls 22.17

22.6 SUPPLY FAN AND RETURN FAN COMBINATION 22.18
- Recirculation Mode 22.18
- Air Economizer Mode 22.20
- Controls 22.21

22.7 COMPARISON OF THREE FAN COMBINATION SYSTEMS 22.21

22.8 PRESSURE FLOW CHARACTERISTICS 22.22
- Pressure Flow Characteristics for a Supply and Return Fan Combination System 22.22
- Variation of Pressure in the Mixing Box 22.23
- Field Survey of System Pressure Characteristics of a VAV System Using a Supply and Relief Fan Combination 22.23

22.9 SMOKE CONTROL AND FIRE SAFETY 22.24
- Fire Safety in Buildings 22.24
- Smoke Movements in Buildings 22.25
- Effective Area and Flow Rate 22.27

22.10 EFFECT OF AUTOMATIC SPRINKLER ON FIRE PROTECTION 22.27

22.11 SMOKE CONTROL IN ATRIA 22.28
- ANSI/NFPA 92A and 92B 22.28
- Smoke Management in Atria, Malls, and Large Areas 22.28

22.12 STAIRWELL PRESSURIZATION AND ZONE SMOKE CONTROL 22.29
- Stairwell Pressurization 22.29
- Characteristics of Stairwell Pressurization 22.29
- Overpressure Relief and Feedback Control 22.30
- Stair and Shaft Vents 22.31
- Zone Smoke Control 22.31
- Design Considerations 22.32
- Volume Flow Rate 22.32
- System Pressure Loss for Stairwell Pressurization System 22.33

REFERENCES 22.38

22.1 RETURN AND EXHAUST SYSTEMS

Types of Return and Exhaust Systems

As described in Sec. 20.3, for any enclosed conditioned space in a building, if the difference of air densities is ignored, the total volume flow rate of air entering the space must be equal to the total volume flow rate of air leaving the space.

Among the three main categories of air systems described in Sec. 20.9 (Fig. 20.16):

- For most of the constant-volume systems, return air is often returned to the packaged unit through door undercuts and a collective return grille in the closet, under the attic, or at the high level in the basement. An exhaust fan is optional and is usually installed in the bathroom or toilet.
- In many dedicated outdoor air and space recirculating systems, an exhaust system with a volume flow rate less than the supply volume flow rate of the outdoor ventilation air is often employed in restrooms and service rooms. This is discussed in Chap. 28.

Return Ceiling Plenum

Among the variable-air-volume and dedicated minimum outdoor fan systems, use of the ceiling plenum as a return air plenum, as shown in Fig. 20.16*b* and *c*, is the most common return air system. In a return ceiling air plenum system, return air is often returned to the ceiling plenum through the return slots or grilles first. Usually a connected return duct connects the ceiling plenum and the fan room. Part of the return air becomes recirculating air and enters the AHU or PU, and the remaining air is exhausted outside.

For a return ceiling plenum, the following must be considered during design:

- *Fire protection*. Because the return air flows through the ceiling plenum, refer to federal and local fire codes for the requirements, materials, and equipment installed inside the ceiling plenum, as well as the fire wall and partitions that separate the ceiling plenum and adjacent fire compartments.
- *Noise control*. Fan noise that has been transmitted into the ceiling plenum is very difficult to attenuate. Do not install silencers inside the ceiling plenum. Install silencers in the fan room. Check both duct-borne noise and radiated noise; also check noise transmitted from both supply and return air sides.
- *Air leakage*. Because most of the ceiling plenums use acoustical ceiling tile and T-bar construction, the airtightness between the conditioned space and the ceiling plenum is hard to maintain. Field measurement showed that the pressure differential between the conditioned space and the ceiling plenum in a VAV system serving a small public library $\Delta p_{r-pl} = 0.004$ in. WC (1 Pa). The critical air leakage from a ceiling plenum is the air leakage to outdoors, or vice versa.

Low-Level Return Systems and Enclosed Parking Garage Ventilation

Low-level return systems have been widely used in the following applications:

- In indoor stadiums and assembly halls, where there are many spectator seats. It is better to extract the return air from the return grilles located at low-level seats to induce part of the diffused

airstreams flowing from the upper level to the low level, as shown in Fig. 18.20. The return air then flows through the return air duct, pulled by a return fan to the fan room, and is either recirculated or exhausted to outdoors.

- In clean rooms or clean spaces, in order to form a unidirectional airflow from the ceiling to the floor, so that dust will not ascend from the low level to contaminate the working area, as shown in Figs. 18.23 and 18.24.

Fully enclosed underground parking garages require mechanical ventilation systems including exhaust systems, makeup air supply systems, or both to maintain a CO concentration not exceeding 35 ppm for up to 1 hour exposure. ASHRAE sponsored a study that found a ventilation requirement of 1.5 cfm/ft^2 (7.5 L/s · m^2) or 6 air changes per hour (ach) is satisfactory.

ASHRAE/IESNA Standard 90.1–1999 mandatorily specifies that garage ventilation air systems with a total volume flow rate greater than 30,000 cfm (14,200 L/s) shall be permited to (*a*) have at least an automatic control to stage fans or modulating fan volume to maintain CO concentration below levels specified in ASHRAE Standard 62, 35 ppm up to 1 hour exposure, or (*b*) be capable of shutting off fans or reducing fan volume when the garage is not in use.

Exhaust Hoods

ASHRAE/IESNA Standard 90.1–1999 specifies that individual kitchen exhaust hoods larger than 5000 cfm (2360 L/s) shall be provided with a makeup air system with a capacity equal to at least 50 percent of exhaust volume flow rate that is (*a*) unheated or heated to no higher than 60°F (10.6°C) and (*b*) uncooled or cooled without the use of mechanical cooling (refrigeration). Exceptions include

- Exhaust hoods used to exhaust ventilation air which otherwise would exfiltrate or be exhausted by other air systems
- Certified grease extractor hoods that require a face velocity no greater than 60 fpm (0.3 m/s)

Standard 90.1–1999 also specifies that buildings with fume hood exhaust systems having a total exhaust volume flow rate greater than 15,000 cfm (7080 L/s) shall include at least one of the following features:

- Variable-air-volume hood exhaust and room supply systems capable of reducing exhaust and makeup volume flow to 50 percent or less of design flows.
- Direct makeup air supply equal to no less than 75 percent of the exhaust rate, heated to no warmer than 2°F (1.1°C) below room set point, cooled to no cooler than 3°F (1.7°C) above room set point, no humidification allowed, and no simultaneous heating and cooling for dehumidification.
- Heat recovery system for makeup air from fume hood exhaust with same exceptions as in Sec. 12.5. Refer to Standard for details.

Standard 90.1–1999 also mandatorily specifies that all outdoor air supply and exhaust hoods, vents, and ventilators shall be installed with motorized dampers that will automatically shut when the spaces served are not in use. Gravity (nonmotorized) dampers are acceptable in buildings less than three stories in height above grade and for buildings of any height located in climates with less than 2700 HDD65. Exceptions are for ventilation systems serving unconditioned spaces as well as during preoccupancy building warmup, colddown, and setback, except when ventilation reduces energy costs (such as, night purge). Both outdoor air supply and exhaust air dampers shall have a maxium leakage rate of 3 cfm/ft^2 at 1.0 in. WC (15 L/s · m^2 at 250 Pa).

22.2 FAN COMBINATIONS AND OPERATING MODES

Fan Combinations

In commercial buildings, constant-volume systems with optional exhaust are widely used in small air systems, and the VAV systems with return ceiling plenum are the most widely used in medium-size and large air systems. As discussed in Sec. 20.9, among these air systems, the following fan combinations are often used:

- Supply and exhaust fan combination
- Supply and relief fan combination
- Supply and return (including unhoused plug/plenum) fan combination

Supply, exhaust, relief, and return fans can either be a single fan or multiple fans. Recently, some manufacturers use unhoused plug/plenum fan as the return fan to form a supply and return fan combination. As discussed in Sec. 15.2 an unhoused plug/plenum fan is simpler in construction and has the flexibility of location of the discharge outlet. However, it requires more power than a centrifugal fan of the same capacity and fan total pressure.

Air systems of various fan combinations are installed to meet the following requirements:

- To provide required outdoor ventilation air for occupants and conditioned space according to ASHRAE Standard 62–1999
- To supply conditioned air to the space to offset space load
- To recirculate space air for energy conservation
- To operate in an air economizer cycle for energy saving
- To exhaust objectionable contaminated air from the space
- To maintain a proper indoor environment

Operating Modes

Constant-volume systems with optional exhaust and VAV systems with return ceiling plenum may be operated in the following modes:

- Cooling mode is one of the basic operating modes in air systems. In cooling mode operation, outdoor air or recirculating air, or a mixture of the two is cooled or is cooled and dehumidified in the water cooling coil or DX coil, and the conditioned air is supplied to the conditioned space to offset the cooling load. Cooling modes can again be subdivided into summer cooling mode and winter cooling mode. Many interior zones that have heavy internal cooling loads still need cold air supply in the winter cooling mode.
- Heating mode is another basic operating mode in air systems. In heating mode operation, outdoor air, recirculating air, or their mixture is heated in the direct-fired furnace, electric heater or water heating coil. Warm air is supplied to the conditioned space to offset the heating load. In heating mode, humidification is optional.
- In part-load operation, the space cooling or heating load is less than the summer or winter design load. This may be due to the variation of the space load, or change of the outdoor climate, or both. Air systems operate at part load most of the time. During part-load operation, the supply volume flow rate as well as the cooling and heating capacities of the air system reduces accordingly. It is assumed that the supply volume flow rate may be reduced to 50 percent, and even dropped to 40 percent of design flows.

- In minimum outdoor air recirculating mode or simply recirculating mode, only minimum outdoor air contained in the mixture of outdoor and recirculating air is supplied to the conditioned space for the occupant and for dilution of contaminated air. Minimum outdoor air circulation mode can be a subsidiary operating mode in a cooling or heating mode operation.
- In an air economizer mode, only the outdoor air or a mixture of outdoor and recirculating air is supplied to the conditioned space to replace part of or all the mechanical cooling (cooling from refrigeration). Air economizer mode is an outdoor air free-cooling mode and is often used as the first stage of cooling to save energy.
- In purge mode, 100 percent outdoor air is extracted during the nighttime or in the morn-ing before the space is occupied, to cool down the space with outdoor air or to flush the conditioned space with cleaner outdoor air while at the same time squeezing out the indoor contaminated air.
- In warm-up and cool-down mode, the space air temperature is warmed up or cooled down to predetermined limits after the air system is started in the morning while the space is not occupied. During warm-up or cool-down mode, the recirculating damper is fully open and the outdoor and exhaust dampers are closed, except when the warm-up or cool-down mode is combined with the purge mode.

  Warm-up and cool-down mode operation should not take place prior to the purge mode.

- In nighttime setback mode, the space temperature is set back when the conditioned space is unoccupied in winter, typically from 70 to 55°F (21.1 to 12.8°C), to prevent freezing indoors and to reduce the warm-up temperature differential the next morning before the space is occupied.
- Occupied mode is for the operating hours when the conditioned space is occupied by the occupants, and the unoccupied mode is for the operating hours when the conditioned space is not occupied by the occupants.

22.3 SYSTEM PRESSURE DIAGRAM

The system pressure diagram of a VAV system during minimum outdoor air recirculating mode is shown in Fig. 22.1. The system characteristics of a VAV system not only affect its fan energy use, but also are closely related to the variation in its supply volume flow rate and operating characteristics. System pressure is usually expressed in total pressure, in in. WG (Pa), in the form of a system pressure diagram. In conditioned space and in places where velocity pressure is so small, it is often ignored. The total pressure is then equal to static pressure, that is, $p_t = p_s$.

System Pressure and Duct Static Pressure Control

In a VAV system with duct static pressure control, the total pressure loss of the VAV system can be divided into two parts: variable part Δp_{var} and fixed part Δp_{fix}, both in in. WC (Pa). The total pressure loss between the centerline of the recirculating damper in the PU and the static pressure sensor of the duct static pressure control is expressed as $\Delta p_{sf,var}$, in in. WC (Pa). As the supply volume flow rate in a VAV system varies, Δp_{var} varies accordingly. The fixed part Δp_{fix} is independent of the variation in the supply flow rate and remains constant as the volume damper in the VAV box increases its pressure loss at reduced volume flow rate (Figs. 20.13 and 22.1).

For a VAV system installed with a supply fan and a return fan connected in series, as shown in Figs. 22.1 and 22.4, its system pressure loss Δp_{sys}, in in. WC (Pa), is also divided into variable part

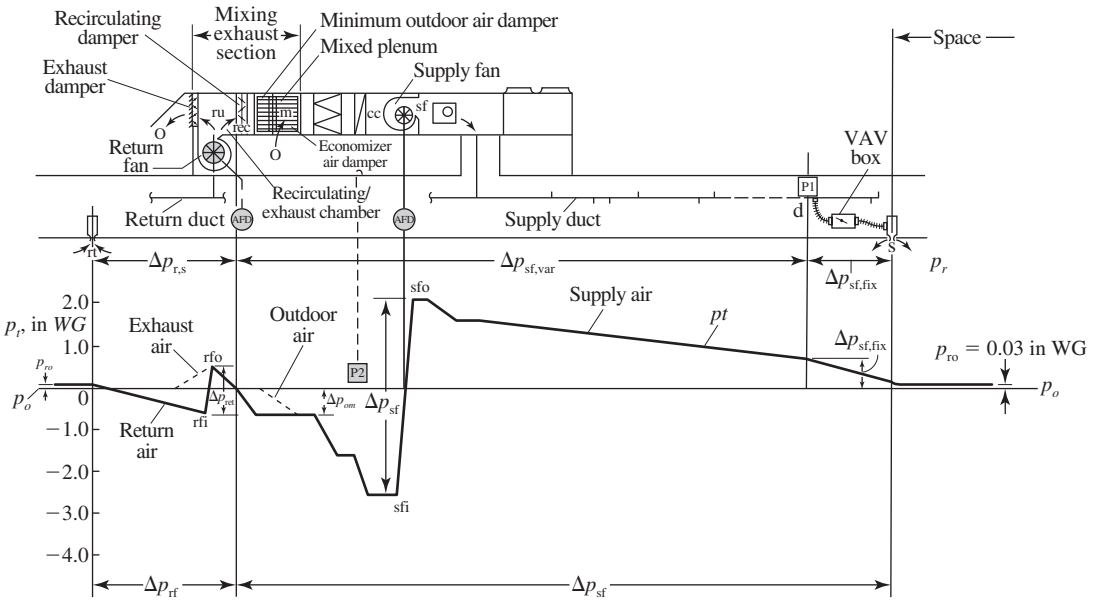


FIGURE 22.1 System pressure diagram for a supply-return fan combination air system (connected in series) (rfi = return fan inlet; rfo = return fan outlet; sfi = supply fan inlet; sfo = supply fan outlet; Δp_{rf} = return fan total pressure; Δpsf = supply fan total pressure; AFD = adjustable-frequency, variable-speed drive).

Δp_{var} and fixed part Δp_{fix}, in in. WC (Pa), and it can be calculated as

$$\begin{aligned}\Delta p_{sys} &= \Delta p_{var} + \Delta p_{fix} = R_{var}\dot{V}_{var}^2 + \Delta p_{fix} \\ &= (\Delta p_{r,s} + \Delta p_{PU} + \Delta p_{sd,var}) + \Delta p_{fix} \\ &= (\Delta p_{r,s} + \Delta p_{sf,var}) + \Delta p_{fix}\end{aligned} \tag{22.1}$$

where Δp_{rs} = pressure loss of return air system, in. WC (Pa)
Δp_{PU} = pressure drop across PU, in. WC (Pa)
$\Delta p_{sd,\,var}$ = pressure loss of supply duct system before duct static pressure sensor, in. WC (Pa)
$\Delta p_{sf,\,var}$ = pressure loss of variable part of supply fan, in. WC (Pa)
R_{var} = flow resistance representing variable part of VAV system, in. WC/(cfm)^2 (Pa · s^2/m^6)
$\dot{V}_{var}$ = supply volume flow rate of VAV system, cfm (m^3/s)

In a VAV system with a supply and return fan combination connected in series, the supply volume flow rate flowing through the supply fan is different from the return volume flow rate flowing through the return air fan. This makes the pressure-volume operating characterirstics more complicated.

If the pressure sensor is installed closer to the supply fan, the fan power savings at reduced volume flow decreases because of the increase in Δp_{fix}. The set point of the static pressure sensor may vary from 0.7 to 1.5 in. WC (125 to 375 Pa).

In a VAV system serving a single floor in a multistory building, if the maximum design air velocity in the main duct v_{max} is 3000 fpm (15 m/s), and if the length of main duct is 150 ft (45.7 m),

then the design total pressure loss of this supply duct system $\Delta p_{\text{sd, var}}$ may vary from 0.75 to 1.5 in. WC (187 to 375 Pa), typically 1 in. WC (250 Pa), depending on the duct fittings in the main duct.

Fan Characteristics

A VAV system with a supply and return fan or supply and relief fan combination is often connected in series in a fan-duct system. The reasons to use a supply fan and a return fan instead of a single supply fan are as follows:

- To meet the requirement of maintaining a slightly positive pressure in the conditioned space as well as at the inlet of the AHU or PU point *ru*, and a negative pressure in the mixing box (mixed plenum) to extract outdoor air
- To exhaust 100 percent free-cooled outdoor air during an air economizer cycle
- To set up different volume flow rates of supply air and return air in order to provide a required amount of outdoor ventilation air as well as to maintain a desirable space pressure

During the design of a series-connected supply and return fan combination system, as shown in Fig. 22.1, it is essential that the following parameters be carefully calculated and analyzed:

- Clearly divide the total pressure loss that should be undertaken by the supply fan, or by the return fan during minimum outdoor air recirculating mode at summer design conditions. The return system pressure loss undertaken by the return fan start from the inlet of return slot *rt* includes the return slot, airflow inside the ceiling plenum which is often ignored because of its lower air velocity, the return grille if any, the return duct and duct fittings, up to part of the recirculating damper, as shown in Figs. 22.1 and 22.4. For the PU with a configuration shown in Fig. 22.1, in minimum outdoor air recirculating mode, a 90° turn plus one-half of the total pressure loss of the recirculating damper at full opening design flow is always greater than a 90° turn plus the total pressure loss of the exhaust damper and louvers at minimum outdoor air recirculating mode.

In a VAV system, the fan total pressure of the return fan $\Delta p_{t,\text{rf}}$ must be carefully calculated, neither overestimated nor underestimated.

- Based on the principle of conservation of mass, from Eq. (20.8), the relationship between the mass flow rates of supply, return, and exhaust flow rate can be expressed as

$$\dot{m}_s = \dot{m}_{\text{rt}} + \dot{m}_{\text{ex}} + \dot{m}_{\text{exf}} \tag{22.2}$$

where $\dot{m}_s$, $\dot{m}_{\text{rt}}$, $\dot{m}_{\text{ex}}$ = mass flow rate of supply air, return air, and exhaust air, lb/min (kg/min)
$\dot{m}_{\text{exf}}$ = mass flow rate of exfiltrated air, lb/min (kg/s)

If the difference in air densities is ignored and a positive pressure is maintained in the conditioned space, from Eq. (20.9), the relationship between the supply, return, and exhaust volume flow rates can be expressed as

$$\dot{V}_s = \dot{V}_{\text{rt}} + \dot{V}_{\text{ex}} + \dot{V}_{\text{exf}} \tag{22.3}$$

where $\dot{V}_s$, $\dot{V}_{\text{rt}}$, $\dot{V}_{\text{ex}}$ = volume flow rates of supply air, return air, and exhaust air, cfm (L/s)
$\dot{V}_{\text{exf}}$ = volume flow rate of exfiltrated air, cfm (L/s)

If the supply air is at 55°F (12.8°C) dry-bulb and 54°F (12.2°C) wet-bulb temperature and the return air is at 78°F (25.6°C) dry-bulb and 65°F (18.3°C) wet-bulb temperature, when the mass flow rates of the supply and return air are the same, the volume flow rate of return air is about 4 percent greater than that of the supply air. During cold air distribution, if the supply air is at 44°F (6.7°C) dry-bulb and 43°F (6.1°C) wet-bulb temperature, and the return air remains the same, the volume

flow rate of the return air is 7 percent greater. For accurate analysis, these differences should be taken into consideration.

Mixing-Exhaust Section and Conditioned Space

In an air system, there are two places where the air is connected to the outside atmosphere: (1) the mixing box–exhaust chamber (simply the mixing-exhaust section), where it is connected with the outside atmosphere through the outdoor air intake and the exhaust passage, and (2) the conditioned space, where it connects with the outside atmosphere through the openings in the building shell, as shown in Fig. 22.1.

In a mixing box of a VAV system, the variation of the negative pressure affects the amount of outdoor air intake and, therefore, the indoor air quality of the conditioned space. Within the conditioned space, the variation of the supply air, return air, exhaust air, infiltration, and exfiltration makes the space airflow balance and associated space pressure complicated.

22.4 SUPPLY FAN AND EXHAUST FAN COMBINATION

System Characteristics

Air systems equipped with a single supply fan and an exhaust fan are often used in buildings in which the total pressure loss of the return duct system is low. A certain volume flow of air is usually exhausted from rooms such as restrooms. A barometric relief damper is often installed in the conditioned space or in the return plenum to avoid excessively high space pressure. When the space positive pressure on the damper is greater than the weight of the damper, the damper opens and the space pressure is relieved and is thus maintained below a predetermined value. Such an air system may be equipped with a water economizer.

To analyze the pressure characteristics of the airstream at two cross-sectional planes along the airflow in an air system, the total pressure is determined from Eq. (17.12): $p_{t1} = p_{t2} + \Delta p_f$.

To determine whether air is flowing from one enclosed space to another, or from an enclosed space to the outdoors, the static pressure difference between these two places $\Delta p_s = p_{s1} - p_{s2}$ should be calculated. However, the static pressure inside an enclosure of uniform pressure, or the mean static pressure on a cross-sectional plane along airflow, must be determined from Eq. (17.11): $p_s = p_t - p_v$. Here p_v represents velocity pressure.

As the atmospheric pressure is assumed $p_o = 0$ for the convenience of analysis, static pressure at the mixing box p_m must be negative in order to extract outdoor air. The total pressure difference between the outdoor air and the air in the mixed plenum (mixing box) $\Delta p_{om} = p_o - p_m$ consists mainly of the pressure loss of the outdoor dampers and louver. If a space is maintained at a positive pressure, the volume flow rate of exfiltration from the space $\dot{V}_{\text{exf}}$, in cfm (L/s), can be calculated from Eq. (20.11). If space is maintained at a negative pressure, the volume flow rate of infiltration into the space $\dot{V}_{\text{inf}}$, in cfm (L/s), can be similarly calculated. From Eq. (20.11), the pressure loss across the building shell Δp_{ro}, in in. WC (Pa), can be calculated as

$$\Delta p_{ro} = p_r - p_o = \left(\frac{1}{4005\, A_{e,l}}\right)^2 \dot{V}_{\text{exf}}^2 = R_{\text{exf}} \dot{V}_{\text{exf}}^2 \tag{22.4}$$

The flow resistance against exfiltrated air across building shell R_{exf}, in in. WC/(cfm)^2 (Pa·s^2/m^6), is

$$R_{\text{exf}} = \left(\frac{1}{4005\, A_{e,l}}\right)^2$$

Because the volume flow rates of the supply air and recirculating air are different, the flow resistance of the supply system and return system as well as the flow resistance of the fixed part and variable part of the air system pressure loss should be calculated separately.

Operating Characteristics

Consider a VAV rooftop packaged system with a supply fan and an exhaust fan serving a typical floor in a commercial building, as shown in Fig. 22.2. This air system has the following operating characteristics at design conditions:

Supply volume flow rate	20,000 cfm (9438 L/s)
Total pressure losses:	
Across recirculating damper at design flow	0.4 in. WC (100 Pa)
Filters and coils	2.5 in. WC (625 Pa)
Supply main duct, $\Delta p_{d,\,var}$	0.85 in. WC (212 Pa)
Recirculating system	0.15 in. WC (37 Pa)
VAV box, branch duct, and diffuser	0.75 in. WC (187 Pa)
Effective leakage area on building shell	0.5 ft^2 (0.46 m^2)
Volume flow of exhaust fan	3000 cfm (1416 L/s)
Minimum outdoor ventilation air required	3350 cfm (1581 L/s)
Space pressure	+ 0.03 in. WG (7.5 Pag)

For simplicity, the difference in air densities between supply, return, and exhaust air is ignored.

In minimum outdoor air recirculating mode, the pressure losses of the outdoor louver can be neglected. A barometric relief damper is mounted in the conditioned space and is opened when the space positive pressure is greater than 0.2 in. WG (50 Pag). The supply fan has the same $\Delta p_t - \dot{V}$ characteristics as in Example 20.3.

The following control systems are installed to maintain the required operating parameters:

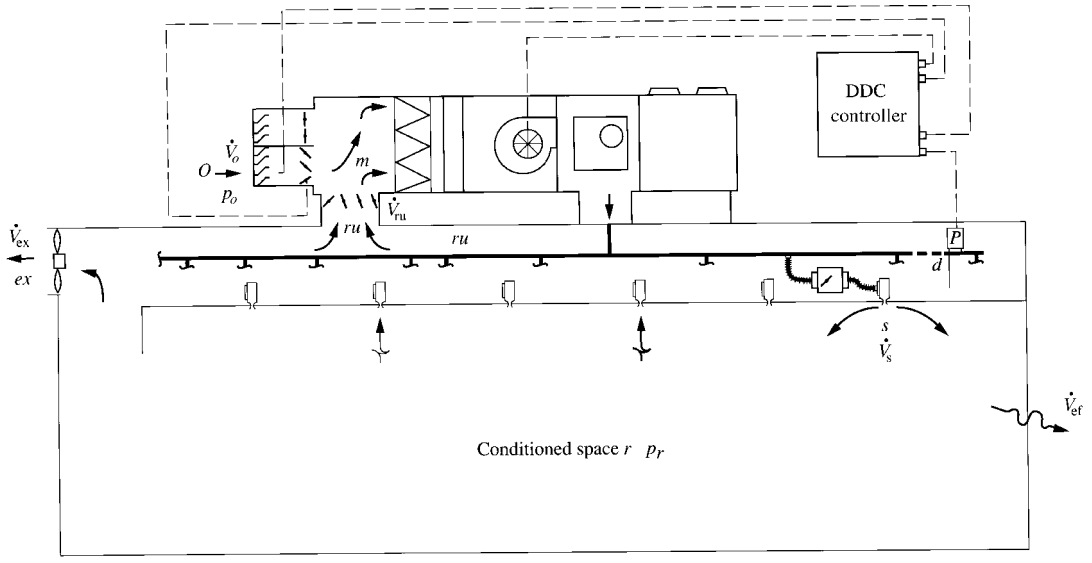
- Zone temperature control
- Discharge temperature control
- Duct static pressure control
- Minimum outdoor ventilation air control
- High pressure limit control

All these control systems are described in Chap. 23.

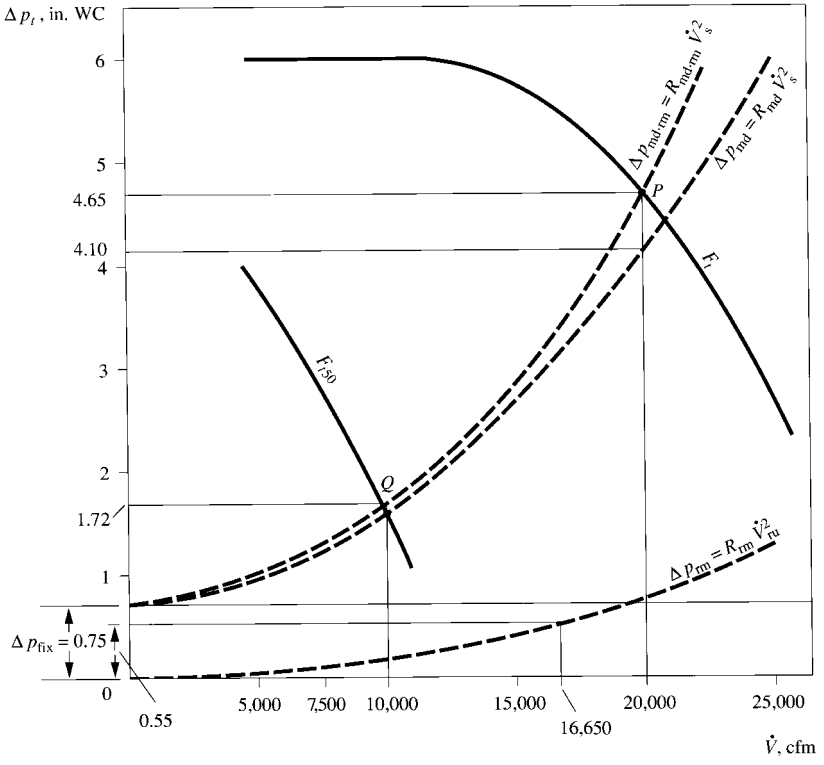
In summer, this air system can be operated in cooling mode, recirculating mode, air economizer mode, or warm-up or cool-down mode.

Recirculation Mode and Design Volume Flow

Minimum outdoor air is extracted through the outdoor intake and is mixed with the recirculating air in the mixing box in a rooftop packaged unit. The mixture is conditioned in the packaged unit. After conditioning, the supply air flows through the supply fan and the supply duct and is discharged to the conditioned space. In the conditioned space, a certain volume flow rate is exhausted through an exhaust fan, and another small portion is exfiltrated through the gaps and openings on the building shell if the space is maintained at a positive pressure between + 0.005 and + 0.03 in. WG (1.25 and 7.5 Pag). The major portion of the space air is returned to the rooftop unit, where it is mixed with the outdoor air for recirculation.



(a)



(b)

FIGURE 22.2 Air system of supply and exhaust fan combination: (*a*) schematic diagram; (*b*) system characteristics on $\Delta p_t - \dot{V}$ diagram.

When the air system of a single supply fan and exhaust fan combination is operated in recirculating mode with design volume flow, the inlet vanes at the supply fan inlet are fully opened and the outdoor air damper is at minimum opening position. The minimum outdoor ventilation air required $\dot{V}_o$ is about 3350 cfm (1581 L/s). The fixed part of the pressure loss of such an air system is $\Delta p_{\text{fix}} = 0.75$ in. WC (187 Pa). The variable part of pressure loss of the air system consists of the following two sections:

- Mixed plenum point m to the static pressure sensor, point d, at a supply volume flow rate of 20,000 cfm (9438 L/s):

$$\Delta p_{md} = 2.5 + 0.85 = 3.35 \text{ in. WC (835 Pa)}$$

Its corresponding flow resistance can be calculated as

$$R_{md} = \frac{\Delta p_{mr}}{\dot{V}_s^2} = \frac{3.35}{(20{,}000)^2} = 8.38 \times 10^{-9} \text{ in. WC/(cfm)}^2 \ (4.42 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

- Recirculating system from point r to m:

$$\Delta p_{rm} = R_{rm}\dot{V}_{ru}^2 = 0.15 + 0.4 = 0.55 \text{ in. WC (137 Pa)}$$

The variable part of pressure loss of the air system is

$$\begin{aligned} \Delta p_{md,rm} &= \Delta p_{md} + \Delta p_{rm} \\ &= R_{md}\dot{V}_s^2 + R_{rm}K_{rm}^2\dot{V}_s^2 \\ &= R_{md,\,rm}\dot{V}_s^2 \end{aligned} \tag{22.5}$$

Because of a positive space pressure of + 0.03 in. WG (+ 7.5 Pag) and an effective leakage area $A_{e,\,l}$ of 0.5 ft^2, the volume flow rate of exfiltration from the conditioned space $\dot{V}_{\text{exf}}$ is calculated as

$$\dot{V}_{\text{exf}} = 4005\, A_{e,\,l}\sqrt{\Delta p_{ro}} = 4005 \times 0.5\sqrt{0.03} \approx 350 \text{ cfm (165 L/s)}$$

And the flow resistance of the opening in the building shell is

$$R_{\text{exf}} = \frac{\Delta p_{ro}}{\dot{V}_{\text{exf}}^2} = \frac{0.03}{(350)^2} = 2.45 \times 10^{-7} \text{ in. WC/(cfm)}^2 \ (0.129 \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

The recirculating volume flow rate $\dot{V}_{ru}$ through the recirculating system is then equal to

$$\dot{V}_{ru} = \dot{V}_s - (\dot{V}_{\text{exf}} + \dot{V}_{\text{ex}}) = 20{,}000 - (350 + 3000) = 16{,}650 \text{ cfm (7857 L/s)}$$

The flow resistance of the recirculating system based on pressure loss between points r and m is

$$R_{rm} = \frac{\Delta p_{rm}}{\dot{V}_{rm}^2} = \frac{0.55}{(16{,}650)^2} = 1.98 \times 10^{-9} \text{ in. WC/(cfm)}^2 \ (1.04 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

And the flow resistance of the variable part of the pressure loss of the air system is

$$\begin{aligned} R_{md,rm} &= R_{md} + R_{rm}K_{rm}^2 \\ &= 8.38 \times 10^{-9} + 1.98 \times 10^{-9}\left(\frac{16{,}650}{20{,}000}\right)^2 \\ &= 9.75 \times 10^{-9} \text{ in. WC/(cfm)}^2 \ (5.14 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6) \end{aligned}$$

The supply fan total pressure at design volume flow rate is

$$\Delta p_{\text{sf}} = 0.75 + 2.5 + 0.85 + 0.15 + 0.4 = 4.65 \text{ in. WC (1156 Pa)}$$

The system operating point of this supply and exhaust fan combination P is plotted in Fig. 22.2*b*. The pressure-volume characteristics for the key points in this system are shown below:

	Point *o*	Point *m*	Point *r*	Point *ru*
p, in. WG	0	− 0.52	+ 0.03	− 0.12
$\dot{V}$, cfm	3350	20,000	20,000	16,650

Recirculation Mode, 50 Percent Design Flow Rate

When the space sensible cooling load is reduced at part-load operation, the supply volume flow rate may be reduced to 50 percent of the design volume flow rate, that is, 0.5 × 20,000 = 10,000 cfm (4719 L/s).

As the dampers in the VAV boxes close to smaller openings than at design volume flow, the static pressure in the supply main duct rises. The sensor senses the rise, and the DDC unit instructs the inlet vanes to close to a smaller opening, until the duct static pressure is maintained approximately at its set point at the location where the pressure sensor is mounted. The static pressure is still equal to the pressure loss of the VAV box, flexible ducts, diffuser in the branch takeoff, and last section of the main duct between the pressure sensor and end of the main duct plus the space static pressure: 0.75 + 0.03 = 0.78 in. WC (194 Pa). The fan performance curve is now a new fan curve F_{t50}, as shown in Fig. 22.2*b*.

The variable-part pressure loss between the mixing box and point d is now decreased to

$$\Delta p_{md50} = R_{md}\dot{V}_{s50}^2 = 8.38 \times 10^{-9}(10{,}000)^2 = 0.84 \text{ in. WC (209 Pa)}$$

Assume that the space pressure at 50 percent design flow is − 0.03 in. WG. The volume flow rate of infiltrated air into the space is

$$\dot{V}_{\text{inf}} = 4005\, A_{e,l}\sqrt{\Delta p} = 4005 \times 0.5\sqrt{0.03} = 347 \text{ cfm (164 L/s)}$$

because the volume flow rate of an exhaust fan is closely related to its pressure. According to the catalog of a fan manufacturer, within the operating range of a 24-in. (600-mm) diameter exhaust fan, for an increase of 0.01 in. WC (2.5 Pa) in fan total pressure, there is a corresponding drop in supply volume flow rate of 135 cfm (64 L/s). Compared with the design conditions, there is a need for an increase of + 0.03 − (− 0.03) = 0.06 in. WC (15 Pa) of fan total pressure of the exhaust fan, i.e., a decrease of 0.06/0.01 × 135 = 810 cfm of volume flow rate. The volume flow rate of recirculating air is then

$$\dot{V}_{ru50} = \dot{V}_{s50} - \dot{V}_{\text{ex}} + \dot{V}_{\text{inf50}} = 10{,}000 - (3000 - 810) + 347 = 8157 \text{ cfm (3849 L/s)}$$

The pressure loss between the space (point r) and the mixed plenum (point m) becomes

$$\Delta p_{rm} = R_{rm}\dot{V}_{rm50}^2 = 1.98 \times 10^{-9}(8157)^2 = 0.132 \text{ in. WC (33 Pa)}$$

Therefore, the total pressure in the mixing box is

$$p_m = -0.03 - 0.132 = -0.162 \text{ in. WG } (-40 \text{ Pag})$$

The flow resistance of the outdoor passage is

$$R_{om} = \frac{\Delta p_{om}}{\dot{V}_o^2} = \left(\frac{0.52}{3350}\right)^2 = 4.63 \times 10^{-8} \text{ in. WC/(cfm)}^2 \text{ (0.0244 Pa} \cdot \text{s}^2/\text{m}^6)$$

Because of a pressure difference $\Delta p_{om} = 0.162$ in. WC, outdoor intake is therefore

$$\dot{V}_o = \sqrt{\frac{\Delta p_{om}}{R_{om}}} = \sqrt{\frac{0.162}{4.63 \times 10^{-8}}} = 1870 \text{ cfm (883 L/s)}$$

If the difference in air density is ignored, the volume flow balance between the incoming infiltrated air and outward exhaust air in this air system becomes

$$\dot{V}_o + \dot{V}_{\text{inf}} - \dot{V}_{\text{ex}} = 1870 + 347 - (3000 - 810) = +27 \text{ cfm } (13 \text{ L/s})$$

The volume flow of incoming air and exhaust air entering and leaving the system is nearly balanced.

From the above analysis, the system operating point at 50 percent design volume flow rate, point *Q*, has a supply volume flow of 10,000 cfm (4719 L/s) and a fan total pressure

$$\Delta p_{t50} = 0.75 + 0.84 + 0.13 = 1.72 \text{ in. WC } (428 \text{ Pa})$$

During minimum outdoor air recirculating mode at 50 percent design volume flow, the volume flow rate of outdoor air extracted through the outdoor intake passage decreases to 1870 cfm (883 L/s). Space is maintained at a negative pressure of − 0.03 in. WG (− 7.5 Pag). An infiltration rate of 347 cfm (164 L/s) enters the space through openings and cracks in the building shell.

The $p_t - \dot{V}$ characteristics at key points during 50 percent design volume flow are as follows:

	Point *o*	Point *m*	Point *r*	Point *ru*
p, in. WG	0	− 0.162	− 0.03	− 0.065
$\dot{V}$, cfm	1870	10,000	10,000	8157

Air Economizer Cycle

When an air economizer cycle is operated at a design flow rate of 100 percent of outdoor air in this air system with a single supply fan and an exhaust fan combination, the recirculating damper is closed, and both the minimum outdoor damper and the other, large outdoor damper for the air economizer cycle are fully opened. Outdoor air is extracted to the mixed plenum, through the filters and coils, and is supplied to the conditioned space. Because only a small portion of supply air is exhausted and exfiltrated, the space positive pressure increases to about 0.2 in. WG (50 Pag). The barometric relief damper then opens. Most of the supply air is discharged through the relief damper outdoors.

When the supply and exhaust fan combination is operated at 50 percent design volume flow in the air economizer cycle, space pressure is still limited to 0.2 in. WG (50 Pa), and most of the supply air is discharged through the relief damper outdoors, as in the design volume flow.

Warm-Up and Cool-Down Mode

During the warm-up and cool-down mode, the outdoor damper is closed and the recirculating damper is fully open, while the exhaust fan is turned off. Recirculated air from the space is conditioned in the packaged unit and is then supplied to the space. Because of the negative pressure in the mixing box, outdoor air is leaked into the box through the closed outdoor dampers and is exfiltrated from the space because of the higher positive space pressure.

Pressure Variation at the Mixing Box

During the recirculating mode, when the supply volume flow rate is reduced from design flow to 50 percent of design flow, the decrease in the recirculating volume flow rate causes a corresponding drop in the pressure loss of the recirculation system. Therefore, the pressure at the mixed plenum, p_m increases from − 0.52 to − 0.16 in. WG (− 129 to − 40 Pag). The greater the pressure loss of the return system at the design flow rate, the higher the fluctuation of Δp_{om}. Because of the reduced volume flow rate at a higher fan total pressure of the exhaust fan at 50 percent design flow, in order

to balance the airflow, the space pressure is changed from + 0.03 to − 0.03 in. WG (+ 7.5 to − 7.5 Pag), and the exfiltration changes to infiltration. The pressure loss at the outdoor air intake passage (excluding the damper), like the intake louver and duct friction loss Δp_{lou}, should be taken into account only in the outdoor air economizer cycle. During minimum outdoor air recirculating mode, Δp_{lou} should be negligible.

Pressure fluctuation Δp_{om} at design flow and at 50 percent design flow causes insufficient outdoor air intake. The effect of this pressure fluctuation can be reduced by using minimum outdoor ventilation air control, as discussed in Chap. 23.

22.5 SUPPLY FAN AND RELIEF FAN COMBINATION

Consider an air system with a supply fan, relief fan, and exhaust fan combination, as shown in Fig. 22.3*a,* with the same operating characteristics as mentioned in Sec. 22.4. A relief fan is a fan that is installed in the relief flow passage (exhaust flow passage), adjacent to the recirculating flow passage leading to the mixing box, to relieve the undesirably high positive space pressure. A relief fan is different from an exhaust fan, which is often used to exhaust fumes and contaminated or toxic gases and to maintain a negative pressure in the space or in a localized enclosure. An axial fan is often used as a relief fan because of its large volume flow and smaller fan total pressure. The control systems used in such a combination are similar to those of a supply fan and exhaust fan combination, except that a relief fan control is added. The relief fan has the following pressure-volume characteristics:

$\dot{V}$, cfm	10,000	15,000	16,640	17,250
Δp_t, in. WC	4.0	2.10	1.08	0.43

Recirculation Mode

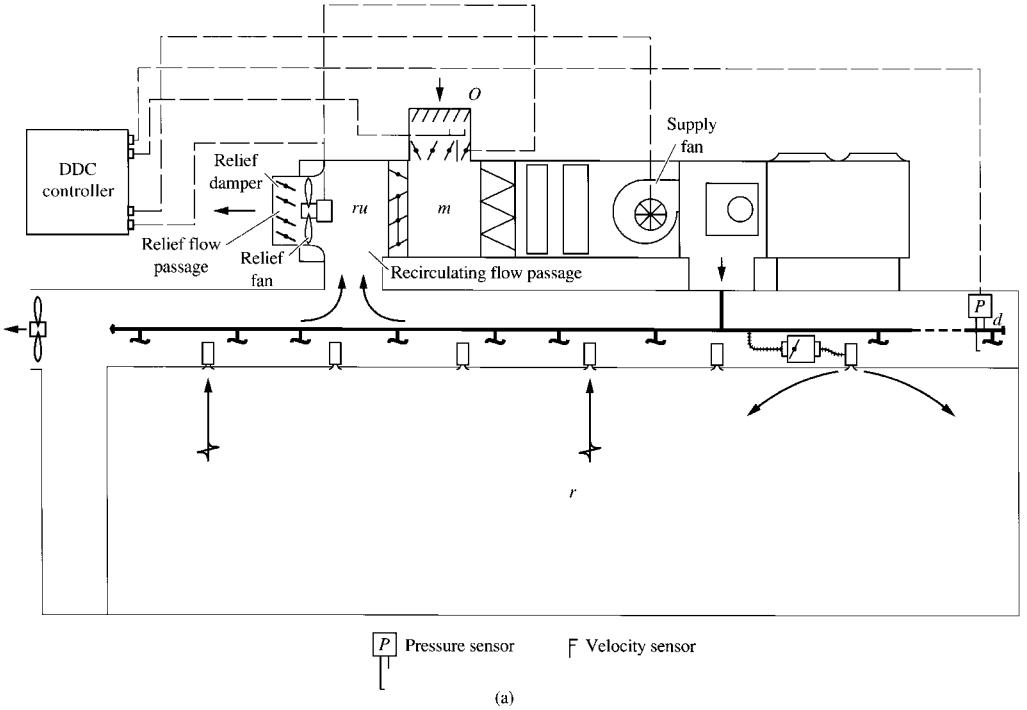
During minimum outdoor air recirculating mode operation, the relief fan is not operating and the relief damper is closed. Outdoor air drawn into the air system is balanced by the exhaust air and the exfiltration through the openings in the building shell. In this case, the system becomes a supply fan and exhaust fan combination, and its operating characteristics are the same as those described and analyzed in Sec. 22.4.

Air Economizer Mode and Design Volume Flow Rate

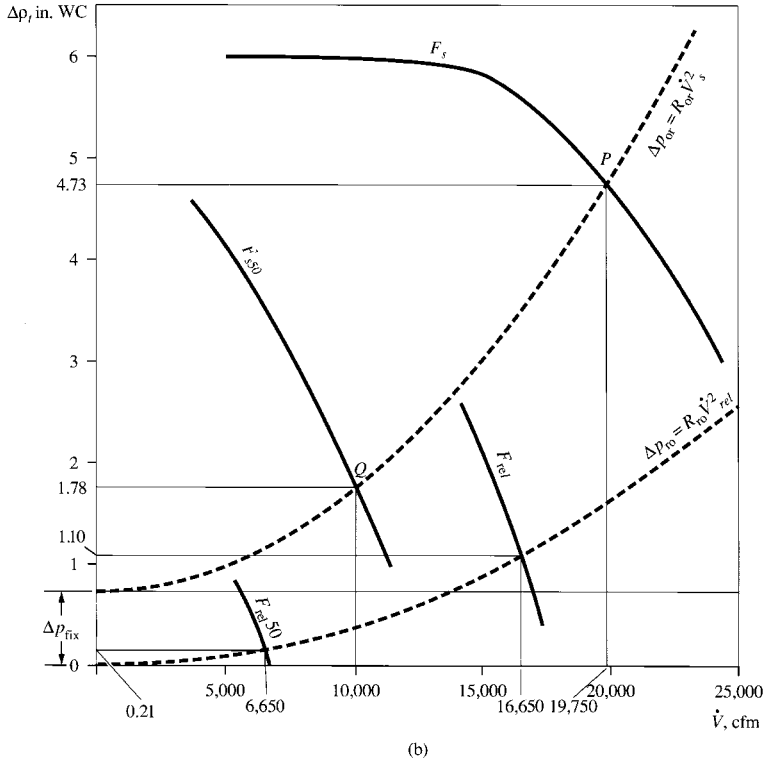
In a VAV system, there are often two outdoor air dampers: a smaller minimum outdoor air damper and a larger economizer damper. Both dampers have a total pressure loss of 0.4 in. WC (100 Pa) at 100 percent outdoor air intake. During air economizer mode with 100 percent outdoor air at the design volume flow rate, the outdoor dampers are fully opened and the recirculating damper is closed. The relief fan relieves the space air pressure and maintains it within predetermined limits. Outdoor air is extracted through the intake louver and both outdoor dampers, and flows through the filter and coils. The conditioned air is then supplied to the space. A small part of the space air is exhausted, and another part is exfiltrated. The majority of space air is relieved by the relief fan through the ceiling plenum and relief passage.

At design volume flow, the duct friction loss of the outdoor intake passage is negligible, and the pressure loss of the intake louver is 0.2 in. WC (50 Pa). The space pressure depends mainly on the airflow balance in the space. The variable part of the pressure loss of the supply air system is then given as

$$\Delta p_{var} = 0.2 + 0.4 + 2.5 + 0.85 + 0.03 = 3.98 \text{ in. WC (990 Pa)}$$



(a)



(b)

FIGURE 22.3 Supply fan and relief fan combination: (*a*) schematic diagram; (*b*) system characteristics on $\Delta p_t - \dot{V}$ diagram.

The fixed part of pressure loss of the air system is still 0.75 in. WC (186 Pa). The fan total pressure of the supply fan is then equal to

$$0.75 + 3.98 = 4.73 \text{ in. WC (1176 Pa)}$$

The flow resistance of the variable part of the supply air system can be calculated as

$$R_{or} = \frac{\Delta p_{or}}{\dot{V}_s^2} = \frac{3.98}{(20{,}000)^2} = 9.95 \times 10^{-9} \text{ in WC/(cfm)}^2 \ (5.24 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

The fan total pressure of the relief fan is used to overcome the pressure loss of the return system, relief damper, and louver in the relief flow passage. Usually, the duct friction loss of the relief passage is ignored. At design volume flow rate, for a space pressure of + 0.03 in. WG (+ 7.5 Pag), an exfiltration of about 350 cfm (165 L/s), and an exhaust volume flow rate of 3000 cfm (1416 L/s), the volume flow rate of relief fan is 20,000 − 3000 − 350 = 16,650 cfm (7857 L/s).

As in the supply fan and exhaust fan combination, the pressure loss between points *r* and *ru* is

$$\Delta p_{r,ru} = R_{r,\,ru} \dot{V}_{ru}^2 = 0.15 \text{ in. WC (37 Pa)}$$

and flow resistance is

$$R_{r,ru} = \frac{\Delta p_{r,ru}}{\dot{V}_{ru}^2} = 5.41 \times 10^{-10} \text{ in. WC/(cfm)}^2 \ (2.85 \times 10^{-4} \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

If the pressure loss of the relief louver at a volume flow of 16,650 cfm (7857 L/s) is 0.2 in. WC (50 Pa), the pressure loss of the relief damper is 0.4 in. WC (100 Pa), and the velocity pressure of the axial fan is 0.38 in. WC (95 Pa), the flow resistance of the relief system is

$$R_{\text{rel}} = \frac{0.2 + 0.4 + 0.38}{(16{,}650)^2} = 3.54 \times 10^{-9} \text{ in. WC/(cfm)}^2 \ (1.86 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6)$$

The sum of flow resistance of the return and relief systems is

$$\begin{aligned} R_{ro} &= R_{r,ru} + R_{\text{rel}} = 5.41 \times 10^{-10} + 3.54 \times 10^{-9} \\ &= 4.08 \times 10^{-9} \text{ in. WC/(cfm)}^2 \ (2.15 \times 10^{-3} \text{ Pa} \cdot \text{s}^2/\text{m}^6) \end{aligned}$$

Because a total pressure of + 0.03 in. WG (7.5 Pag) is provided by the space positive pressure and is undertaken by the supply fan, therefore, the fan total pressure of the relief fan is calculated as

$$\Delta p_{\text{rel}} = 0.2 + 0.4 + 0.38 + 0.15 - 0.03 = 1.10 \text{ in. WC (274 Pa)}$$

The supply fan curve F_s, the relief fan curve F_{rel}, the system curve of the supply system $\Delta p_t = R_{or}\dot{V}^2$, and the system curve of the return and relief system $\Delta p_t = R_{ro}\dot{V}^2$ can then be plotted. At design flow of an air economizer cycle, the system operating point for supply system *P* has $\dot{V}$ = 19,750 cfm (9320 L/s) and Δp_t = 4.73 in. WC (1176 Pa). The system operating point of the relief system has $\dot{V}$ = 16,650 cfm (7857 L/s) and Δp_t = 1.10 in. WC (273 Pa).

The pressure-volume characteristics at key points for air economizer mode at design volume flow are as follows:

	Point *o*	Point *m*	Point *r*	Point *ru*
p, in. WG	0	− 0.6	+ 0.03	− 0.12
$\dot{V}$, cfm	19,750	19,750	19,750	16,650

Air Economizer Mode, 50 Percent Design Flow

At 50 percent design volume flow rate in the air economizer mode, the variable part of the pressure loss of the supply air system when inlet vanes are partly closed is

$$\Delta p_{\text{var50}} = 9.95 \times 10^{-9}(10{,}000)^2 = 1.0 \text{ in. WC (249 Pa)}$$

And the fan total pressure of the supply fan at 50 percent design flow is

$$\Delta p_{\text{sf50}} = 1.0 + 0.75 + 0.03 = 1.78 \text{ in. WC (443 Pa)}$$

Total pressure loss of the return and relief systems is then

$$\Delta p_{r,\,ru50} = 4.08 \times 10^{-9}(6650)^2 = 0.18 \text{ in. WC (45 Pa)}$$

At 50 percent design flow, the velocity pressure of the relief fan is

$$p_{v,rf} = \left(\frac{6650}{16{,}650}\right)^2 (0.38) = 0.06 \text{ in. WC (15 Pa)}$$

The fan total pressure of the relief fan at 50 percent design flow is therefore

$$\Delta p_{\text{ref50}} = 0.18 - 0.03 + 0.06 = 0.21 \text{ in. WC (52 Pa)}$$

Pressure-volume characteristics at key points in air economizer mode at 50 percent design volume flow are as follows:

	Point *o*	Point *m*	Point *r*	Point *ru*
p, in. WG	0	− 0.15	+ 0.03	− 0.02
$\dot{V}$, cfm	10,000	10,000	10,000	6650

Warm-Up and Cool-Down Mode

During warm-up and cool-down mode, the relief fan is turned off and the relief damper is closed. As in the supply fan and exhaust fan combination, the exhaust fan is also turned off, the outdoor damper is closed, and the recirculating damper is fully opened. Air is recirculated from the space to the AHU or PU and to the space again.

Design Considerations and Controls

- As in the supply fan and exhaust fan combination system, control systems such as zone temperature, discharge temperature, duct static pressure, minimum outdoor air, and high-pressure limit control systems are discussed in Chap. 23.
- The operation of recirculating mode or air economizer mode is actuated by the air economizer control which is either temperature-based or enthalpy-based, as described in Sec. 21.2. Warm-up and cool-down mode is activated when the sensed zone temperature deviates from the set point after the AHU or PU is turned on, as described in Chap. 23.
- At minimum outdoor air recirculating mode, when the relief fan is not operating, the space positive pressure is maintained because of the airflow balance in the space. Suppose that 20,000 cfm (9438 L/s) of air is supplied to the space. If the space pressure is zero, as 3000 cfm (1416 L/s) of air is exhausted from the space by the exhaust fan, only 16,650 cfm (7857 L/s) of recirculating

air is extracted by the supply fan to mix with the 3350 cfm (1581 L/s) of outdoor air in the mixed plenum at design volume flow. The excessive amount (350 cfm or 165 L/s) of supply air raises the space pressure until the 20,000 cfm (9438 L/s) supply air is balanced with the sum of 16,650 cfm (7857 L/s) recirculating air, 3000 cfm (1416 L/s) exhaust air, and 350 cfm (165 L/s) exfiltrated air at + 0.03 in. WG (7.5 Pag) space positive pressure.

- The relief fan is often controlled by a DDC unit controller according to the input from the space pressure sensor to maintain a space positive pressure within predetermined limits.
- The main drawback of the supply and relief fan combination is the increase in pressure at the mixed plenum p_m when the supply volume flow rate of a VAV system is reduced at part-load operation during minimum outdoor air recirculating mode. The higher the pressure loss of the return system, the greater the increase in p_m. Higher pressure in the mixed box and, therefore, a lower Δp_{om} ($p_o - p_m$) cause a deficiency in outdoor air intake. Various outdoor ventilation controls have been developed recently and are discussed in Chap. 23. The operation of a relief fan reduces both mixing box pressure and space pressure.

22.6 SUPPLY FAN AND RETURN FAN COMBINATION

Supply fan and return fan combinations are widely used, especially in air systems that serve large conditioned areas. A return fan is located upstream from the junction of the recirculating flow passage and the exhaust air passage, point *ru*, as shown in Fig. 22.4*a*. Consider an AHU or PU with a supply fan and return fan combination and an exhaust system. Its operating characteristics are the same as those in Sec. 22.4 except for the following:

- A return fan is installed with the following pressure-volume characteristics:

$\dot{V}$, cfm	10,000	15,000	20,000
Δp_t, in. WC	1.47	1.23	1.10

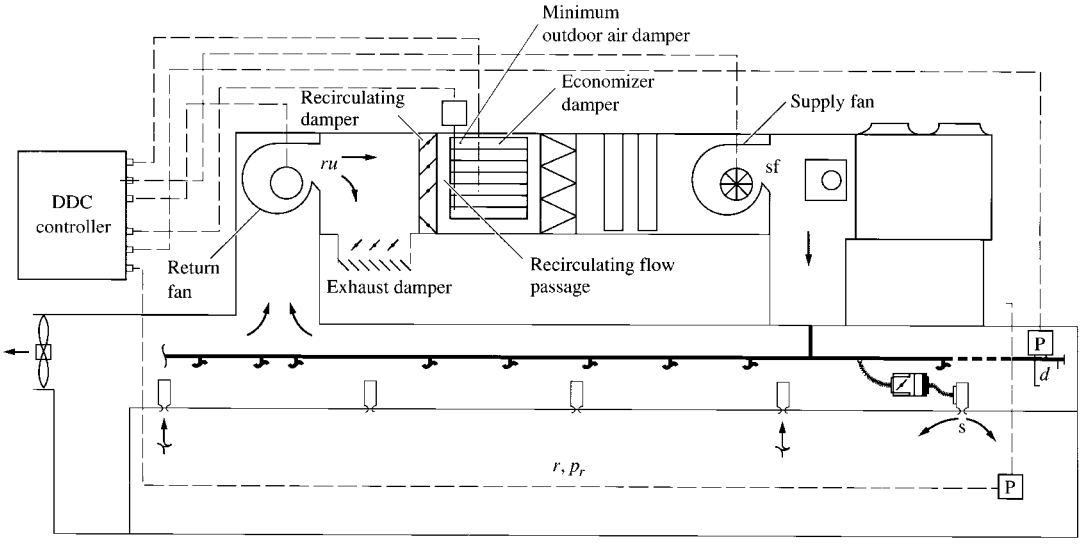
- The pressure loss of the exhaust damper after point *ru* in the exhaust passage is 0.4 in. WC (100 Pa) and is 0.2 in. WC (50 Pa) for the air louver at the maximum exhaust volume flow rate. At minimum outdoor airflow, louver pressure losses can be ignored. As in the supply and relief fan combination, a minimum outdoor air damper and an economizer damper are installed, and both have a total pressure loss of 0.4 in. WC (100 Pa) at design airflow.

Recirculation Mode

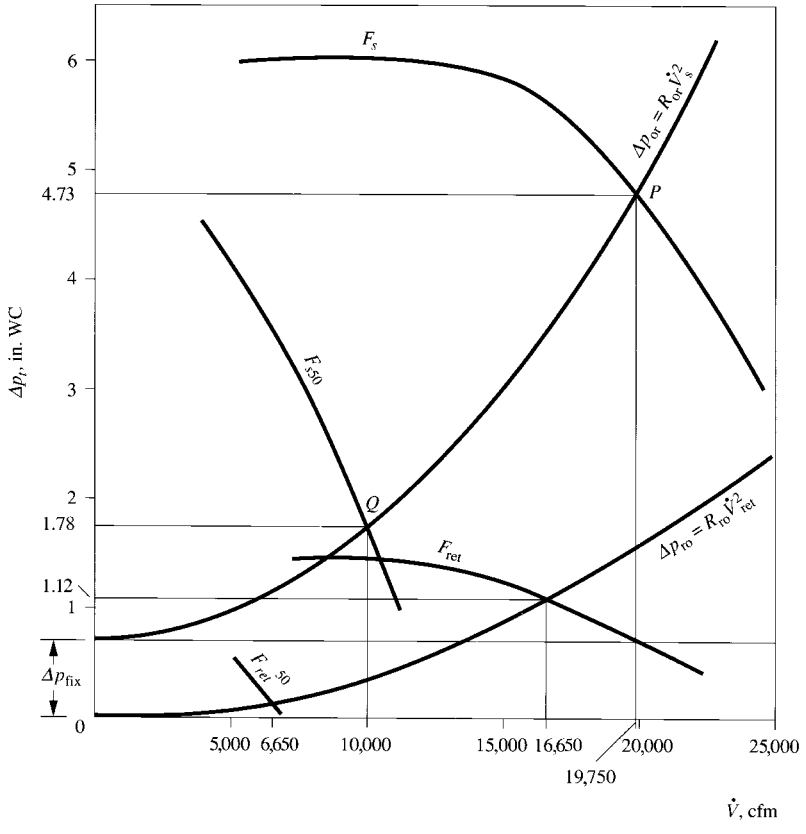
During minimum outdoor air recirculation mode, outdoor air is drawn through the outdoor air louver, duct, and damper and is mixed with recirculating air. The mixture then flows through the filters and coils and is supplied to the conditioned space. At design volume flow, a small part is exhausted, another part is exfiltrated, and the remaining portion is extracted by the return fan flowing through the return system. At point *ru*, almost all the return air is recirculated through the recirculating damper.

During minimum outdoor air recirculating mode, at design volume flow rate:

- If the return fan is not operating and the exhaust damper is closed, then the fan total pressure of the supply fan is higher than 4.65 in. WC (1156 Pa) because of the air flowing through the turned-off return fan.
- If the return fan is operating, 3000 cfm (1416 L/s) of air is exhausted through the exhaust passage and exhaust damper instead of from the separate exhaust fan. The total pressure loss of the return



(a)



(b)

FIGURE 22.4 Supply fan and return fan combination: (*a*) schematic diagram; (*b*) system characteristics on $\Delta p_t - \dot{V}$ diagram.

system Δp_{ret} increased from 0.15 to 0.7 in. WC (37 to 74 Pa). The fan total pressure of the supply fan is then

$$0.2 + 0.4 + 2.5 + 0.85 + 0.75 + 0.03 = 4.73 \text{ in. WC (1176 Pa)}$$

During the minimum outdoor air recirculating mode, the recirculating air turns 90° and flows through the recirculating damper and has a greater total pressure loss than the 3000 cfm (1416 L/s) exhaust air exhausted through exhaust damper and louver. Assume that the dynamic loss of the 90° turn of the recirculating air is 0.2 in. WC (50 Pa). Then the total pressure loss of the recirculating damper is 0.4 in. WC (100 Pa) and the fan total pressure of the return fan is

$$0.7 + 0.2 + 0.4 - 0.03 = 1.27 \text{ in. WC (316 Pa)}$$

The pressure-volume characteristics at the key points are shown below:

Δp_{ret}, in. WC		Point *o*	Point *m*	Point *r*	Point *ru*
0.15	p, in. WC	0	< -0.52	$+0.03$	< -0.12
0.15	$\dot{V}$, cfm	3,350	20,000	20,000	16,650
0.70	p, in. WG	0	-0.6	$+0.03$	1.0
0.70	$\dot{V}$, cfm	3,350	20,000	20,000	19,650

At 50 percent of the design flow rate, if the return fan is turned off and the exhaust damper after point *ru* is closed, then the fan total pressure of the supply fan will be higher than 1.72 in. WC (428 Pa), as in the supply fan and exhaust fan combination. The space pressure p_r drops to -0.03 in. WG (-7.5 Pag). Total pressure at point *ru* may be lower than -0.07 in. WG (-17 Pag) and at point *m* lower than -0.26 in. WG (-65 Pa) because of the pressure loss of air flowing through the turned-off return fan.

If the total pressure loss of the return system is at 0.7 in. WC (174 Pa) and 3000 cfm (1416 L/s) of air is exhausted through the return fan instead of the exhaust fan, and also if the return fan is operating at 50 percent of design flow rate, then the fan total pressure of the return fan is about $(10{,}350/20{,}000)^2(0.67 + 0.2 + 0.4) = 0.34$ in. WC (85 Pa) and the total pressure at *ru* is about 0.16 in. WG (40 Pag).

Air Economizer Mode

During air economizer mode operation at the design volume flow rate, the supply and exhaust volume flow rate and the pressure losses of various sections of the supply air system are the same as in minimum outdoor air recirculating mode. The fan total pressure of the supply fan is 4.73 in. WC (1176 Pa).

Because the total pressure at point *ru* is raised to 1.0 in. WG (249 Pag) to overcome the pressure loss of the exhaust flow passage and the discharge velocity pressure at the outlet of 0.40 in. WC (100 Pa), the return fan total pressure is then

$$\Delta p_{ret.f} = 0.15 - 0.03 + 0.60 + 0.40 = 1.12 \text{ in. WC (278 Pa)}$$

If the total pressure loss of the return system is increased to 0.70 in. WC (174 Pa) and the return air is exhausted through the return fan and exhaust damper, during air economizer mode operation at design flow rate, then the return fan total pressure is the same as in the recirculating mode

$$\Delta p_{ret.f} = (0.70 - 0.03) + 0.60 + 0.40 = 1.67 \text{ in. WC (415 Pa)}$$

Pressure-volume characteristics at the key points in air economizer mode at design volume flow, when the total pressure loss of the return system is 0.15 in. WC (37 Pa) and space air is exhausted through the exhaust fan, are as follows:

	Point *o*	Point *m*	Paint *r*	Point *ru*
p, in. WG	0	− 0.6	+ 0.03	1.0
$\dot{V}$, cfm	20,000	20,000	20,000	16,650

During air economizer mode, if the total pressure loss of the return system is increased to 0.70 in. WC (174 Pa) and the return air is exhausted through the return fan and exhaust damper, at 50 percent of design volume flow, then the fan total pressure of the return fan is

$$\Delta p_{\text{ret.f50}} = \left(\frac{9650}{20{,}000}\right)^2 (0.70 + 1.0) - 0.03 = 0.37 \text{ in. WC (91 Pa)}$$

and the pressure-volume characteristics at the key points are as follows:

	Point *o*	Point *m*	Point *r*	Point *ru*
p, in. WG	0	− 0.15	+ 0.03	0.23
$\dot{V}$, cfm	10,000	10,000	10,000	9650

Controls

As in the supply and relief fan combinations, the controls for supply and return fan combination systems operated at minimum outdoor air recirculating mode and air economizer mode operations are discussed in Chap. 23.

22.7 COMPARISON OF THREE FAN COMBINATION SYSTEMS

An air system with supply and exhaust fan combination is simpler and less expensive. Such a system is not suitable for operation in air economizer mode for a high space pressure. In a supply and exhaust fan combination, a comparatively lower pressure drop in the return system is necessary to prevent an unacceptable low pressure in the mixed plenum, such as $p_m < -1$ in. WG (− 250 Pag). Such low mixed plenum pressure induces air leakage and may impede minimum outdoor air control.

The operating characteristics of the supply and relief fan combination and the supply and return fan combination can be compared as follows:

- In a supply and relief fan combination in recirculating mode, total pressure at point *ru* may be negative; in a supply and return fan combination, total pressure at *ru* must be positive, in order to overcome the pressure loss of the exhaust passage.
- With a relief fan, the mixed plenum pressure p_m is always negative. With a return fan, p_m may be positive if the return fan is too large. A positive p_m means that there will be no outdoor air intake into the mixing box. Such a condition should never occur.
- When a negative space pressure must be maintained, a relief fan causes the pressure in the mixed plenum to be smaller (more negative) than a return fan during minimum outdoor air recirculating mode.
- When there is a considerable pressure drop in the exhaust or relief flow passage, a return fan requires high positive pressure at point *ru*, so energy is wasted across the recirculating damper.
- A supply and relief fan combination with a lower total pressure loss return system is often more energy-efficient than a supply and return fan combination during recirculating mode operation, for the following reasons: (1) The pressure drop across the recirculating damper is smaller.

(2) The relief fan heat gain does not increase coil load. (3) If the return fan is not operating and there is no bypass passage, there is an additional pressure drop across the idle return fan.

- An axial relief fan is sometimes simpler in layout and installation, and is therefore less expensive to install, than a centrifugal return fan. If an axial fan is used, noise attenuation must be considered.

Therefore, for those air systems with a smaller total pressure loss in return duct system, usually less than 0.5 in. WC (125 Pa), especially for those systems also with a considerable pressure drop in the exhaust or relief flow passage, a supply and relief fan combination is recommended.

For air systems with a greater total pressure loss in the return system, or those that require a negative space pressure, a supply and return fan combination may be more appropriate.

22.8 PRESSURE FLOW CHARACTERISTICS

Pressure Flow Characterisitcs for a Supply and Return Fan Combination System

During a supply and return fan combination VAV system design, it is critical to have low enough negative static pressure in the mixing box to extract the required amount of outdoor ventilation air and a positive pressure in the return chamber (junction of the return system and exhaust system) to exhaust the proper amount of return air to outdoors to maintain a predetermined space pressure.

According to the calculations and analyses in the previous sections, to achieve these:

- Correctly divide the system total pressure loss into two parts: (1) that part undertaken by the fan total pressure of the return fan, which starts from the return slot inlet *rt* and includes obstructions in the ceiling plenum, if any, return ducts and fittings, 90° elbow in the return chamber, the positve pressure required in the return chamber to exhaust return air through the exhaust damper and louver or the recirculating damper, as shown in Fig. 22.1; and (2) that part undertaken by the fan total pressure of the supply fan which starts from the negative pressure in the mixing chamber, total pressure loss of the filter, coils, direct-fired furnace, 90° elbow, supply duct and duct fittings, flexible ducts, VAV box, diffuser, and the positive pressure maintained in the conditioned space.
- For a rooftop unit with system components as shown in Fig. 22.1, the total pressure in the return chamber point *ru* must be high enough to exhaust part of the return air from the exhaust damper and louver, or to overcome the total pressure loss of a 90° elbow of airflow and the recirculating damper during the recirculating mode at summer design conditions. The total pressure loss of the 90° turn and the recirculating damper is often greater.
- In VAV systems, they are often installed with two outdoor dampers: minimum outdoor damper and 100 percent outdoor damper. During the minimum outdoor air recirculating mode, only the minimum outdoor air damper is activated. The total pressure loss of the minimum outdoor 0.4 + 0.2 = 0.6 damper and louvers is still assumed to be 0.4 + 0.2 = 0.6 in. WC (150 Pa) or less for effective control and energy saving. Then a negative pressure of 0.6 in. WC (150 Pa) or less is needed during minimum outdoor air recirculating mode.

  During the air economizer mode, 100 percent outdoor air is required at the design flow rate. Both outdoor air dampers are opened. A negative pressure of 0.6 in. WC (150 Pa) or less is required to extract the required amount of outdoor air.
- For a supply and return fan combination VAV system, accurately calculate the total pressure loss of the system components, supply and return ducts and duct fittings, flexible ducts, VAV box, and diffusers. It is necessary to draw a system pressure diagram for a supply and return fan combination system and ensure that the negative pressure, at a low enough magnitude, is in the mixing box and that there is adequate positive pressure at the junction of the recirculating passage and the exhaust passage, point *ru* (return chamber, see Fig. 22.1).

System pressure diagram should be drawn by system total pressure. During the construction of a system pressure diagram of a supply fan, the static regain in the supply duct is obtained at the expense of the velocity pressure of the supply fan. Only during the minimum outdoor air recirculating mode is the velocity pressure of the return fan utilized.

- It is often cost-effective to install adjustable-frequency variable-speed drive for both supply and return fans to adjust the fan speed for large fans during operation. Set the fan total pressure of the supply and return fans as well as the damper position during commissioning to achieve a required system pressure distribution and an optimum energy use.

Variation of Pressure in the Mixing Box

The negative total pressure in the mixing box $-p_m$ is related to the upstream outside atmospheric pressure p_o, all in in. WG (Pa), as follows:

$$-p_m = p_o - \Delta p_{om} \tag{22.6}$$

where Δp_{om} = pressure differential between the outside atmosphere and the air pressure in the mixing box, in. WC (Pa). The negative total pressure in the mixing box $-p_m$ is also related to the outside atmospheric pressure p_o through the space pressure $-p_r$, in in. WG (Pag), by the following relationship:

$$\begin{aligned} -p_m &= p_o + \Delta p_{or} + \Delta p_{r,s} + \Delta p_{\text{rec},d} \\ -p_m &= p_r + \Delta p_{r,s} + \Delta p_{\text{rec},d} \end{aligned} \tag{22.7}$$

where Δp_{or} = pressure differential between outside atmosphere and space, in. WC (Pa)
$\Delta p_{r,s}$ = total pressure loss of return system, in. WC (Pa)
$\Delta p_{\text{rec},d}$ = total pressure loss of recirculating damper, in. WC (Pa)

During recirculating mode, part-load operation, the volume flow rates of the supply and return air are reduced, and the total pressure loss of the return system $\Delta p_{r,s}$ reduces accordingly. At the same time, the variation of the mass flow rates of the supply, return, exhaust, and exfiltrated air forms new airflow balances and new space pressures. Because of the relationship between $-p_m$, p_r, and p_o, as shown in Eq. (22.7), the result is often a higher $-p_m$ (less negative p_m) and less extracted outdoor ventilation air through the outdoor dampers at recirculating mode during part-load operation, as analyzed in Secs. 22.4, 22.5, and 22.6. The opening of the outdoor and recirculating dampers should be adjusted in order to extract the required amount of outdoor air.

Field Survey of System Pressure Characteristics of a VAV System Using a Supply and Relief Fan Combination

Figure 22.5 shows the pressure characteristics of a VAV system using a supply and relief fan combination. This VAV cooling system serves the fifth floor of a high-rise public library with a conditioned floor area of about 16,500 ft^2 (1534 m^2). There are two AHUs and two fan rooms, each with a volume flow rate of about 12,650 cfm (5970 L/s) and a fan total pressure of 4.25 in. WC (1057 Pa) to serve this floor. Each fan room has three relief fans and three relief dampers. The volume flow rate of relief air is modulated by relief dampers, which are controlled by an electronic controller actuated by a signal from a pressure sensor in the ceiling plenum. In the perimeter zone, winter heating is provided by the electric heating coil located downstream from the VAV box.

On September 12, 1983, the system pressure characteristics of the VAV cooling system were measured. Control of relief dampers was deliberately deenergized. The supply volume flow rate of AHU 2 measured was about 12,000 cfm (5663 L/s). The pressure differences between various points in fan room 2 in the VAV cooling system during minimum outdoor air recirculating mode and 100 percent air economizer cycle were as follows:

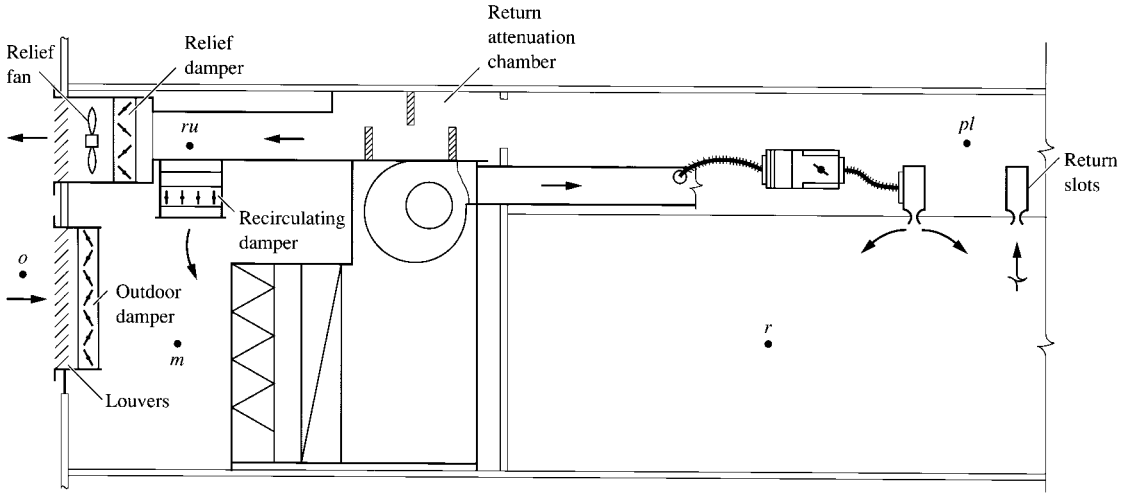


FIGURE 22.5 Pressure characteristics of a VAV cooling system using supply-relief fan combination.

	Recirculating mode	Air economizer
Outdoor damper	10% open	100% open
Outdoor air, cfm	3330	
Recirculating damper	100% open	100% closed
Relief fans operated	1 fan	All 3 fans
Relief damper	15% open	100% open
$\Delta p_{\text{pl},o}$ in. WC (Pa)	+ 0.06 (15)	+ 0.018 (4.5)
$\Delta p_{\text{pl},m}$ in. WC (Pa)	+ 0.20 (50)	
$\Delta p_{o,m}$ in. WC (Pa)	+ 0.14 (35)	
$\Delta p_{r,\text{pl}}$ in. WC (Pa)	+ 0.004 (1)	+ 0.004 (1)

During the air economizer cycle, when the outdoor damper is fully open and the recirculating damper was closed, the pressure difference between the ceiling plenum and outdoors $\Delta p_{\text{pl},o}$ was + 0.038 in. WC (9.5 Pa) when only two relief fans were operating, and $\Delta p_{\text{pl},o}$ was + 0.04 in. WC (10 Pa) when only one relief fan was operating.

The pressure difference between the conditioned space and the air in the ceiling plenum $\Delta p_{r,\text{pl}}$ was so small because of gaps between the ceiling tiles. During recirculation mode, the space pressure is about + 0.064 in. WG (16 Pag). Space pressure was lower when more relief fans were operating during the air economizer cycle.

22.9 SMOKE CONTROL AND FIRE SAFETY

Fire Safety in Buildings

Fire safety is a critical design factor in high-rise buildings. On November 21, 1980, in Las Vegas, the MGM Grand Hotel fire took 85 lives. Smoke inhalation is the primary killer in building fires. According to the annual survey taken by the National Fire Protection Association (NFPA), there was a slight increase in civilian fire deaths to a total of 4585 people who lost their lives in 1995,

which is a 7.3 percent increase over a record low in 1994. In 1995, in the United States, public fire departments responded to approximately 2 million fires, and property damage due to fire was nearly $9 billion.

McGreal (1997) noted that building fires produce both smoke and heat. Smoke is the real killer. Smoke consists of the airborne particulates, in the form of either solid or liquid, and gases which evolve when a material undergoes pyrolytic combustion.

Studies around 1980 found that more than 80 percent of fire deaths were due to the inhalation of fire combustion products. About one-half of the deaths are due to the inhalation of carbon monoxide (CO). Another 16 percent of the deaths were caused by the combined effect of CO and hydrogen cyanide and heart disease.

Fire protection and fire safety in buildings include the following measures:

- Fire compartmentalization
- Fire-resistant construction
- Fire alarm system
- Automatic sprinkler system
- Smoke control system including stairwell pressurization and zone smoke control system
- Fire protection management and coordination

Smoke control systems, which include stairwell pressurization and zone smoke control, are features of HVAC&R system design and are discussed here.

National codes and local codes must be followed during the design of the smoke control system in buildings.

Smoke Movements in Buildings

Figure 22.6 shows smoke movements in a 10-story experimental building. According to *ASHRAE Handbook 1999, HVAC Applications*, the typical leakage area ratio A_{leak}/A_{fl}, which is the ratio of leakage area A_{leak} to floor area A_{fl}, both in ft^2 (m^2) for commercial buildings of average tightness, is as follows:

External walls, including cracks around windows and doors	0.00021
Stairwell walls, construction cracks only	0.00011
Elevator shaft walls, construction cracks only	0.00084
Floors, construction cracks, and area around penetrations	0.000052

The cracks around the stairwell door can be typically taken as 0.25 ft^2 (0.023 m^2), and the cracks around each elevator on each floor can also be taken as 0.25 ft^2 (0.023 m^2).

A building fire can be simulated by burning wooden sticks in a second-floor corridor with a peak energy release of 900 Btu/s (950 kJ/s) for a certain period while the building is under the following operating conditions:

- The stair pressurization system and smoke control system are not operated.
- All stairwell doors are closed except that on the second floor, which is opened $^1/_2$ in. (13 mm) by the high temperature of the hot gas.
- All windows and exit doors are closed.
- Outdoor temperature is 30°F (− 1.1°C), and there is a north wind at 5 mph (2.23 m/s).

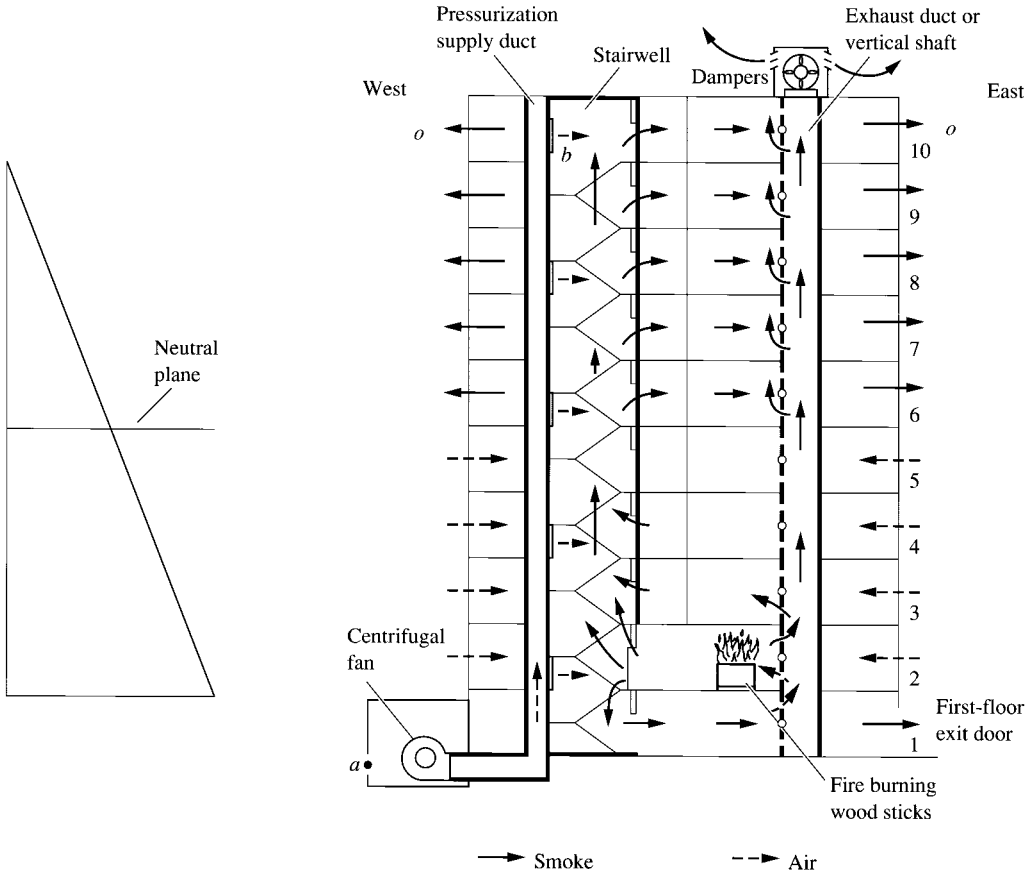


FIGURE 22.6 Smoke movements in a 10-story experimental building.

When the fire is ignited, smoke movements are mainly caused by the expansion and buoyancy forces of hot gas at a maximum temperature between 850 and 1000°F (454 and 538°C), the stack effect from outdoor-indoor temperature differences, and the wind effect.

Smoke moves from fire on the second floor to the upper floors through stairwells, elevator shafts, service shafts, vertical risers, and floor cracks; and smoke discharges to the outdoor atmosphere through window cracks, elevator machine-room openings, and other openings in the upper floors. Outdoor air enters the building below the neutral plane and discharges to the outdoors above the neutral plane because of the stack effect. Oxygen supply to the fire enters through second-floor window cracks, openings in the building envelope, and vertical air passages from the first and third floors.

According to Klote (1990), the CO_2 and CO levels on the highest floor of the experimental building during tests without stairwell pressurization and smoke control are 0.15 percent CO_2 and 0.015 percent CO. In tests with stairwell pressurization and smoke control, the levels are 0.002 percent CO_2 and 0.001 percent CO.

In winter, the stack effect assists the stairwell pressurization in preventing the smoke from contaminating the stairwell. This result is verified by experiments in Tamura (1990*b*).

Effective Area and Flow Rates

In several airflow passages connected in parallel, the effective area A_e, in ft^2 (m^2), can be calculated as

$$A_e = A_1 + A_2 + \cdots + A_n \tag{22.8}$$

where $A_1, A_2, \cdots, A_n$ = airflow areas for path 1, 2, $\cdots$, n, ft^2 (m^2)

In airflow paths connected in series, each with a flow area $A_1, A_2, \cdots, A_n$, in ft^2 (m^2), the effective area for these airflow paths connected in series A_e can be calculated as

$$A_e = \left(\frac{1}{A_1^2} + \frac{1}{A_2^2} + \cdots + \frac{1}{A_n^2}\right)^{-1/2} \tag{22.9}$$

If the flow coefficient is taken as 0.65 and air density $\rho_a = 0.075$ lb/ft^3 (1.2 kg/m^3), the air volume flow rate $\dot{V}$, in cfm, flowing through a crack, gap, or opening can be calculated as

$$\dot{V} = 2610A(\Delta p)^{1/2} \tag{22.10}$$

where A = flow area or effective area, ft^2 (m^2)
Δp = pressure difference across flow path or opening, in. WC (Pa)

22.10 EFFECT OF AUTOMATIC SPRINKLER ON FIRE PROTECTION

Automatic sprinkler systems are effective and reliable fire protection systems and should be installed in buildings to provide fire protection for the occupants during a building fire. Based on experimental results, Mawhinney and Tamura (1994) summarized the effect of automatic sprinklers on fire protection as follows:

- For unshielded fires, automatic fire sprinklers reduced both smoke and fire hazard to negligible levels. Unshielded fire is a fire exposed to the water sprays from activated sprinklers.
- For shielded fires with activated water sprays from the sprinklers, fire continues to burn at a reduced rate and to produce smoke until the fuel inside the shielded area has all burned. Concentrations of CO_2 and CO in the smoke were dangerously high. Shield fires represent fires that are shielded from the water sprays delivered by the sprinklers above because of a desk, table, or other furniture.

  According to experiments, shielded crib fires produced CO concentrations as high as 8 and 9 percent and CO concentrations between 1.0 and 1.5 percent (10,000 and 15,000 ppm) in a multistory building. Peacock et al. (1989) indicated that a concentration of 15,000 ppm CO would present a severe life-safety hazard to any one trapped on the fire floor.
- Sprinklers reduced the temperature and radiant heat from shielded fires to nonthreatening levels within the boundary of the area covered by the sprinkler system. For a sprinkler system design that meets NFPA Standard 13 requirements, a moderate increase in ventilation to the fire floor during a building fire should not significantly increase the fire temperatures.
- Buoyancy forces for shielded, sprinklered fire were almost negligible. The recommendation for the pressure difference between different smoke control zones (fire zone and adjacent zones) $\Delta p = 0.05$ in. WC (12.5 Pa) in NFPA (1988) for zone smoke control design in sprinklered buildings (ceiling height < 9 ft or 2.7 m) is more than sufficient to prevent smoke movement, provided that the door to the fire floor remains closed.
- Recommended practice for zone smoke control design should allow for some airflow into the stairwell to prevent the spread of smoke into the stairwell when the door to the fire floor is opened.

- The assumption that smoke will never become a threat to life safety in a fully sprinklered building needs to be reexamined.

22.11 SMOKE CONTROL IN ATRIA

ANSI/NFPA 92A and 92B

The NFPA first published ANSI/NFPA 92A, *Recommended Practice for Smoke Control Systems*, in 1987. It covered smoke control barriers, airflows, and the pressure differential between different control zones. It also contained the guidelines to provide a smokeproof enclosure using stairwell pressurization, as well as smoke control for elevator shafts by using fire floor exhaust, and elevator lobby pressurization. The latest NFPA 92A edition, in 1996, includes smoke control system supervision and instrumentation.

Because the smoke control in large zones is quite complicated, another document, ANSI/NFPA 92B, *Guide for Smoke Management Systems in Malls, Atria, and Large Areas,* was first published in 1991. The latest edition is the 1995 edition.

Smoke Management in Atria, Malls, and Large Areas

Klote (1997) and ANSI/NFPA 92B recommended the following smoke management in atria:

- Exhaust the smoke from the top of the atrium in order to achieve a steady, clear height for a steady fire. Steady fire is an idealization of actual unsteady fire. Steady fire has a constant heat release rate. Consider that the only flow into the smoke layer formed at the upper layer of a atrium is from the plume, and the only flow from the smoke layer is the smoke exhaust. Plume comprises the combustion products above the fire. The exhaust flow must equal the flow from the plume.

  This method and approach have many simplified assumptions, such as that the simple plume mass flow equation is valid, there is a constant heat release rate, clear height is greater than the mean flame height, smoke layer is adiabatic, and plume flow and exhaust are the only significant mass flows into or out from the smoke layer.
- Upper-layer exhaust is not necessary for an atrium with sufficient smoke-filling capacity. Occupants have time for both decision making and evacuation before the smoke fills the place where they are located. Various approaches to smoke filling can be used to conservatively estimate the filling time.
- Air can be supplied to the communicating space to provide a specific average velocity at the opening to the atrium to prevent smoke flowing from the atrium to the communicating space.
- When the above basic methods are not applicable, physical modeling and computing fluid dynamics (CFD) analysis can be used.
- Because a layer of hot air is formed under the ceiling of an atrium, and this layer of hot air prevents the smoke from reaching the ceiling, ceiling-mounted smoke detectors are usually not recommended for atrium applications. Beam smoke detectors mounted on balconies oriented horizontally are recommended to detect smoke in the plume.
- The exhaust system can pull some air below the smoke layer into the exhaust inlet if the smoke layers are relatively thin. This plugholing of outdoor air can lower the smoke layer and expose occupants to smoke. Klote (1999) noted the maximum flow Q_{smoke} of smoke without plugholing depends on the depth of the smoke layer and the temperature of the smoke. If the total smoke exhaust needed is greater than Q_{smoke}, a number of inlets to compensate exhaust are needed. These inlets should be placed far enough from each other to avoid influence on their flow.

22.12 STAIRWELL PRESSURIZATION AND ZONE SMOKE CONTROL

Stairwell Pressurization

A stairwell pressurization system uses fans to pressurize the stairwells to provide a smokefree escape route for the occupants in case of a building fire. A stairwell pressurization system is a kind of smoke control system. National and local codes require stairwell pressurization systems in high-rise buildings. In a stairwell pressurization system, all interior stairwells are pressurized to a minimum of 0.15 in. WC (37 Pa) and a maximum of 0.35 in. WC (87 Pa) when all stairwell doors are closed.

City of New York Local Code 1979 requires a minimum air supply flow rate of 24,000 cfm (11,326 L/s) plus 200 cfm (94 L/s) per floor for the stairwell pressurization system. The maximum allowable pressure difference between the stairwell and the floor space is 0.40 in. WC (100 Pa) whether stairwell doors are opened or closed. The minimum allowable pressure difference is 0.10 in. WC (25 Pa) when all stairwell doors are closed, or 0.05 in. WC (13 Pa) when any three stairwell doors are open. An alternative is to maintain at least 0.05 in. WC (13 Pa) or a minimum average air velocity of 400 fpm (2 m/s) at the stairwell door when any three stairwell doors are opened. The force required to open a stairwell door must not exceed 25 lb_f (111 N) at the doorknob.

A stairwell pressurization system consists of centrifugal or vane-axial fans, a stairwell pressurization supply duct with several supply air inlets, relief vents, and a control system, as shown in Fig. 22.7. Outdoor air is extracted directly by the centrifugal fan. It is forced into the supply duct and then supplied to the stairwell through supply inlets. When the stairwell is pressurized to a pressure typically 0.10 to 0.40 in. WC (25 to 100 Pa) higher than that of the air outside the stairwell on various floors across the stairwell doors, the smoke will not enter the stairwell, even an open stairwell door. Air supplied into the stairwell is discharged through the open stairwell doors, leakage area around closed stairwell doors, relief vents, or other openings to the rest of the building, and then is discharged to the outdoors.

If the stairwell is an airtight enclosure or its doors have very small leakage areas, it will be overpressurized when all stairwell doors are closed. The pressure difference across the stairwell doors may be greater than 0.3 in. WC (75 Pa). Often, a large force is required to turn the stairwell doorknob to open it. The total force required to open a stairwell door should not exceed 25 to 30 lb_f (111 to 134 N), or it will be too difficult to open stairwell doors during evacuation. Methods of overpressure relief are discussed later.

The air velocity at the opened stairwell door on the fire floor required to prevent the backflow of smoke from the fire to the stairwell is called the *critical velocity* v_{crit}, in fpm (m/s). The outward flow air velocity from the stairwell through the open door on the fire floor should be greater than v_{crit}.

Characteristics of Stairwell Pressurization

Centrifugal or vane-axial fans can be used for stairwell pressurization. Fans can be installed either at the bottom level of the building (bottom injection) or on the rooftop (top injection). Bottom injection is preferable because it minimizes the possibility of smoke inhalation and optimizes the stack effect to assist stairwell pressurization during winter. In any case, the fan intake must be remote from the smoke exhaust during a building fire. If the fan room for stairwell pressurization is located on the rooftop, facilities must be provided to minimize the influence of wind pressure on fan performance.

Multiple injections, in which air is supplied from multiple inlets into the stairwell, provide a more even pressure distribution along the stairwell than a single injection from the top or bottom. Typically, each supply inlet serves two or three floors.

Open-tread stairs provide less flow resistance than closed-tread stairs. This difference becomes more prominent when occupants are walking on the stairs during evacuation. Compartmentalization

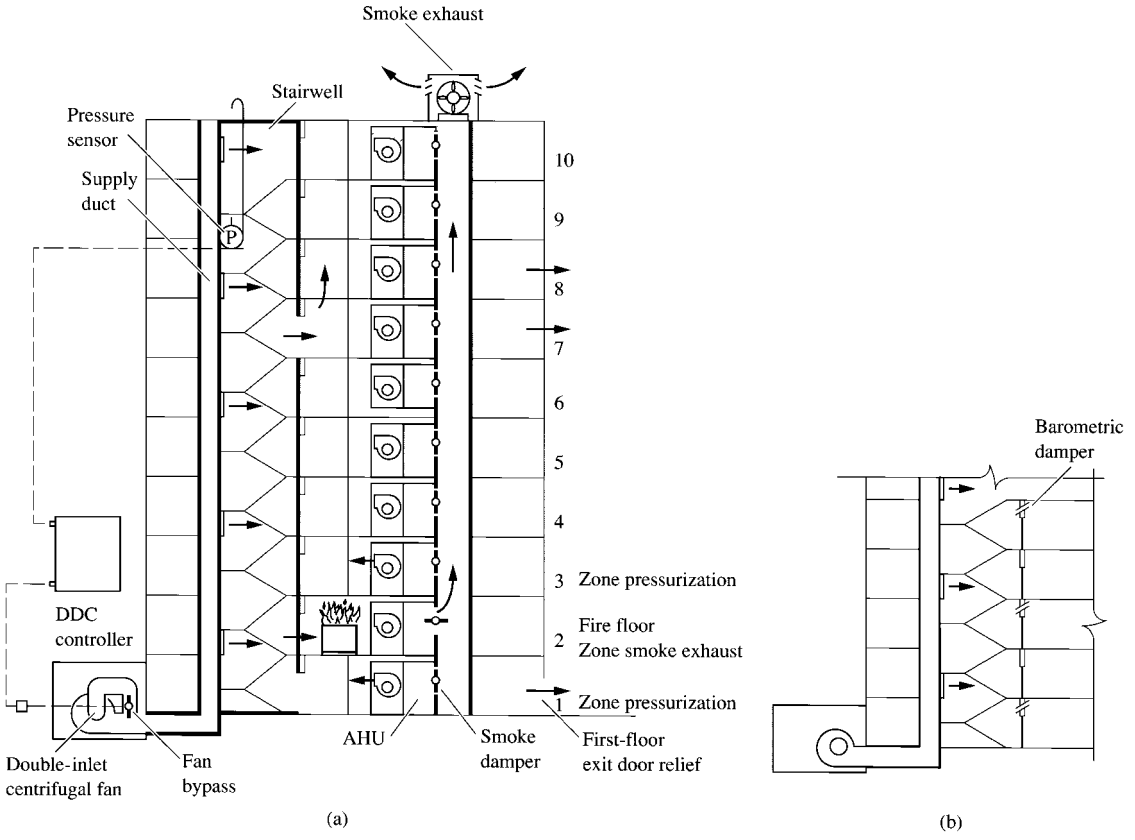


FIGURE 22.7 Stairwell pressurization and zone smoke control systems: (*a*) fan bypass overpressure relief; (*b*) overpressure relief vents (barometric damper).

of the stairwell into many sections, such as serving four to five floors, may not provide the airflow rate necessary for stairwell pressurization when two or three stairwell doors are open at the same time.

Overpressure Relief and Feedback Control

When a stairwell pressurization system is operating and all the stairwell doors are closed, the pressure difference across the stairwell doors eventually exceeds the maximum permissible limit. Two methods are currently used to relieve the stairwell pressure. Overpressure relief is achieved by opening first-floor exit doors (as shown in Fig. 22.7*a*) or overpressure relief vents (barometric dampers, as shown in Fig. 22.7*b*). Variable volume of supply air is achieved by means of feedback control (as shown in Fig. 22.7*a*).

Exit Door Relief. When a stairwell pressurization system is turned on, the interlocked control system automatically opens the first-floor exit door to relieve stairwell pressure. Overpressurized air in the stairwell is discharged to the outdoors. Because it is necessary to open the first-floor exit door to evacuate the occupants during a building fire, this is a simple and effective means of overpressure relief.

Test results in Tamura (1990*b*) showed that when stairwell pressurization was activated and the first-floor exit door was used as overpressure relief, the maximum pressure difference across the

stairwell doors was lower than 0.25 in. WC (65 Pa) in summer and winter. During nonfire conditions, pressure differences were between 0.05 and 0.1 in. WC (13 and 25 Pa) across the stairwell doors if there was a vent in the external wall on the second floor. When a building fire was simulated on the second floor, there was no smoke backflow from the fire floor through the stairwell door when the stairwell doors on the first and second floors were opened and the vent on the second floor external wall was also opened. Smoke backflow occurred when stairwell doors on the first, second, and third floors were open.

Overpressure Relief Vents. A relief vent is typically an assembly in which a fire damper is connected in series with a barometric damper. The barometric damper is normally closed. If the stairwell is pressurized above a predetermined limit, the vents open and relieve air to various building floors. A counterweight in the barometric damper sets the maximum pressure limit. Fire dampers are normally closed, and open when the stairwell pressurization system is turned on in case of a building fire.

According to the experimental results of Tamura (1990*b*), the performance of the overpressure relief vents at a supply flow rate of 28,000 cfm (13,210 L/s) was slightly better than the performance of a ground-floor exit door at a supply flow rate of 17,800 cfm (8400 L/s). If supply flow rates are the same, the difference is further reduced.

Fan Bypass. When the stairwell is overpressurized, a pressure sensor located inside the stairwell signals a feedback control to open a fan bypass damper so that part of the supply air returns to the centrifugal fan inlet, as shown in Fig. 22.7*a*. The air volume flow supplied to the stairwell is reduced until the pressure at the pressure sensor drops below a preset value. If the pressure sensor is located on the rooftop or outdoors, it should be shielded from the influence of wind.

Variable-Speed or Controllable-Pitch Fan. When excessive pressure is detected in the stairwell, a controller actuates an adjustable-frequency variable-speed drive to reduce the speed of a centrifugal fan or to vary the blade pitch of a vane-axial fan to maintain the required pressure inside the stairwell at the pressure sensor. The test results in Tamura (1990*b*) showed that both fan bypass and variable-speed controls require a response time of more than 5 min. Variable-speed control is slightly faster. Because reliability is the primary factor in stairwell pressurization control, and because of its very short operating period, it may not be worthwhile to install an expensive adjustable-frequency ac inverter to provide variable-speed control.

Stair and Shaft Vents

ASHRAE/IESNA Standard 90.1-1999 mandatorily specifies that stair and elevator shaft vents shall be equipped with motorized dampers which are automatically closed during normal building operation and are interlocked to open by fire and smoke control detection systems when required.

Zone Smoke Control

Zone smoke control requires that a building be divided into a number of smoke control zones. The control zones are separated from one another by barriers, such as walls, floors, ceilings, and doors. In high-rise buildings, each floor is a separate zone, or it may be subdivided into many smoke control zones, or a smoke control zone consists of more than one floor.

In case of a building fire, the spread of smoke from the zone of fire origin (the smoke zone) to adjacent zones is limited by pressure differences and airflows. The required pressure differential depends on the pressure produced by the fire gases, i.e., the temperature of the building fire. To form a pressure difference between the smoke zone and adjacent control zones, zone smoke control provides smoke exhaust on the fire floor (smoke zone) by opening the smoke damper connected to the smoke exhaust duct. At the same time, it supplies outdoor air to the floor or floors above and below the fire floor (to control zones adjacent to the smoke zone) and pressurizes

them to prevent smoke contamination by operating the air-handling units (AHUs) on these floors and closing the smoke dampers connected to the smoke exhaust duct, as shown in Fig. 22.7*a*. During the smoke control mode, recirculation dampers in these AHUs are fully closed. Outdoor air should be extracted directly through the coils, supply fans, and supply to the duct diffusers to pressurize these control zones These mechanisms should be controlled by a DDC unit for fire protection management overriding all other HVAC&R controls during a building fire.

Lougheed (1997) noted that the 0.05 in. WC (12.5 Pa) minimum design pressure difference between the smoke zone and adjacent zones suggested by ANSI/NFPA 92A (1993) would be adequate for zone smoke control applications in sprinklered office buildings. In most cases, a lower pressure of approximately 0.028 in. WG (7 Pa) would still exceed the pressure produced by the fire gases. Lougheed (1997) recommended a minimum required pressure difference of 0.1 in. WC (25 Pa) for an unsprinklered building with a ceiling height of 9 ft (2.74 m).

The supply and exhaust volume flow rates for zone smoke control are often matched with the HVAC&R system in buildings, especially when an air economizer cycle is used. The exhaust air from the AHU can be connected to a smoke exhaust duct, and the exhaust fan is generally located on the rooftop.

In an air system operated only in recirculation mode, a smoke exhaust system for zone smoke control should be installed. In such circumstances, the smoke exhaust volume flow rate can be determined to equal about 6 air changes per hour (ach).

Design Considerations

The object of smoke control is to provide a smokefree escape route for occupants through the stairwell to the outdoors during a building fire. This is a part of the building fire protection scheme. The primary considerations are safety and reliability. Because the performance of stairwell pressurization and the zone smoke control are related, they should be considered an integrated smoke control system during system design.

Operation of both stairwell pressurization and zone smoke control systems must meet the requirements of national and local codes and provide a smokefree escape route even under the following conditions:

- A fire reaches a fire temperature of up to 1200°F (650°C) when the sprinkler system fails to operate.
- Three or four stairwell doors, including the door on the fire floor, are opened simultaneously.
- Fire floor smoke exhaust is performed mainly by a zone mechanical smoke exhaust system.
- During summer operating conditions, the stack effect does not act as an additional assistant.

Recently, it has been recommended that smoke control technology be extended to elevator shafts and lobbies used to evacuate disabled persons during a building fire. An elevator shaft and elevator lobby pressurization system should then be installed.

During a building fire, HVAC&R should operate according to fire safety and smoke control requirements. Smoke control systems are automatically actuated by a water flow indicator from the automatic sprinkler system or from an area smoke detector. It is important to have feedback mechanisms to verify system operation and performance, such as separate alarms to signal smoke migration or burning fire, and adequate pressure level at key points during pressurization and evacuation.

Volume Flow Rate

Volume flow rate is the primary determinant of the performance of stairwell pressurization and zone smoke control systems. The volume flow rate of a zone smoke exhaust system from the fire floor is

usually 6 ach. This volume flow rate is used by many designers and has been verified as appropriate in field tests by Tamura (1991) and Klote (1990, Part II). The volume flow rates of air supplied to the floors immediately above and below normally should be the same as the supply volume flow rate of the AHU or packaged unit for that floor.

Air discharged through the open stairwell doors and leaked through the closed stairwell doors and walls can be summarized into the following types:

- Air discharged through an open stairwell door on the fire floor with a critical velocity v_{crit}
- Air discharged through the open first-floor exit door
- Air discharged through open stairwell doors on floors other than the fire floor and the first floor
- Air leaked through cracks around the stairwell door and in the stairwell wall

The volume flow rate supplied to a stairwell pressurization system then can be calculated as follows:

$$\dot{V}_{s,p} = (v_{crit} + v_{exit})A_{door} + \dot{V}_{o,d}N_{o,d} + \dot{V}_{leak}N_{c,d} \tag{22.11}$$

where v_{exit} = average air velocity at first-floor exit door, fpm (m/s)
A_{door} = area of opened stairwell door on fire floor, ft^2 (m^2)
$\dot{V}_{o,d}$ = volume flow rate discharged through open stairwell door on floors other than fire floor and first floor, cfm (m^3/s)
$N_{o,d}$ = number of open doors in stairwell other than fire floor and first floor
$\dot{V}_{leak}$ = air leakage through cracks across stairwell wall on floors with closed stairwell doors, cfm (m^3/s)
$N_{c,d}$ = number of closed doors in the stairwell

According to the experimental results in Tamura (1991), for a fire in a building with stairwell pressurization and a smoke exhaust system to exhaust smoke from the fire floor, the critical velocity v_{crit} can be taken as 300 fpm (1.5 m/s) for a fire temperature of 1200°F (650°C) and a mechanical exhaust from the fire floor of 5.5 ach.

In stairwell pressurized systems serving up to 10 floors, $N_{o,d} = 3$, including the door on the fire floor. In systems serving 15 or more floors, $N_{o,d} = 4$, including the door on the fire floor.

For safety, air velocity at the first-floor stairwell exit door v_{exit} can be assumed to be equal to v_{crit}. Discharge air velocity at open stairwell doors on floors other than the fire floor $v_{o,d}$ is less than v_{crit}, in fpm (m/s). This is because the flow resistance of the cracks around doors and windows in floors other than the fire floor is greater than that on the fire floor.

The air volume flow rate discharged through an open stairwell door other than that on the fire floor can be calculated from Eq. (22.10) by using the effective area A_e of the airflow path instead of the area of the door or opening A. Air leakage through cracks in the stairwell wall can also be calculated from Eq. (22.10).

System Pressure Loss for Stairwell Pressurization System

The frictional loss of airflow per floor inside the stairwell pressurization system $\Delta p_{f,s}$, in in. WC (Pa), can be calculated by considering the stairwell as a rectangular duct, as follows:

$$\Delta p_{f,s} = K\left(\frac{1}{5.19}\right)\left(\frac{L}{D_e}\right)\left(\frac{\rho_a}{2g_c}\right)\left(\frac{\dot{V}}{60A_o}\right)^2 \tag{22.12}$$

where L = vertical height of each floor, ft (m)
g_c = dimensional constant, 32.2 $lb_m \cdot ft/lb_f \cdot s^2$

ρ_a = air density, lb/ft^3 (kg/m^3)
$\dot{V}$ = air supply volume flow rate, cfm (m^3/s)
A_o = cross-sectional area of free air passage in stairwell, ft^2 (m^2)

and the ratio R of the area of free flow passage (orifice) to the interior cross-sectional area of the stairwell A_s, in ft^2 (m^2), can be calculated as

$$R = A_o/A_s \tag{22.13}$$

In Eq. (22.12), D_e, in ft (m), indicates the circular equivalent of the stairwell, and it can be calculated as

$$D_e = \frac{4A_s}{P_s} \tag{22.14}$$

where P_s = perimeter of the cross-sectional area of the stairwell, ft (m).

The pressure drop coefficient is K. It is mainly a function of the configuration of the stairwell and the occupant density on the stairs. According to the experimental results of Achakji and Tamura (1988), for a stairwell tested under the following conditions

- With a cross-sectional area of 134 ft^2 (12.5 m^2)
- A floor height of 8.5 ft (2.6 m)
- An occupant density of 0.18 person/ft^2 (2.0 persons/m^2)

the frictional loss of airflow per floor $\Delta p_{f,s}$ has the following values:

	v_o, fpm (m/s)		$\Delta p_{f,s}$, in. WC (Pa)	
Open-tread stair, bottom injection	879	(4.47)	0.128	(32)
	659	(3.35)	0.070	(17.5)
	439	(2.23)	0.032	(8)
Closed-tread stair, bottom injection	1055	(5.36)	0.200	(50)
	848	(4.31)	0.116	(29)
	527	(2.68)	0.05	(12.5)

Here v_o represents the air velocity calculated based on the free flow area, in fpm (m/s). The difference in $\Delta p_{f,s}$ between top and bottom injection is small if other conditions remain the same.

When calculating the system pressure loss of a stairwell pressurization system with multiple injection Δp_{sy}, in. WC (Pa), as shown in Fig. 22.6, is calculated as

$$\Delta p_{sy} = \Delta p_{a-b} + \Delta p_{b-o} \tag{22.15}$$

where Δp_{a-b}, Δp_{b-o} = pressure loss between points a and b and points b and o, respectively, in. WC (Pa) Pressure loss Δp_{b-o} usually varies between 0.10 and 0.40 in. WC (25 and 100 Pa) and is only a small portion of the system pressure. However, it is difficult to calculate accurately because of complicated airflow paths at various operating conditions and the influence of outdoor conditions.

Pressure loss Δp_{a-b} should be calculated between points a and b according to the procedure described in Chap. 17, including the velocity pressure at the supply inlet, and Δp_{b-o} should be estimated as 0.4 in. WC (100 Pa). The calculated system pressure Δp_{sy} should be multiplied by a safety factor 1.2. The actual Δp_{b-o} can be adjusted during acceptance testing.

In a stairwell pressurization system with a bottom single injection, as shown in Fig. 22.8*a*, the maximum pressure difference between the stairwell and outdoors often occurs at the top of the stairwell Δp_{c-o} because the cross section of the stairwell is constant. The system pressure can therefore be calculated as

$$\Delta p_{sy} = \Delta p_{a-b} + \Delta p_{b-c} + \Delta p_{c-o} \tag{22.16}$$

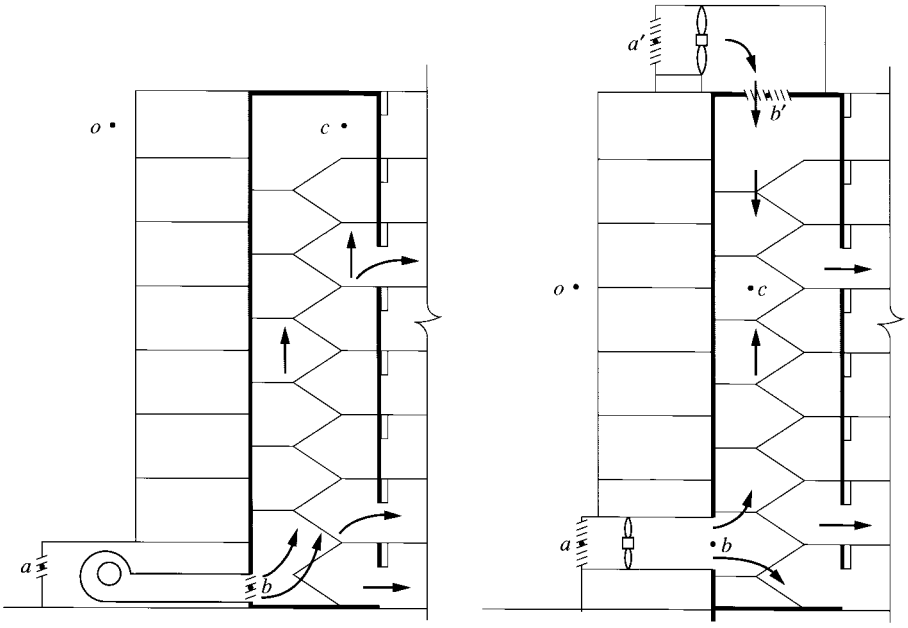


FIGURE 22.8 Bottom injection, and bottom and top injection: (*a*) bottom injection; (*b*) bottom and top injection.

Here, for simplicity, Δp_{c-o} again can be taken as 0.4 in. WC (100 Pa). Then

$$\Delta p_{b-c} = \Delta p_{f,s} N_{\text{fl}} \tag{22.17}$$

where N_{fl} = number of floors. Actually, part of the velocity pressure discharged from the bottom inlet may be converted to static pressure, so that Δp_{b-c} is smaller. However, more data are needed before it can be calculated accurately.

In a stairwell pressurization system with top and bottom injection, as shown in Fig. 22.8*b*, the system pressure loss can be calculated as it is in bottom injection, but Δp_{b-c} may be smaller.

Computer-aided design software for zone smoke control systems are available to assist designers.

Example 22.1. In a smoke control system serving a 20-story high-rise building, each floor has the following construction characteristics:

Floor area	10,000 ft^2 (929 m^2)
Area of the external wall	4800 ft^2 (446 m^2)
Vertical distance between floor and floor	10 ft (3m)
Area of the stairwell wall	600 ft^2 (56 m^2)
Elevator shaft (three elevators) wall area	1000 ft^2 (93 m^2)
Area of stairwell door	$3 \times 7 = 21$ ft^2 (2 m^2)
Volume flow rate of supply air to conditioned space through AHU on each floor	12,000 cfm (5663 L/s)

The pressure difference across the closed stairwell door is 0.1 in. WC (25 Pa). The pressure differences between the stairwell and the outdoors on floors other than the fire floor whose stairwell doors remain open are also 0.1 in. WC (25 Pa), because the pressure drop of the open stairwell door is ignored. Determine the volume flow rates of the zone smoke control and stairwell pressurization systems.

Solution

1. The volume flow rate of the zone smoke exhaust system from the fire floor $\dot{V}_{ex}$ at a rate of 6 ach is

$$\dot{V}_{ex} = 6 \times 10{,}000 \times \frac{10}{60} = 10{,}000 \text{ cfm } (4719 \text{ L/s})$$

The volume flow rate of air supplied to the floors immediately above and below the fire floor is 12,000 cfm (5663 L/s).

2. The volume flow rate of air that is discharged through the open stairwell door to the fire floor $\dot{V}_{fire}$, in cfm (L/s), and is exhausted through the zone smoke control system is

$$\dot{V}_{fire} = v_{crit} A_{door} = 300 \times 3 \times 7 = 6300 \text{ cfm } (2973 \text{ L/s})$$

3. The volume flow rate discharged from the first-floor exit door is assumed to be the same as that from the open stairwell door on the fire floor (6300 cfm, or 2973 L/s).

4. From the given data, the leakage area on the stairwell wall, including cracks around the stairwell door, is

$$A_{leak} = 0.25 + 0.00011 \times 600 = 0.32 \text{ ft}^2$$

If the pressure difference across the closed stairwell door is 0.1 in. WC, from Eq. (22.10), the air leakage rate $\dot{V}_{leak}$ through the cracks of the stairwell wall on each floor whose stairwell door is closed can be calculated:

$$\dot{V}_{leak} = 2610 A(\Delta p)^{1/2} = 2610 \times 0.32(0.1)^{1/2} = 264 \text{ cfm } (125 \text{ L/s})$$

5. It is possible that the fire floor and the pressurized floors immediately above and below are all in the lower 10 floors. Because of the stack effect in this 20-story building, air mixed with hot gas from the burning fire is discharged from the upper 10 stories.

In a 20-story building, assume that there are four open stairwell doors. One is on the fire floor, and another is the first-floor exit door. Then the remaining two opened stairwell doors should each discharge a combined air path that may be a combination of six parallel paths, as shown in Fig. 22.9.

a-b-o
a-c-d-o
a-c-e-f-o
a-g-h-o
a-g-i-j-o
a-k-l-o

6. The leakage area for the external wall for each floor is

$$A_{leak} = 4800 \times 0.00021 \approx 1.0 \text{ ft}^2$$

The leakage area for a floor area of 10,000 ft^2 is

$$A_{leak} = 10{,}000 \times 0.000052 = 0.52 \text{ ft}^2$$

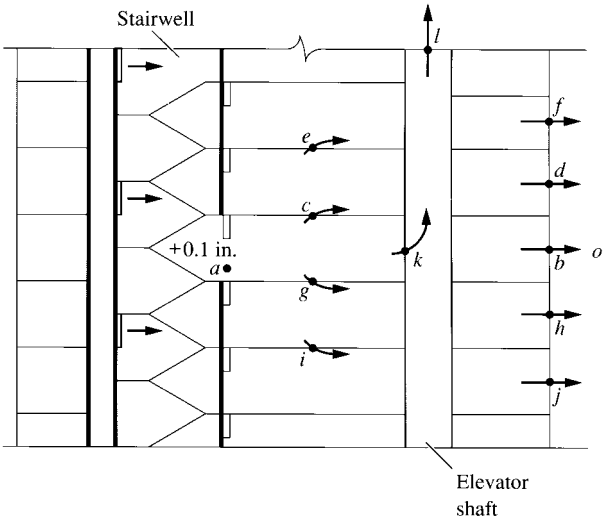


FIGURE 22.9 Air discharged from an open stairwell door through a combined air path to the outdoors.

The leakage area to the elevator shaft is

$$A_{\text{leak}} = 3 \times 0.25 + 1000 \times 0.00084 = 1.59 \text{ ft}^2$$

If the total opening at the top of the elevator shaft is 5 ft^2, assume that only 0.5 ft^2 will be allocated for the open stairwell door. For an airflow path *a-b-o*, if the pressure drop at the open door is negligible, the effective area is

$$A_e = \left(\frac{1}{A_1^2}\right)^{-1/2} = 1 \text{ ft}^2$$

For airflow path *a-c-d-o*, the effective area is

$$A_e = \left(\frac{1}{A_1^2} + \frac{1}{A_2^2}\right)^{-1/2} = \left(\frac{1}{0.52^2} + \frac{1}{1^2}\right)^{-1/2} = 0.46 \text{ ft}^2$$

For airflow path *a-c-e-f-o*, the effective area is

$$A_e = \left(\frac{1}{A_1^2} + \frac{1}{A_2^2} + \frac{1}{A_3^2}\right)^{-1/2}$$

$$= \left(\frac{1}{0.52^2} + \frac{1}{0.52^2} + \frac{1}{1^2}\right)^{-1/2} = 0.35 \text{ ft}^2$$

For airflow path *a-k-l-o*, the effective area is

$$A_e = \left(\frac{1}{1.59^2} + \frac{1}{0.5^2}\right)^{-1/2} = 0.48 \text{ ft}^2$$

The effective area of airflow path *a-g-h-o* is the same as that of *a-c-d-o*, that is, 0.46 ft^2; and the effective area for *a-g-i-j-o* is the same as that of *a-c-e-f-o*, that is, 0.35 ft^2. Therefore, the total

effective area of this combined airflow path is

$$A_e = 2 \times 1 + 2 \times 0.46 + 2 \times 0.35 + 0.48 = 4.1 \text{ ft}^2 \ (0.38 \text{ m}^2)$$

7. From Eq. (22.10), air discharged from this opened stairwell door is

$$\dot{V} = 2610A(\Delta p)^{1/2} = 2610 \times 4.1(0.10)^{1/2} = 3384 \text{ cfm } (1597 \text{ L/s})$$

8. From Eq. (22.11), the total supply volume flow rate of this stairwell pressurization system is then

$$\dot{V}_{s,p} = (v_{\text{crit}} + v_{\text{exit}})A_{\text{door}} + \dot{V}_{o,d}N_{o,d} + \dot{V}_{\text{leak}}N_{c,d}$$
$$= (300 + 300)3 \times 7 + 2 \times 3384 + 264(20 - 4) = 23{,}592 \text{ cfm } (11{,}133 \text{ L/s})$$

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CHAPTER 23
AIR SYSTEMS: MINIMUM VENTILATION AND VAV SYSTEM CONTROLS

23.1 MINIMUM VENTILATION CONTROL 23.2
- Basic Approach 23.2
- Specific Controls in VAV Systems 23.2
- Types of Ventilation Control 23.2
- ASHRAE Standard 62-1999 23.3
- Recirculation of Unused Outdoor Air in Multizone Systems 23.4
- Ventilation Controls for High-Occupancy Areas 23.5

23.2 CO_2-BASED DEMAND-CONTROLLED VENTILATION 23.5
- CO_2-Based Demand-Controlled Ventilation or Time-Based Constant-Volume Control 23.5
- CO_2 Sensor or Mixed-Gases Sensor 23.7
- Location of CO_2 Sensor 23.7
- Substantial Lag Time in Space CO_2 Concentration Dilution Process 23.8
- Base Ventilation and Purge Mode 23.9
- A CO_2-Based Demand-Controlled Ventilation System 23.10
- Application of CO_2-Based Demand-Controlled Ventilation System 23.11

23.3 MIXED PLENUM PRESSURE CONTROLS 23.12
- Basics 23.12
- Case Study: Mixed Plenum Pressure Control Monitoring Plenum Pressure 23.12
- Monitoring Pressure Drop of Louver and Damper Controlling Mixed Plenum Pressure 23.13
- Applications 23.14

23.4 OUTDOOR AIR INJECTION FAN, DIRECT MEASUREMENT, AND FAN TRACKING SYSTEMS 23.14
- Outdoor Air Injection Fan 23.14
- Direct Measurement of Minimum Outdoor Air Intake 23.15
- Fan Tracking Systems 23.15

23.5 CONFERENCE ROOMS 23.16

23.6 SPACE PRESSURIZATION AND RETURN VOLUME FLOW CONTROLS 23.16
- Characteristics of Space Pressure Control 23.16
- VAV Systems Return/Relief Fan Volume Flow Control 23.17

23.7 DISCHARGE AIR TEMPERATURE CONTROLS 23.18
- Basics 23.18
- System Description 23.19
- Operation of Air Economizer and Outdoor Air Intake—Case Study 23.21
- Discharge Air Temperature Reset 23.22

23.8 DUCT STATIC PRESSURE AND FAN CONTROLS 23.23
- Duct Static Pressure Control 23.23
- Set Point of Duct Static Pressure and Sensor's Location 23.24
- Comparison between Adjustable-Frequency Drives and Inlet Vanes 23.24

23.9 FUNCTIONAL CONTROLS FOR VAV SYSTEMS 23.26
- Nighttime Setback, Warm-Up, and Cool-Down Control 23.26
- Steam Humidifier Control 23.27
- Dew Point Control 23.27
- Diagnostics 23.28

23.10 RECOMMENDATIONS AND INTERACTION BETWEEN CONTROLS 23.28
- Recommendations for VAV Controls 23.28
- Interaction between Controls 23.29
- Stability Problems 23.30

23.11 SEQUENCE OF OPERATIONS OF A VAV REHEAT SYSTEM WITH MINIMUM VENTILATION, DISCHARGE AIR TEMPERATURE, AND DUCT STATIC PRESSURE CONTROLS—CASE STUDY 23.30
- HVAC&R System 23.30
- Sequence of Operations 23.30
- Primary Considerations 23.34

REFERENCES 23.35

23.1 MINIMUM VENTILATION CONTROL

Basic Approach

As discussed in Sec. 4.10, indoor air quality (IAQ) is defined as an indication of harmful concentration of indoor air contaminants, such as particulates, combustion products, volatile organic compounds, nicotine, radon, and biological compounds that affects the health of the occupants or the degree of satisfaction of a substantial majority of occupants exposed to such an indoor environment.

Contaminated source control, removal by air cleaners, and using outdoor ventilation air to dilute the concentrations of indoor air contaminants are three basic strategies to improve IAQ. Emissions and odors from the human body, volatile organic compounds, and many gaseous contaminants of 0.003- to 1-μm minute particles can only be removed by expensive, high-efficiency air filters and activated carbon filters. Use of adequate outdoor air to dilute the air contaminants has been proved an essential, practical, and cost-effective means to improve IAQ.

Among the air systems, ventilation controls for constant-volume systems and dedicated ventilation systems can be achieved either by fixing the outdoor and recirculating damper position during the operating period or modulating the fan speed to match the variation of outdoor air supply. Also, during the air economizer cycle, outdoor air intake in the AHU or PU is always greater than the specified amount of outdoor air for ventilation required. However, the reduction in supply volume flow rate at part-load operation of variable-air-volume (VAV) systems, a tighter building shell, and the ineffective ventilation control may reduce the amount of outdoor air intake to less than the requirement for ventilation. Minimum ventilation control (control in minimum outdoor air recirculating mode) becomes one of the critical problems in VAV systems in commercial buildings. As discussed in Sec. 4.10, ventilation means supplying and removing ventilation air, and ventilation air consists of outdoor air plus treated recirculated air for acceptable indoor air quality.

Specific Controls in VAV Systems

In a multizone VAV system, zone temperature control directly affects the thermal comfort of the occupants and is the basis of the VAV system control. During the minimum ventilation control, only the amount of minimum outdoor air supply to a conditioned space is controlled; the space airflow balance and space pressurization vary accordingly. There are interactions between ventilation and space pressurization controls.

Zone temperature control is achieved by modulating the volume flow rate supplied to each control zone. As the variation of zone load causes the change of damper positions and the airflow passages as well as the zone volume flow in VAV boxes, the duct static pressure in the main supply duct may rise or fall. Duct static pressure control then modulates the supply fan speed and thus the system supply volume flow rate and fan total pressure to match the variation in zone loads.

Discharge air temperature control adjusts the cooling or heating capacity of a VAV system to match the system load. It also resets the discharge air temperature and optimizes energy use. Therefore, specific controls for a VAV system include

- Zone temperature controls
- Minimum ventilation control and space pressurization control
- Discharge air temperature control, including economizer, cooling capacity, mixed air, and heating capacity controls in sequence, and low-temperature limit control
- Duct static pressure control

Types of Ventilation Control

In VAV systems, ventilation control includes air economizer control and minimum outdoor ventilation air control during the recirculation mode. During the air economizer cycle including mixed air

control, a volume flow of outdoor air that exceeds the amount of minimum outdoor ventilation air is always supplied to the conditioned space. Ventilation problems often occur during minimum outdoor air recirculating mode, i.e., minimum ventilation controls. Currently used minimum ventilation controls in VAV systems include the following:

CO_2-based demand-controlled ventilation
Mixed plenum pressure controls
Outdoor air injection fan control
Fan tracking control

ASHRAE Standard 62-1999

As discussed in Sec. 4.10, ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality*, specifies two alternative procedures to obtain acceptable IAQ:

1. *Ventilation rate procedure*. In the ventilation rate procedure, acceptable indoor air quality is achieved by providing ventilation air of specified quality and quantity to the conditioned space. This procedure prescribes the following:

- The quality of the outdoor air shall be acceptable for ventilation, i.e., contaminants in outdoor air do not exceed the concentrations listed in the Table of National Primary Ambient-Air Quality Standards by EPA in Sec. 4.10.
- Outdoor air treatments are required when necessary.
- Ventilation rates for typical spaces are listed in Table 4.5. For details, refer to ASHRAE Standard 62-1999.
- Ventilation systems for spaces with intermittent or variable occupancy may have their outdoor air quantity adjusted by damper or cycling of fan operation to provide sufficient dilution to maintain contaminant concentrations within acceptable levels at all times.
- For intermittent occupancy, when contaminants are associated only with occupants or occupant activities, do not result a health hazard because contaminants are dissipated during unoccupied periods.
- When contaminants are independent of occupants or occupant acivities, the supply of outdoor air should lead occupancy lag or lead time depending on the ventilation rate and air capacity per person in the space.
- Where peak occupancies are less than 3 h in duration, the outdoor airflow rate may be determined according to average occupancy for the duration of operation of the system provided the average occupancy used is not less than half of the maximum.

2. *Indoor air quality procedure*. In the indoor air quality procedure, acceptable air quality is achieved within the conditioned space by restricting the concentration of all known contaminants of concern to some specified acceptable levels. The acceptable concentration level in Table of National Primary Ambient-Air Quality Standards, as listed in Sec. 4.10, also applies indoors for the same exposure time. The following limits apply for four other indoor contaminants according to Standard 62-1999:

Contaminant	Concentration	ppm	Exposure time
Carbon dioxide CO_2	1.8 g/m^3	1000	Continuous
Chlordane	5 μg/m^3	0.0003	Continuous
Ozone	100 μg/m^3	0.05	Continuous
Radon	4 pCi/L		Annual average

The new Standard 62-1999 contains the entire 1989 version, unchanged, along with the following four new addenda:

- Addenda 62e removes the statement that ventilation rates in Table 2 accommodate a moderate amount of smoking because of the dangerousness of second-hand tobacco smoke.
- Addenda 62c removes the requirement that the temperature and humidity conditions specified in Standard 55 be maintained in all ventilated space, such as even in garages.
- Addenda 62d rewords the standard scope to state that compliance with the standard will not ensure acceptable indoor air quality.
- Addenda 62f clarifies that carbon dioxide (CO_2) is simply a useful indicator of the concentration of human bioeffluents.

For details, refer to ASHRAE Standard 62-1999. Standard 62-1999 is under continuous maintenance by a Standing Standard Project Committee for which the Standards Committee has established a documented program for regular publication of addenda or revisions.

Recirculation of Unused Outdoor Air in Multizone Systems

For a multizone VAV system, the ratio of outdoor air to supply air required to meet the ventilation and thermal load requirements is often different from zone to zone. The system ventilation air required, in most cases, is actually the amount of outdoor air intake at the AHU or PU. The problem is how the system-fixed ratio of outdoor air to supply air can satisfy the different ratios of outdoor air to supply air in various control zones.

In a specific control zone, if the ratio of outdoor air to supply air is higher than is needed in this zone, the result is an increase in unused outdoor air in the recirculating air (more truly, the concentration of air contaminants is lower) which raises the dilution power of the supply air (including outdoor and recirculating air) for other control zones.

Methods of calculating ventilation requirements using the unused outdoor air contained in recirculating air were developed in Australia and are included in ASHRAE Standard 62-1999. The required corrected fraction of outdoor air supply in system supply air Y, in cfm (L/s), considering the outdoor requirement of the critical zone and the unused outdoor air contained in the recirculating air can be calculated as

$$Y = \frac{X}{1 + X - Z} \tag{23.1}$$

and

$$\begin{aligned} Y &= \dot{V}_{o,\text{cor}}/\dot{V}_s \\ X &= \dot{V}_{o,\text{sys}}/\dot{V}_s \\ Z &= \dot{V}_{o,\text{cr}}/\dot{V}_{s,\text{cr}} \end{aligned} \tag{23.2}$$

where $\dot{V}_s$ = supply volume flow rate of air system, cfm (L/s)
$\dot{V}_{o,\text{cor}}$ = corrected outdoor air supply volume flow rate, considering critical zone and unused outdoor air in the recirculating air, cfm (L/s)
$\dot{V}_{o,\text{sys}}$ = calculated system outdoor air volume flow rate, cfm (L/s)
$\dot{V}_{o,\text{cr}}$ = outdoor air volume flow rate required in critical zone, cfm (L/s)
$\dot{V}_{s,\text{cr}}$ = critical zone supply volume flow rate, cfm (L/s)
X = uncorrected fraction of outdoor air supply
Y = corrected fraction of outdoor air supply
Z = fraction of outdoor air in supply air in critical zone

The system outdoor air volume flow rate can be evaluated as

$$\dot{V}_{o,\text{sys}} = N_{oc}\dot{V}_{o,\text{req}} \tag{23.3}$$

where N_{oc} = number of occupants in conditioned area served by air system
$\dot{V}_{o,\text{req}}$ = ventilation air requirements as specified by ASHRAE Standard 62-1999, cfm/person (L/s · person)

Outdoor air volume flow rates $\dot{V}_{o,\text{cor}}$, $\dot{V}_{o,\text{sys}}$, and $\dot{V}_{o,\text{cr}}$ and supply volume flow rates $\dot{V}_s$, and $\dot{V}_{s,\text{cr}}$ all must occur at the same time.

Ventilation Controls for High-Occupancy Areas

ASHRAE/IESNA Standard 90.1-1999 mandates that air systems with design outdoor ventilation air intake greater than 3000 cfm (1420 L/s) serving areas having a design occupancy density exceeding 100 people per 1000 ft^2 (93 m^2) shall include means which automatically reduce outdoor air intake below design volume flow rate when spaces are partially occupied. Outdoor ventilation air control shall be in compliance with ASHRAE Standard 62 and local codes except for systems with heat recovery.

23.2 CO_2-BASED DEMAND-CONTROLLED VENTILATION

CO_2-Based Demand-Controlled Ventilation or Time-Based Constant-Volume Control

As discussed in Sec. 4.10, among the indoor contaminants, particulates should be removed by medium- and high-efficiency filters; vented combustion products including CO do not normally exceed the allowable indoor concentration levels in a conditioned space; volatile organic compounds including formaldehyde may have above-allowable indoor concentration levels when the building is new or after remodeling; smoking is prohibited in many public places and is often limited in a specific area in many buildings; only 6 percent of U.S. homes in their annual average radon concentration exceeds the EPA specified level and needs subslab depression system and others; and airborne virus may become dangerous when any one of the occupants is sick and carries such a virus. Under many circumstances, occupant-generated CO_2, as a useful indicator of the concentration of body bioeffluents, emissions, and related contaminants, may be used as one of the indicators of poor indoor air quality.

Many buildings, such as auditoriums, assembly halls, airport terminals, retail stores, offices, and classrooms, have a variable occupancy during working hours and thus a fluctuation of the amount of indoor contaminants released. Figure 23.1 shows the CO_2 concentration profiles measured in the floor space and in the return air duct on the 18th floor with a 20 percent fixed outdoor air supply of a 22-story building in Ottawa, Canada. On a weekday, the concentration of CO_2 gradually rose from 400 ppm in the morning to about 800 ppm before the office workers went out for lunch. After lunch, the concentration of CO_2 fluctuated and reached a daily maximum at about 3 p.m. Then the CO_2 concentration dropped gradually to 450 ppm at 8 p.m. If the amount of ventilation air required to dilute the concentration level of the indoor contaminants varied accordingly, energy can be saved. More than the required amount of outdoor ventilation air intake means more than the required amount of energy is used to cool and dehumidify the outdoor air during summer and to heat it during winter.

Demand-controlled ventilation (DCV) means that when the number of occupants (occupant density) in the conditioned space during the occupied period, and thus the demand, drops, less outdoor air is taken into the AHU or PU and supplied to the conditioned space to meet the demand of dilution of concentration levels of contaminants with nearly the same specified ventilation rate

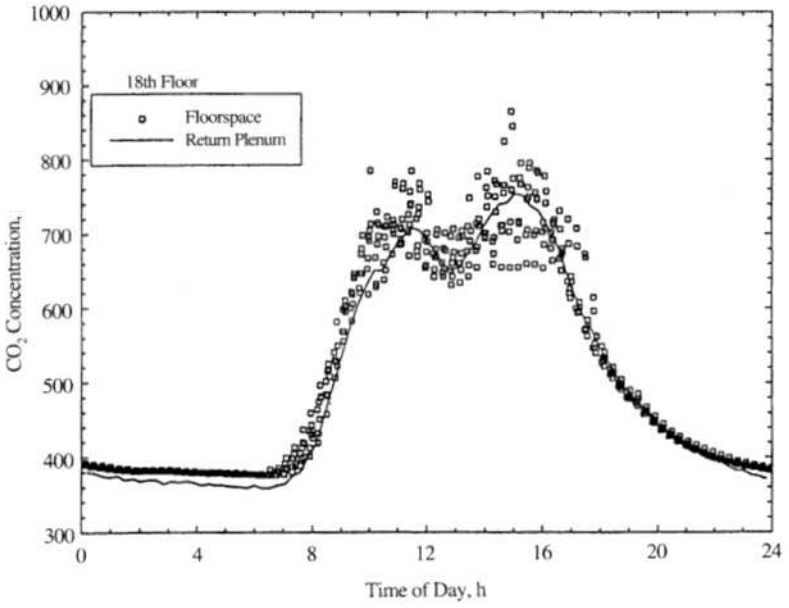


FIGURE 23.1 CO_2 concentration profiles measured in the floor space and in the return air duct on the 18th floor with minimum 20 percent outdoor air supply. *Source: Reardon et al., ASHRAE Transactions, Part II, 1994*

cfm/person (L/s·person). Time-based constant-volume ventilation control intends to supply constant-volume conditioned outdoor air to the conditioned space for dilution during the occupied period even if the occupant density drops.

Compared with the time-based constant-volume ventilation control, demand-controlled ventilation has the following advantages:

- DCV supplies the demanded amount of outdoor air to the conditioned space to maintain an acceptable air quality and, at the same time, saves heating and cooling energy to condition the excessive amount of supplied outdoor air in the time-based constant-volume ventilation control. According to the simulation results in Carpenter (1996), compared to a system without DCV, 1000-ppm CO_2-based DCV reduces 25 to 30 percent of space heating energy use. With 1000-ppm CO_2-based demand-controlled ventilation, the average ventilation rate is 0.1 to 0.3 ach (air changes per hour), 10 to 30 percent lower than that of the constant-volume ventilation control.
- DCV actually has taken into account the dilution power of the infiltrated outdoor air, which results in a greater energy saving.
- DCV often uses CO_2 or a group of mixed gaseous contaminants as an indicator (controlled variable). The space concentration of CO_2 itself provides feedback of the control system, whereas in a constant-volume ventilation control system, the volume flow rate of the outdoor air is difficult to measure exactly.

In a VAV system, DCV is normally activated only in minimum outdoor air recirculating mode. During the air economizer cycle, the amount of outdoor air intake in the AHU or PU is always greater than the system outdoor air volume flow rate $\dot{V}_{o,sys}$. The amount of outdoor air intake is determined by the free-cooling requirement of the VAV system.

CO_2 Sensor or Mixed-Gases Sensor

In DCV systems, two kinds of air sensors (air transmitter) are most often used as an indicator: CO_2 sensors and a volatile organic compound (VOC, or mixed-gases) sensor, as discussed in Sec. 5.4. Compared to a CO_2 sensor, a mixed-gases sensor senses a variety of mixed gases, is less expensive, and needs less maintenance. However, according to Stonier (1995), a CO_2 sensor has the following advantages:

- The concentration level of CO_2 directly indicates the concentration of occupant-generated bioeffluents in the occupied zone.
- The concentration of CO_2 of outdoor air is usually rather steady, between 300 to 350 ppm in many locations in the United States. If each occupant exhales 0.0106 cfm (0.3 L/min) of CO_2, from the balance of CO_2 by using Eq. (4.26), the ventilation rate supplied to the occupied zone can be roughly estimated. Or if the ventilation rate is known, then the number of occupants can be approximately estimated.
- If the ventilation rate of outdoor air supplied to a specific control zone and its number of occupants are known, then a higher-than-normal concentration level of CO_2 may be due to a poor ventilation effectiveness, because of the short circuit of supply air or improper location of the CO_2 sensor.

ASHRAE Standard 62-1999 recommends that "Where only dilution ventilation is used to control indoor air quality, an indoor to outdoor differential concentration not greater than about 700 ppm of CO_2 indicates comfort (odor) criteria related to body bioeffluents are likely to be satisfied." Standard 62-1999 also points out that using CO_2 as an indicator of bioeffluents does not eliminate the need for consideration of other contaminants.

Steven Taylor, chair of the committee revising Standard 62-1989, was asked in IAQ and Energy '98 New Orleans, October 24 to 27, if CO_2 is an appropriate indicator of indoor air quality. His response: "The use of CO_2 as a reliable indicator of IAQ has been overblown, . . . CO_2 is not itself a contaminant of concern at the concentrations found in most buildings. It can provide an indication of the amount of outdoor air being supplied to the space per person, but it cannot be used to accurately determine that rate unless the space is in nearly steady-state conditions. In an office, it typically takes several hours to reach steady-state conditions, and people in most offices do not stay in one place that long."

Although CO_2 is not itself an indoor contaminant or is a comprehensive indicator of IAQ, the indoor CO_2 concentration can provide an indication of outdoor air supplied to a space per person. Reardon (1994) and many other field measurements and investigations had clearly shown

- The relationship between the concentration of space CO_2 and the air change rate (as shown in Fig. 23.2)
- The floor-space concentration profiles of CO_2 or average concentration profiles of CO_2 in a return duct (as shown in Fig. 23.1)

The use of a CO_2 sensor to measure the space CO_2 concentration to control the amount of outdoor air intake in a ventilation control system is often an appropriate choice unless there is a still better sensor.

As the outdoor ventilation air dilution is the primary issue to improve IAQ, the space concentration of CO_2, to a certain degree, most often is an indirect indicator of IAQ.

Location of CO_2 Sensor

Test results of Reardon (1994) showed that the CO_2 concentrations measured in the occupied floor spaces are more variable than data measured in the openings of the return air shaft on the same floor. The concentration of CO_2 of return air fairly well represents the space average CO_2 concentration

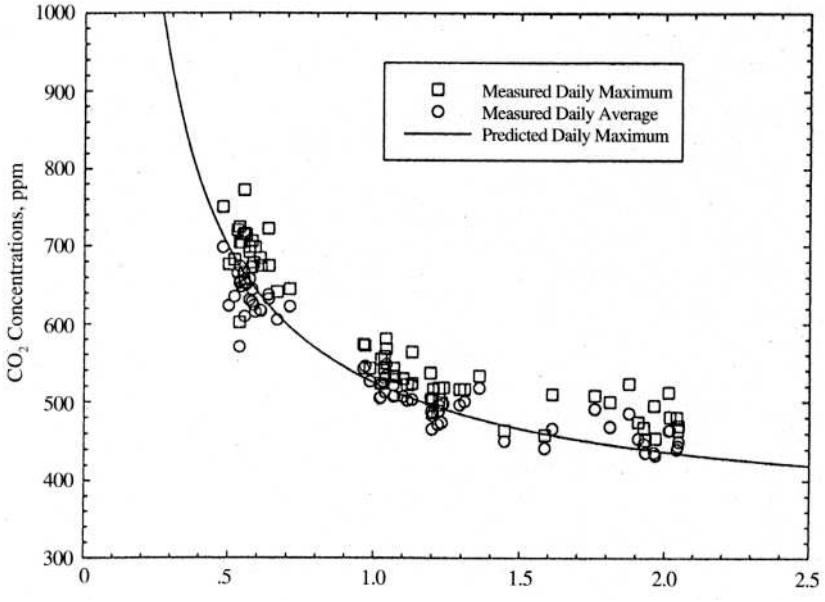


FIGURE 23.2 Measured daily peak and daily average CO_2 concentrations in the return duct versus air changes per hour. (*Source: Reardon et al., ASHRAE Transactions, Part II, 1994*)

data measured of that particular floor after air mixes in the return air plenum. On the other hand, the floor space concentration data were consistently higher than the return air data. This may be due to some degree of short-circuiting of outdoor air supply to the occupied zone in that floor.

Emmerich and Persily (1997) noted that air movements due to open doors or open windows cause differences and instabilities in CO_2 concentrations. Enermodal Engineering (1995) reported that an air system with CO_2 sensors in the return duct on each floor showed no significant difference in IAQ compared to a system with a CO_2 sensor in the central return. Central control with a set point of 800 ppm provided similar performance but at a much reduced installation cost compared to individual zone control with a set point of 1000 ppm. Ruud et al. (1991) found that concentrations of CO_2 measured at the wall and in the exhaust air were nearly identical with the wall-mounted sensor having an additional 2-min delay compared to that located in the exhaust air.

Therefore, for most VAV systems, the suitable location to install a CO_2 sensor lies in the return main duct (or return air intake) after the return ceiling plenum and close to the AHU or PU for sensing an average CO_2 concentration. The operator should check the difference in CO_2 concentrations between the floor space and the return duct periodically during operation so that the CO_2 set point can be properly adjusted; e.g., the set point in the return duct is 100 to 200 ppm lower than the 1000-ppm target concentration level. In a VAV system, a single centrally-controlled sensor is recommended.

For a dedicated ventilation and space recirculating system, a better location of a CO_2 sensor is in the exhaust air main duct if there is an exhaust air system for the ventilation air supply. A critical zone or a representative floor space is also a candidate for the location of a CO_2 sensor.

Substantial Lag Time in Space CO_2 Concentration Dilution Process

When a certain amount of outdoor air is extracted into an AHU or a PU, it should thoroughly mix with the recirculating air first and form an outdoor-recirculating mixture. The mixture is then

supplied to a control zone in a VAV system, where it needs again to mix thoroughly with the space air in that control zone and to distribute over the entire conditioned space, in order to dilute the concentration of occupant-generated and other contaminants. It takes a substantial lag time to perform such a mixing and dilution process.

Elovitz (1995) reported that there is a substantial lag time in the space CO_2 concentration dilution process. If CO_2 rises, an increase in the outdoor air ventilation rate will not make it drop instantly. The control continues calling for more and more dilution and opens the outdoor damper wider and wider, until the drop in the CO_2 concentration is sensed by the CO_2. Recently a CO_2 minimum ventilation control system in an insurance office building retrofit showed such a phenomenon. As the concentration of CO_2 raised above the set point, the control system kept cranking the system up to 100 percent outdoor air intake until eventually the dilution effect showed up in the return air; at that point, the control cranked the outdoor damper down until it was fully closed, as the concentration of CO_2 continued to fall long after the damper started to close.

The following are two methods that will solve the time-lag problem:

1. Determine the relationship between the sensed CO_2 concentration and the required air changes per hour of minimum outdoor air intake expressed in minimum outdoor damper setting in the AHU or PU, as shown in Fig. 23.2 during commissioning. For each sensed CO_2 concentration in the return duct or floor space, the DDC controller resets the required minimum outdoor air intake, in air changes per hour, as shown in Fig. 23.2, and the corresponding outdoor damper setting according to the determined relationship between them during commissioning.
2. When the sensed CO_2 concentration exceeds the set point, the outdoor damper will open an additional 5 percent. If, after a prespecified time interval, the sensed CO_2 level still exceeds the set point, the outdoor damper will open an additional 5 percent until the outdoor air intake in the AHU or PU reaches the upper limit $\dot{V}_{o,\text{sys}}$. The prespecified time interval should be determined during commissioning.

Base Ventilation and Purge Mode

In a CO_2-based demand-controlled ventilation system serving a typical office building, even if the number of occupants drops to zero during the occupied period, the outdoor air intake should not drop below a base ventilation. Base ventilation $\dot{V}_{\text{bv}}$, in cfm (L/s), is the lower limit of minimum outdoor ventilation air required to dilute mainly the nonoccupant-generated air contaminants, such as building-generated volatile organic compounds (VOCs), to provide exhaust air in restrooms, and to maintain a positive space pressurization. For a typical office building, the amount of base ventilation required depends on interior building materials, building construction, and the tightness of the building shell. The exhaust air in restrooms is often provided by the transfer air that also dilutes the building-generated VOCs. The base ventilation $\dot{V}_{\text{bv}}$ is the sum of the outdoor air required for restroom exhaust and exfiltration to maintain a space pressurization. Consider a one-story office building of 20,000 ft^2 (1859 m^2) of floor area with an exterior wall area of 10,000 ft^2 (929 m^2):

- Two bathrooms have typically 12 water closets or urinals. According to ASHRAE Standard 62-1999, these two restrooms need about $50 \times 12 = 600$ cfm or 0.03 cfm/ft^2 (0.15 L/s · m^2).
- As discussed in Sec. 20.2, the flow coefficient of an average leaky building is 1.0 cfm/ft^2 (5.0 L/s · m^2) of exterior wall at 0.3 in. WC (75 Pa). From Eq. (20.2), the overall leakage rate is

$$\dot{V}_{\text{leak}} = C_{\text{flow}} A_w (\Delta p)^{0.65} = 1.0(10{,}000)(0.3)^{0.65} = 4572 \text{ cfm } (2157 \text{ L/s})$$

From Eq. (20.1) the effective leakage area of this average leaky building is:

$$A_{e,l} = -\frac{\dot{V}_{\text{inf}}}{4005\sqrt{\Delta p}} = \frac{4572}{4005\sqrt{0.3}} = 2.08 \text{ ft}^2 \text{ (0.194 m}^2\text{)}$$

Under base ventilation, the amount of exfiltration required to maintain a space positive pressure of + 0.005 in. WG (1.25 Pa) can be calculated as

$$-\dot{V}_{\text{inf}} = 4005 A_{e,l}\sqrt{\Delta p} = 4005 \times 2.08\sqrt{0.005} = 589 \text{ cfm } (278 \text{ L/s})$$

The volume flow rate of base ventilation is then

$$\dot{V}_{\text{bv}} = 600 + 589 = 1189 \text{ cfm}$$

or roughly 1200 cfm (566 L/s), that is, 1200/20,000 = 0.06 cfm/ft^2 (0.30 L/s·m^2) of floor area, which is about 43 percent of the value of the minimum outdoor air required ($\dot{V}_{bv} \approx 0.43\ \dot{V}_{o,\text{sys}}$).

Formaldehyde and other possible indoor air contaminants in the space should be measured. If their concentrations are above allowable levels, or if the building shell is too leaky, the base ventilation should be increased accordingly.

The amount of outdoor ventilation air provided for CO_2-based demand-controlled ventilation in a VAV system in an office building with a ceiling height of 9 ft (2.7 m) during various operating modes is as follows:

	cfm/ft^2 (L/s·m^2) of floor area	ach
Air economizer cycle, free-cooling mode	0.14 – 1.3 (0.7 – 6.5)	0.93 – 10
Minimum outdoor air recirculating mode $\dot{V}_{o,\text{sys}}$ (at design occupancy)	0.14 (0.7)	0.93
Base ventilation $\dot{V}_{\text{bv}}$ (lower limit)	0.06 (0.25)	0.40

The volume flow rate of minimum outdoor air during recirculating mode at design occupancy is the calculated system outdoor air intake in the AHU or PU at summer design conditions $\dot{V}_{\text{o,sys}}$ by Eq. (23.3).

It is recommended that a purge operation take place every weekday morning if the concentration of the volatile organic compounds or other indoor contaminants is above the allowable level before the building is occupied or if the difference between the indoor and outdoor CO_2 concentrations is greater than 200 or 300 ppm.

A CO_2-Based Demand-Controlled Ventilation System

Figure 23.3 shows a CO_2-based demand-controlled ventilation system in a VAV system. It consists of a CO_2 sensor, a DDC unit controller, a minimum outdoor damper, an economizer damper, a recirculating damper, an exhaust damper, and damper motors. The CO_2 sensor is located in the return duct that connects the return ceiling plenum and the return fan inlet. During minimum outdoor air recirculating mode:

- The supply fan and the minimum outdoor air damper of this VAV system must have the capacity to provide the required amount of outdoor air at a volume flow rate of $\dot{V}_{o,\text{sys}}$, in cfm (L/s), calculated by Eq. (23.3).
- Outdoor air should be supplied to the occupied zone with a high ventilation effectiveness for the dilution of the concentration of CO_2 and other indoor contaminants.
- During commissioning, damper settings of required minimum outdoor air at design occupancy $\dot{V}_{o,\text{sys}}$, and base ventilation $\dot{V}_{\text{bv}}$ should be clearly marked, and the relationship between the measured CO_2 concentration and the minimum outdoor damper opening setting from $\dot{V}_{\text{bv}}$ up to $\dot{V}_{o,\text{sys}}$ should be determined after tests.
- When a VAV system is in the occupied, minimum outdoor air recirculating mode, after supply and return fans start, the minimum outdoor damper should be opened to an opening setting of base ventilation.

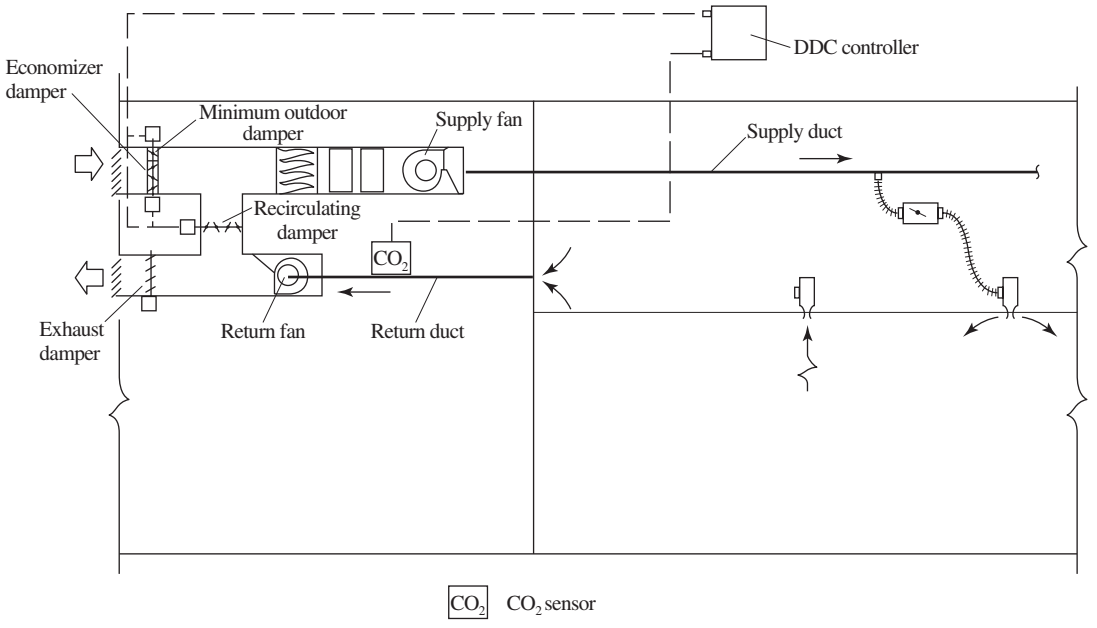


FIGURE 23.3 A CO_2-based demand-controlled ventilation control in a VAV system.

- If the signal of CO_2 concentration sensed by the CO_2 sensor located in the return duct or a representative floor space is sent to the DDC controller, the DDC controller resets the minimum outdoor air intake $\dot{V}_{o,\text{sys}}$ to a corresponding minimum outdoor damper opening setting according to the relationship between the measured return duct or space CO_2 concentration and the damper opening setting.

  If the required amount of minimum outdoor air corresponding to the sensed CO_2 level is less than the base ventilation, the DDC controller resets the minimum outdoor air intake down to the base ventilation $\dot{V}_{\text{bv}}$.
- The recirculating damper is opened 100 percent.
- When the VAV system is in the off position, the minimum outdoor, economizer, and exhaust dampers are closed, and the recirculating damper is opened fully.

During 100 percent outdoor air free-cooling mode, the air economizer control overrides the CO_2-based DCV control.

Application of CO_2-Based Demand-Controlled Ventilation System

Chen and Demster (1996) and Emmerich and Persily (1997) recommended the use of CO_2-based DCV systems under the following circumstances:

- The existence of high occupancy density with considerable fluctuations in occupancy
- Low pollutant emissions from nonoccupant-generated sources
- In location with climate where heating or refrigeration equipment has a significant number of working hours annually

For a multizone VAV system that serves control zones with an obvious occupancy variation schedule, such as in an office building where offices and conference rooms use a common AHU or PU, when many occupants in the offices may go out for lunch, but a meeting is still taking place in a conference room, remedies should be considered during design to supply additional outdoor air to the conference room under such circumstances.

23.3 MIXED PLENUM PRESSURE CONTROLS

Basics

As discussed in Secs. 22.5 and 22.6, for VAV systems operating at minimum outdoor air recirculating mode, when the supply volume flow reduces during part-load operation, the static pressure in the mixed plenum (mixing box) where outdoor air mixes with the recirculating air tends to rise and causes a small pressure difference between the mixed plenum and outdoors. The minimum outdoor air intake in the AHU or PU reduces accordingly. Mixed plenum pressure control aims to maintain a nearly constant static pressure in the mixed plenum and thus keeps a nearly constant pressure difference between the mixed plenum and outdoors to extract a nearly constant amount of minimum outdoor air in both design load and part-load operation.

Case Study: Mixed Plenum Pressure Control Monitoring Plenum Pressure

Graves (1995) developed a mixed plenum pressure control system in a VAV system by monitoring the pressure difference between the mixed plenum pressure and outdoors atmospheric pressure and modulating the recirculating damper opening to maintain a nearly constant mixed plenum pressure, as shown in Fig. 23.4*a*.

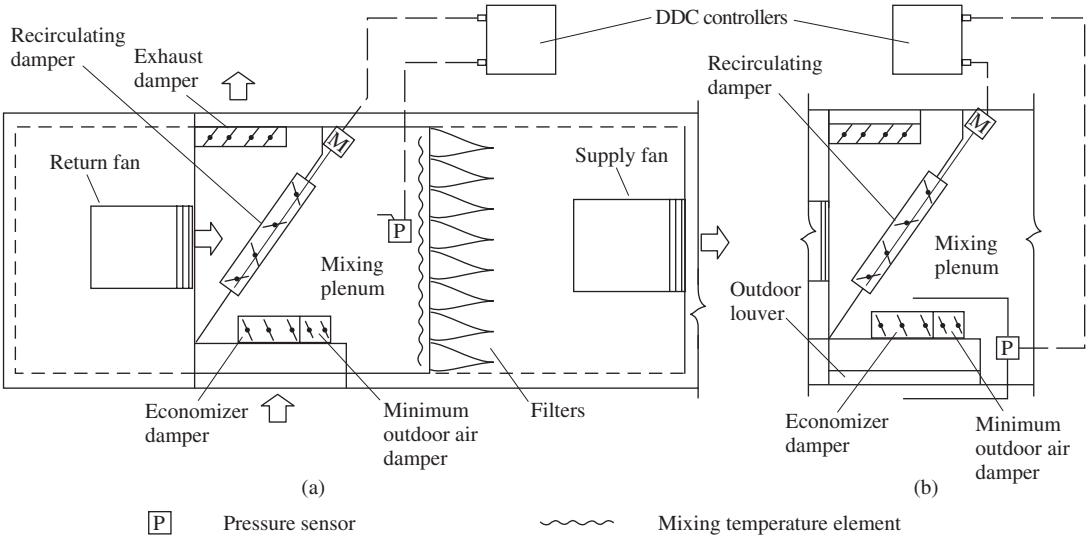


FIGURE 23.4 Mixed plenum pressure controls: (*a*) monitoring mixed plenum–outdoors pressure differential; (*b*) monitoring pressure drop across the outdoor damper and louver.

System Description. The VAV system includes two rooftop packaged units. Each has a capacity of 8300 cfm (3917 L/s) with a variable-speed supply fan and a variable-speed return fan. A fixed minimum oudoor damper supplies 1800 cfm (850 L/s) of outdoor air to the conditioned space, and an economizer damper supplies the additional outdoor air during the air economizer cycle. When the outdoor temperature is higher than 70°F (21.1°C), the VAV system is operated in minimum outdoor air recirculating mode. A DDC unit closes the economizer damper and opens the recirculating damper. Whenever the packaged unit is operating, the exhaust damper is opened 100 percent.

In the mixed plenum, outdoor air is drawn through a louver and damper on one side of the unit, and exhaust air is discharged through the exhaust damper and louver on the opposite side. The opening of the fixed minimum outdoor damper was set to provide 1800 cfm (3917 L/s) with a plenum pressure of − 0.50 in. WG (− 125 Pag). At design conditions, the airflow and pressure drops across dampers and louvers are as follows:

	Airflow, cfm (L/s)	Δp, in. WC (Pa)
Recirculating damper	6500 (3067)	0.55 (137)
Minimum outdoor damper	1800 (850)	0.30 (75)
Economizer damper	6500 (3067)	0.30 (75)
Exhaust damper	6500 (3067)	0.10 (25)
Outdoor air louver	8300 (3917)	0.25 (63)
Exhaust air louver	6500 (3067)	0.25 (63)

Minimum Ventilation. A DDC control system is used for the controls of this rooftop unit including a mixed plenum pressure control. A static pressure sensor located in the mixed plenum senses the plenum pressure, and a shielded outdoor air pressure reference is located above the rooftop packaged unit. The DDC system modulates the recirculating damper to maintain a static pressure difference between the mixed plenum pressure and outdoors $\Delta p_{m,p}$ of 0.55 in. WC (137 Pa) ± 0.05 in. WC (12.5 Pa). If $\Delta p_{m,p}$ is too large, the recirculating damper is gradually opened until it reaches the set point. If $\Delta p_{m,p}$ is too small, the recirculating damper is gradually closed until the set point is maintained. According to field-measured results, the DDC mixed plenum pressure control system has achieved a very stable control of $\Delta p_{m,p.}$

Supply and Return Fans. The speed of the supply fan is modulated to maintain a duct static pressure of 1.50 in. WG (375 Pa) at the location of the duct static pressure sensor in the supply main duct during both design and part-load operations. The volume flow rate of the return fan is always approximately 1800 cfm (850 L/s) less than the supply air volume flow. A linear relationship between the supply fan speed and return fan speed was determined by testing the rooftop packaged VAV system during commissioning. Roughly, when the supply fan was at 100 percent fan speed, the return fan was at 90 percent fan speed. When the supply fan was operated at approximately 40 percent speed, the recirculating damper was closed and the return fan was turned off.

Actual operation of this system showed that the supply fan speed was always above 40 percent because the minimum supply volume flow rate required by the VAV boxes was 35 percent, or about 2900 cfm (1369 L/s).

Monitoring Pressure Drop of Louver and Damper Controlling Mixed Plenum Pressure

Mumma and Wong (1990) first introduced a kind of mixed plenum pressure control to modulate the recirculating damper to maintain a predetermined pressure drop across an outdoor air louver and damper combination, as shown in Fig. 23.4*b*. This mixed plenum pressure control consists of a minimum outdoor damper, an economizer damper, a recirculating damper, a pressure sensor, a DDC controller, damper motors, a variable-speed supply fan, and a variable-speed relief fan. It proceeds as follows:

- The minimum outdoor damper is fixed at an opening setting which delivers minimum outdoor air $\dot{V}_{o,\text{sys}}$. This damper setting is determined during commissioning.
- When the supply fan reduces its speed at part-load operation, the pressure differential across the outdoor louver and minimum outdoor damper combination drops. As this reduction is sensed by the pressure transducer, a DDC unit closes the recirculating damper to maintain a preset pressure drop across the combination, and therefore ensures that the amount of minimum outdoor air $\dot{V}_{o,\text{sys}}$ intake is always extracted.
- If the supply fan increases its speed, the pressure drop across the outdoor louver-damper combination rises. The DDC controller opens the recirculating damper to reduce the pressure drop across the combination and maintains the required amount of minimum outdoor air $\dot{V}_{o,\text{sys}}$ intake.

Applications

In VAV systems, mixed plenum pressure controls ensure time-based constant volume of minimum outdoor ventilation air supply to the conditioned space. The controls are simple and effective and can be used for buildings where there is less variation in occupancy.

23.4 OUTDOOR AIR INJECTION FAN, DIRECT MEASUREMENT, AND FAN TRACKING SYSTEMS

Outdoor Air Injection Fan

To provide a guaranteed amount of minimum outdoor air $\dot{V}_{o,\text{sys}}$ to a VAV system, recently an outdoor air injection fan has sometimes been installed in parallel to the outdoor air passage leading to the main supply fan, as shown in Fig. 23.5. An outdoor air injection fan has the following operating charateristics:

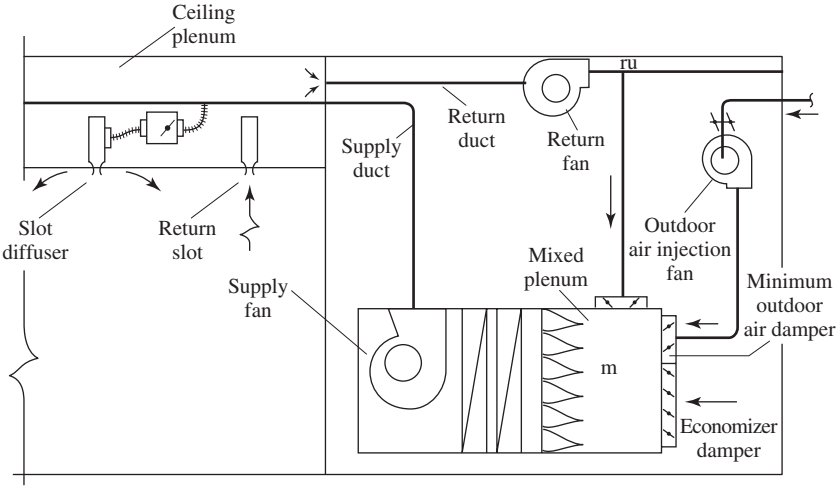


FIGURE 23.5 A VAV system with an outdoor air injection fan.

- The maximum supply volume flow rate of this outdoor air injection fan should be equal to the amount of minimum outdoor air $\dot{V}_{o,\text{sys}}$ required. The fan total pressure of this injection fan is often equal to the total pressure loss between the outdoor air intake and the mixed plenum inlet.
- The ductwork of the injection fan system is sized so that the velocity pressure is easily measured.
- An airflow measuring station with a velocity sensor sends signals to a DDC controller and adjusts the fan speed as well as its volume flow through a variable-speed drive to correct for wind and backpressure effects.
- The injection fan system is a constant-volume minimum outdoor air supply system and operates whenever outdoor air is required.
- According to Avery (1996), if an outdoor air injection fan is installed with a supply and return fan combination, the mixed plenum may be varied from negative pressure when the VAV system is operating on 100 percent outdoor air or mixed air control to positive pressure during minimum outdoor air cooling mode.

Outdoor injection fan systems have the advantages of easy determination and measurement of the minimum outdoor airflow and control its required amount. They have been used in many applications by designers. However, a small injection fan is often energy-inefficient. Also, an injection fan needs extra fan and ductwork and results in a more complicated system pressure distribution and control, especially for a retrofit in existing buildings.

Direct Measurement of Minimum Outdoor Air Intake

Minimum ventilation control by direct measurement of the minimum outdoor airflow during minimum outdoor air recirculating mode needs a DDC controller to open or close the minumum outdoor damper and recirculating damper according to the directly measured airflow signal, to provide the required amount of minimum outdoor air for ventilation. The outdoor air at the intake is usually designed according to a velocity of 500 fpm (2.5 m/s). During part-load operation, it may reduce to 35 percent of the design value, that is, 175 fpm (0.87 m/s), or a velocity pressure of 0.002 in. WC (0.5 Pa). In such a velocity pressure, even a multiple-point Pitot-tube averaging probe has difficulty sensing a 10 percent change in outdoor air volume flow rate. The direct measurement of minimum outdoor air intake cannot accurately sense, measure, and provide minimum ventilation control.

Fan Tracking Systems

In a VAV system with a supply and return fan combination using a fan tracking system for minimum ventilation control, the volume flow rate of the supply fan is varied according to the system load while at the same time maintaining a preset duct static pressure at the static pressure sensor in the supply main duct. The volume flow rate of the return fan is controlled to "track" the supply fan at a constant cfm (L/s) less than the supply volume flow rate. If the exhaust damper is tightly closed and does not leak, if there is no infiltration or exfiltration, and if one ignores the difference in densities between the outdoor and recirculating air, the volume flow rate of the supply fan minus the volume flow rate of return fan $\dot{V}_{\text{sf}} - \dot{V}_{\text{rf}}$ is the amount of outdoor air intake in the AHU or PU, designated $\dot{V}_o$, in cfm (L/s).

Because outdoor air is often from one-fifth to one-seventh of the supply air, consider a supply fan with a design volume flow $\dot{V}_{\text{sf}}$ of 10,000 cfm (4719 L/s) and a return fan with a design volume flow $\dot{V}_{\text{rf}}$ of 8500 cfm (4011 L/s), which results in a design outdoor air intake of 10,000 − 8500 = 1500 cfm (708 L/s). Due to the measuring and instrumental errors, $\dot{V}_{\text{sf}}$ reads 2 percent low and $\dot{V}_{\text{rf}}$ reads 2 percent high. The controlled outdoor air intake is then 9800 − 8670 = 1130 cfm (533 L/s), which is about 25 percent less than the design value. Field experience had proved that fan-tracking minimum ventilation control failed to provide the required amount of outdoor air correctly and energy-efficiently.

However, for a VAV system with a supply and return fan combination, when a supply fan reduces its supply volume flow rate as the space load drops, the volume flow of the return fan must decrease accordingly, to meet the following requirements:

- Maintain a desirable space pressurization.
- Exhaust a specified amount of exhaust air and recirculate a specified amount of recirculating air in the AHU or PU.
- Form a new airflow balance in the space.

The volume flow of the return fan tracks the volume flow of the supply fan in order to provide outdoor air that exactly equals $\dot{V}_{sf} - \dot{V}_{rf}$ but it failed to do so in actual practice. However, reduce the speed of the return fan approximately according to $\dot{V}_{sf} - \dot{V}_{rf}$ while the outdoor air intake is guaranteed by DCV or mixed plenum pressure control, and this gives a successful minimum ventilation control experience. This is discussed in a later section.

23.5 CONFERENCE ROOMS

Conference rooms have characteristics of high occupancy density (20 ft^2/person or 2 m^2/person) and a variable occupancy with a schedule that may be quite different from private and general offices. Conference rooms are often critical spaces to provide ventilation for acceptable indoor air quality in office buildings. The following are recommendations of minimum ventilation control for conference rooms in office buildings:

- Use a separate AHU or PU for the conference room with its own outdoor air intake and minimum ventilation control, if possible.
- Increase the total supply airflow rate as well as the outdoor ventilation air rate. According to ASHRAE Standard 62-1999, for a conference room with an occupancy density of 20 ft^2/person (2 m^2/person) instead of 140 ft^2/person (13 m^2/person) in an office, the outdoor air requirement for ventilation should be increased from 0.14 cfm/ft^2 (0.7 L/s·m^2) to about 1.0 cfm/ft^2 (5 L/s·m^2).

In a conference room, the supply air volume flow rate should be determined based on the outdoor air requirement for minimum ventilation. Even the VAV system load and the supply volume flow rate reduce during the occupied period; based on the ASHRAE Standard 62-1999 specified outdoor air requirements, most of the conference rooms still have the required minimum outdoor air to provide ventilation for acceptable indoor air quality.

23.6 SPACE PRESSURIZATION AND RETURN VOLUME FLOW CONTROLS

As discussed in Sec. 20.3, for low-rise buildings except on stormy days, building pressure characteristics are dominated by air systems (including mechanical ventilation systems) most of their operating time.

Characteristics of Space Pressure Control

The following are characteristics of space pressure control:

- According to the airflow balance, the mass flow rate of the outdoor air entering air system $\dot{m}_o$, in lb/min (kg/min), is balanced with the mass flow rate leaving the system, or.

$$\dot{m}_o = -\dot{m}_{inf} + \dot{m}_{ex,r} + \dot{m}_{eu}$$

where $\dot{m}_{ex,r}$, $\dot{m}_{eu}$ = mass flow rate of air exhausted, including air exhausted from the restroom and from the AHU, lb/min (kg/min). In most conditioned space, a space positive pressure $+p_{im}$ is thus formed because of a surplus of outdoor air $-\dot{m}_{inf}$ entering the system, which is then exfiltrated through the gaps and openings on the building shell to outdoors and can be calculated as

$$-\dot{m}_{inf} = -\dot{V}_{inf}\rho_r$$

$$-\dot{V}_{inf} = 4005A_{e,l}\sqrt{\Delta p_{ro}} \tag{23.4}$$

where $-\dot{V}_{inf}$ = volume flow rate of exfiltration, cfm (m^3/min)
$A_{e,l}$ = effective leakage area on building shell, ft^2 (m^2)
ρ_r = space air density, lb/ft^3 (kg/m^3)
Δp_{ro} = space and outdoor pressure difference, in. WC (Pa)

- As discussed in Sec. 20.3, the space and outdoor pressure difference often has a value between + 0.005 and 0.03 in. WC (1.25 and 7.5 Pa). If a pressure sensor in a space pressurization control system needs to sense a 10 percent change in p_{im}, that is, typically + 0.0005 in. WG (0.13 Pa), it would be difficult and expensive to specify such a level of accuracy for most comfort systems.
- For an average leaky one-story office building with an exterior wall area of 10,000 ft^2 (929 m^2) and a floor area of 20,000 ft^2 (1858 m^2), as discussed in Sec. 23.2, from Eq. (20.2)

$$\dot{V}_{leak} = C_{flow}A_w(\Delta p)^{0.65} = 1.0(10{,}000)(0.3)^{0.65} = 4572 \text{ cfm } (2157 \text{ L/s})$$

From Eq. (23.4), for this average leaky office building, the effective leakage area is

$$A_{e,l} = \frac{-\dot{V}_{inf}}{4005\sqrt{\Delta p_{ro}}} = \frac{4572}{4005\sqrt{0.3}} = 2.08 \text{ ft}^2 \text{ } (0.193 \text{ m}^2)$$

Also from Eq. (23.4) the outdoor air intake required to maintain various space positive pressures and to exfiltrate them in this average leaky office building with 10,000 ft^2 (929 m^2) of exterior wall is as follows:

	Required outdoor air intake	
Δp_{ro}, in. WC (Pa)	Exterior wall area, cfm/ft^2 (L/s·m^2)	Floor area, cfm/ft^2 (L/s·m^2)
+ 0.005	0.059 (0.30)	0.0295 (0.15)
+ 0.01	0.083 (0.42)	0.0417 (0.21)
+ 0.02	0.118 (0.60)	0.059 (0.30)
+ 0.03	0.144 (0.73)	0.072 (0.36)
+ 0.05	0.186 (0.95)	0.093 (0.47)
+ 0.1	0.263 (1.33)	0.132 (0.67)

VAV Systems Return/Relief Fan Volume Flow Control

For a VAV system with either a supply and relief fan combination as shown in Fig. 22.3, or a supply and return fan combination as shown in Figs. 22.1 and 22.4, if the volume flow rate of supply air reduces during part-load, the amount of return air extracted along points *ru* should be reduced accordingly in order to provide the required outdoor ventilation air and to maintain a desirable space pressure. Modulating the speed of the return/relief fan with adjustable-frequency drives, and thus its volume flow, to maintain space pressure within predetermined limits is a simple, direct, and effective means of return/relief fan volume control.

Place a pressure sensor in a representative floorspace. The location of the indoor pressure sensor must be away from the door and openings to the ouside, as well as away from the elevator lobbies. The outdoor reference pressure should be typically 10 to 15 ft (3 to 4.5 m) above the building and

well shielded from wind and air velocity effects. As a DDC unit controller reduces the speed of supply fan to maintain a preset duct static pressure during part-load, another DDC unit controller also modulates the speed of return fan to adjust the space pressure p_r, floating within a differential or dead band between an upper limit of + 0.03 in. WG (+ 7.5 Pag) and a lower limit of + 0.005 in. WG (+ 1.25 Pag) using a floating control mode. The exhaust/relief damper is often opened between 0 and 30 percent during minimum outdoor ventilation recirculating mode and opened 100 percent during all outdoor air economizer mode.

If a VAV system using a supply and relief fan combination is operated during minimum outdoor air ventilation recirculating mode with a constant-volume flow of minimum outdoor ventilation air provided by mixing plenum pressure control, and if the building that this VAV system serves has an air tightness between an average leaky and leaky building, then

- Either the relief fan is turned off and the relief damper is closed, so that all the outdoor air is exhausted from the restrooms, other exhausts, and exfiltrates through the exterior leakage area.
- Or the relief fan is turned on at low speed and the relief damper exhausts only a volume flow less than 30 percent of the designed volume flow. The remaining portion of the outdoor air is exhausted outdoors through restrooms, other exhausts, and exfiltrates through the exterior leakage area.

During minimum ventilation recirculating mode, these exhausts and exfiltration, plus the recirculating air extrated along points *ru* (recirculating/exhaust chamber) and *m* (mixing plenum) by the supply fan are automatically balanced with the design and reduced supply volume flow rate at part-load.

Another alternative of return/relief fan volume control in VAV systems is airflow tracking. Required return/relief volume flow corresponding to various reduced supply airflow at part-load are determined during commissioning by means of multipoint Pitot tube measurements. Return/relief fan speed and volume flow can then be modulated at part-load accordingly. Airflow tracking is an open-loop control. Supply fan volume flow must be adjusted to match with the volume flow of the return fan during morning warm-up and cool-down mode.

23.7 DISCHARGE AIR TEMPERATURE CONTROLS

Basics

In a multizone VAV system, the discharge air temperature T_{dis}, in °F (°C), is the conditioned air temperature leaving the supply outlet of the AHU or PU, where $T_{\text{dis}} = T_{cc} + \Delta T_{\text{sf}}$. Here T_{cc} represents the temperature of the conditioned air leaving the coil, and ΔT_{sf} indicates the temperature rise across the supply fan, both in °F (°C). Typically, $T_{\text{dis}} = 55 + 2 = 57$°F (13.9°C). The difference between the supply temperature from the ceiling diffuser mounted on the ceiling plenum and the discharge air temperature $T_s - T_{\text{dis}}$, in °F (°C), is the rise or drop in temperature resulting from the duct heat gain or duct heat loss ΔT_{du}. For a typical multizone VAV system, $T_s - T_{\text{dis}}$ is usually a function of the temperature difference between the supply air and the ambient air surrounding the supply duct and the supply volume flow rate. Its relationship can be expressed as $T_s = T_{\text{dis}} \pm \Delta T_{du}$.

As discussed in Secs. 20.18 and 21.3, for a summer indoor design temperature of 75°F (23.9°C), a lower conditioned air off-coil temperature T_{cc} and discharge air temperature T_{dis} result in the following:

- A greater supply temperature differential $\Delta T_s = T_r - T_s$, a lower space relative humidity, a smaller supply volume flow rate $\dot{V}_s$, fan sizes, fan energy use, duct, terminal, and diffuser sizes
- A lower chilled water temperature entering and leaving the cooling coil T_{we} and T_{wl}, a lower evaporating temperature in the DX coil T_{ev}, and therefore higher power input to the refrigeration compressors

Excluding the the cold air distribution in the ice storage system, discussed in Chap. 31, a ΔT_s between 15 and 20°F (8.3 and 11.1°C) during summer design conditions is most widely used in air

systems. The following list relates parameters when ΔT_s = 15 and 20°F except in an ice storage system in which ΔT_s may increase up to 30°F (16.7°C). All units are in °F (°C), except the space relative humidity φ_r is in percent:

ΔT_s	T_{cc}	ΔT_{sf}	T_{dis}	ΔT_{du}	T_s	T_r	φ_r
15 (8.3)	55 (12.8)	2 (1.1)	57 (13.9)	3 (1.7)	60 (15.6)	75 (23.9)	50
20 (11.1)	50 (10.0)	2 (1.1)	52 (11.1)	3 (1.7)	55 (12.8)	75 (23.9)	45

System Description

Discharge air temperature control consists of the following control loops, as shown in Fig. 23.6*a*:

- *Economizer control loop*. For a temperature air economizer control, it consists of an outdoor temperature sensor T2, a recirculating air temperature sensor T3, a DDC controller, outdoor damper actuators D1 and D2, a recirculating damper actuator D3 and a relief damper actuator D_4.
- *Cooling coil control loop*. It consists of a discharge air temperature sensor T1, a DDC controller, and a cooling coil valve actuator V1.
- *Mixed air control loop*. It consists of a mixed air temperature sensor T4, a DDC controller, outdoor damper actuators D1 and D2, and a recirculating damper actuator D3.
- *Heating coil control loop*. It consists of a discharge air temperature sensor T1, a DDC controller, and a heating coil valve actuator V2.
- *Low-limit control loop (coil freeze protection)*. It consists of a temperature sensor T4, a DDC controller, and outdoor air damper actuators D1 and D2.

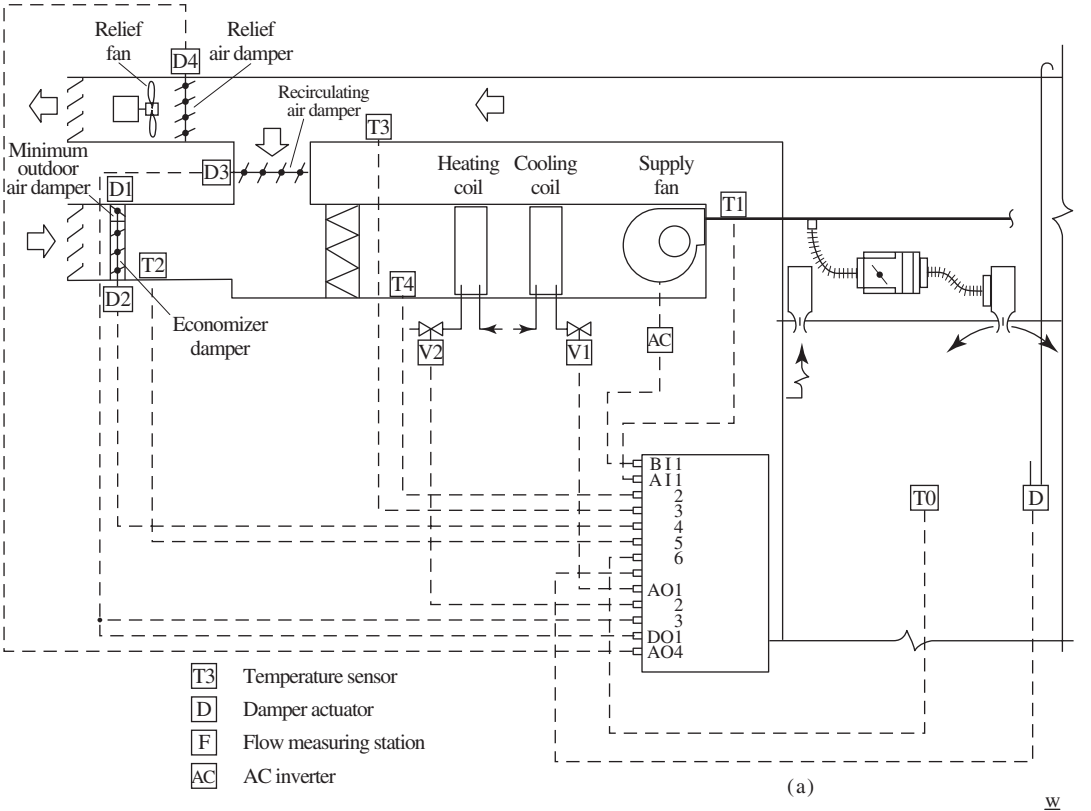
For VAV systems, discharge air temperature control is a kind of cooling and heating capacity control so that conditioned air is discharged from an AHU or PU with a predetermined temperature and relative humidity, such as 57°F (13.9°C) and 95 percent relative humidity, and can perform the following requirements:

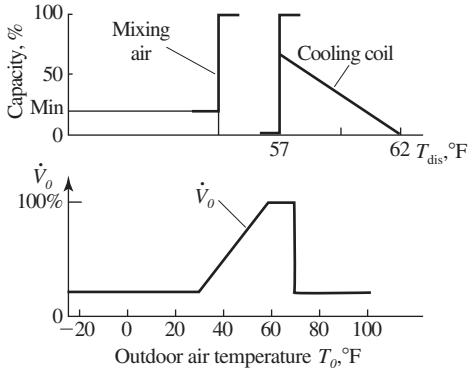
1. Provide a kind of conditioned air for zone temperature control by varying the supply volume flow rate.
2. Based on the conditions of air leaving the coil sensed by sensor T1, provide economizer control and modulate the cooling or heating capacity of the AHU or PU to maintain a predetermined T_{dis}.
3. In VAV systems, if an AHU or a PU serves both perimeter and interior zones, usually perimeter heating is provided by the heating coil installed in the reheating VAV boxes or fan-powered VAV boxes or by baseboard heating directly. In such a condition, T_{dis} should meet the free-cooling requirement in interior zones even in winter.
4. According to the outdoor temperature or system load, reset T_{dis} to an optimum value for both effective operation and energy saving.

For a cooling coil or heating coil in the AHU, the DDC controller modulates the coil valves continuously. However, in a rooftop packaged unit, refrigeration capacity is often adjusted by cycling the compressors on and off in the separate refrigerant circuits of the DX coils. The capacity control of packaged units is discussed in Sec. 11.8.

Most VAV systems use the air economizer or water economizer as the first-stage cooling to save energy. Cooling coil, mixed air, and heating coil control loops are operated in sequence so as to maintain a predetermined discharge air temperature while preventing simultaneous cooling and heating.

For a typical AHU, the set point of the discharge air temperature at summer design conditions may be 57°F (13.9°C). To avoid sudden repeated changes in operating mode among the cooling coil and mixed air control loops, resulting in unstable operation, the set point for mixed air control is better set at 55°F (12.8°C) as shown in Fig. 23.6*b*.





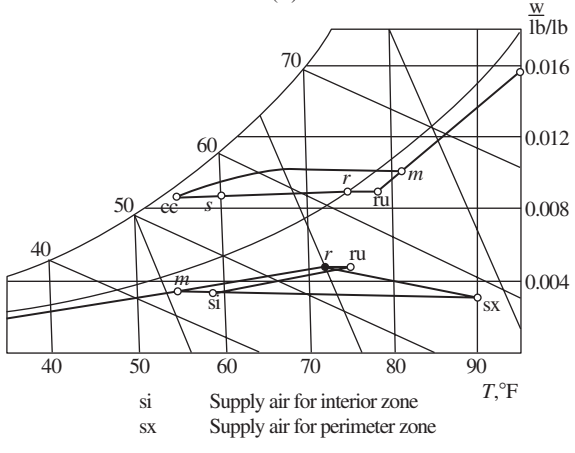


FIGURE 23.6 Discharge air temperature control for a typical AHU: (*a*) control diagram; (*b*) output diagram; (*c*) air conditioning cycle.

The modulation of valve V1 in the cooling coil control loop of discharge air temperature control can be performed in parallel with the control operations in 100 percent air economizer control. Although mixed air control is part of air economizer control, it is also a part of the sequence control in a discharge air temperature control. Mixed air control mixes the outdoor air with the recirculating air by modulating the economizer damper and the recirculating damper as to provide a mixed air temperature, such as 55°F (12.8°C) for the sake of free cooling in the interior zone. Mixed air control functions only when $T_o < 55$°F (12.8°C) and ends during minimum outdoor air heating mode.

For VAV reheat and fan-powered VAV systems, heating is often provided by a water heating coil or an electirc heating coil in a terminal directly above the conditioned space. The modulation of the VAV box and the control valve of the heating coil in sequence forms the zone cooling and heating control.

For a single-zone VAV system, a DDC unit actuates the economizer dampers, cooling coil valve, economizer and recirculating dampers, and heating coil valves in sequence, based on the input signal from a zone temperature sensor to maintain a predetermined zone temperature. The discharge air temperature control is replaced by a zone temperature control.

Operation of Air Economizer and Outdoor Air Intake—Case Study

The outdoor air intake of a fixed dry-bulb air economizer in an AHU of a VAV system shown in Fig. 23.7 is from Nabinger et al. (1994). In Fig. 23.7, the abscissa of the diagram is the indoor-outdoor temperature difference of the conditioned space that the VAV system serves and the outdoor

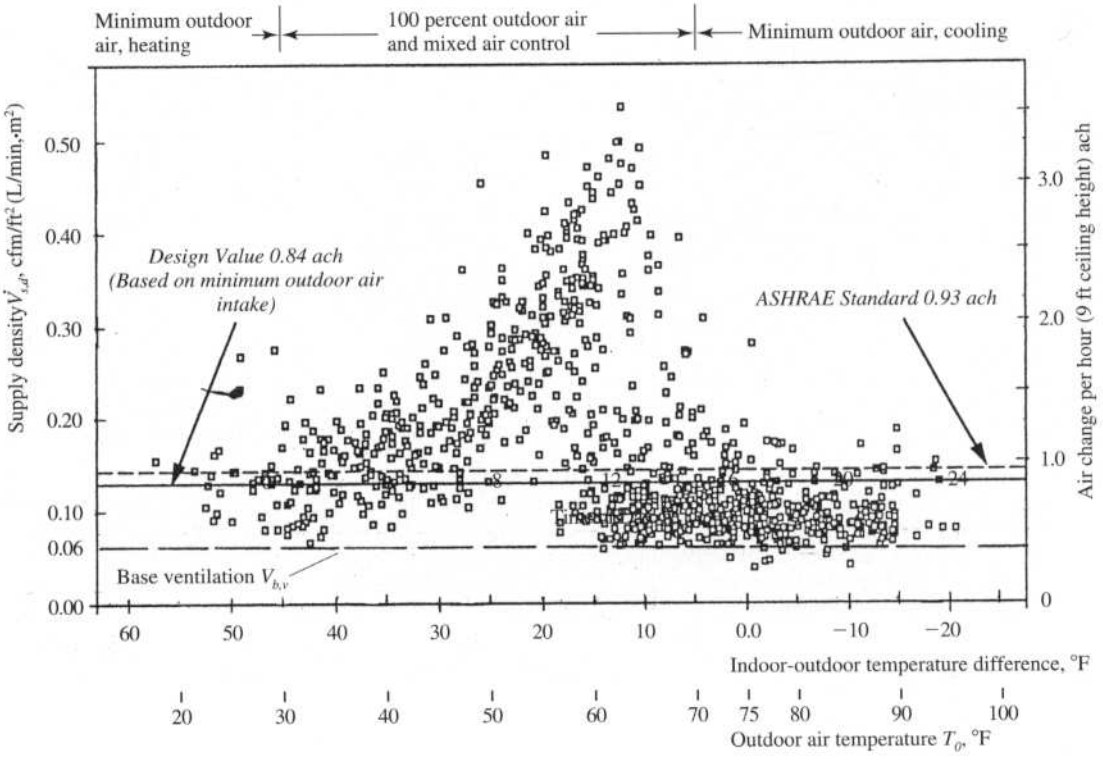


FIGURE 23.7 The outdoor air intake of a temperature air economizer in a VAV system. (*Modified from Nabinger et al., ASHRAE Transactions, Part II, 1994. Reprinted by permission.*)

air temperature, both in °F (°C). The ordinate is the measured outdoor air intake of a fixed dry-bulb air economizer in a VAV system, expressed in supply flow density, cfm/ft^2 (L/s · m^2), and ach based on a ceiling height of 9 ft (2.74 m). According to Nabinger et al. (1994), the test office building is located in Overland, Missouri. The building consists of seven floors with a floor area of 378,000 ft^2 (35,100 m^2). Construction began in 1988, and occupancy began in late 1990.

The air conditioning system is a VAV system with fan-powered VAV boxes in the perimeter zone and VAV boxes in interior zones. There are all together 15 AHUs with a total supply volume flow rate of 267,000 cfm (126,000 L/s), and the design minimum outdoor air intake for this building is 50,600 cfm (23,900 L/s).

Outdoor air intake in the AHUs was measured by using the trace gas decay technique with SF_6 as the trace gas. Trace gas was injected into each of the 15 AHUs and allowed a period of about 20 to 30 min to mix. Four decay tests were conducted each day. The tracer gas injection rates were calculated based on a target concentration of 150 ppb (parts per billion). Tracer gas concentration was monitored at 19 indoor locations and an outdoor location with the concentration measured at each location every 10 min.

Operation of Air Economizer. When the outdoor air temperature T_o was below a set point value, such as $T_o < 30$°F (−1.1°C), the AHU or PU was set to minimum outdoor intake to protect it from freezing and to avoid heating more outdoor air than necessary. The AHU was operated in minimum outdoor air heating mode, as shown in Fig. 23.7.

When the outdoor temperature 30°F $\leq T_o \leq$ 70°F (−1.1°C $\leq T_o \leq$ 21.1 °C), the DDC controller in mixed air control modulated the economizer and recirculating dampers to provide a mixture of outdoor and recirculating air at the required mixed air temperature, such as 55°F (12.8°C). The air economizer was operated in mixed air control. Here the outdoor air temperature from which the mixed air control began $T_{o,\text{mix}} = 30$°F (−1.1°C) depended on the percentage of outdoor air among the supply air, and on the load ratio.

When the outdoor air temperature $T_o > 70$°F (21.1°C), it was more economical to cool the recirculating air than the outdoor air. The minimum outdoor and recirculating dampers opened, and the economizer damper was closed. The AHU or PU was operated in minimum outdoor air cooling mode.

In Fig. 23.7, the highest measured outdoor air supply flow density was out 0.52 cfm/ft^2 (2.6 L/s · m^2), which was lower than the average building supply flow density 267,000/378,000 = 0.71 cfm/ft^2 (3.5 L/s · m^2). The measured minimum outdoor air intake was lower than the ASHRAE Standard 62-1999 specified minimum outdoor air intake of 0.93 ach, or 0.14 cfm/ft^2 (0.7 L/s · m^2). The average supply flow density for VAV systems at design flow for a typical floor (including perimeter and interior zones) usually varies between 0.8 and 1.3 cfm/ft^2 (4 and 6.5 L/s · m^2). The reason for having lower flow density was a lower space load density at part load.

There were a certain number of minimum outdoor air intakes even lower than the base ventilation. It is interesting to note that the amount of minimum outdoor air intake during heating mode was higher than the amount of minimum outdoor air intake during cooling mode. This may be due to the higher density of cold outdoor air in cold weather than the lower density of outdoor air in warm weather.

Carbon Dioxide Concentrations. From November 1991 to November 1992, CO_2 measurements under full occupancy revealed that instantaneous daily peak values in the return ducts ranged from 450 to 750 ppm. In the space, they ranged from 500 to 850 ppm. On a few isolated occasions, CO_2 concentrations in the return ducts were as high as 1100 ppm; in the space, they were as high as 1400 ppm when some of the AHUs were shut off.

Discharge Air Temperature Reset

Discharge air temperature or supply temperature reset saves energy in the following ways:

- It reduces the simultaneous cooling and heating if VAV reheating and dual-duct VAV systems are used.

- It lowers the duct heat gain or loss.
- It raises the suction temperature of the refrigeration system.
- It increases the number of hours that the refrigeration system can be shut off during an air or water economizer cycle.
- It increases the zone supply volume flow rate at low zone loads.

Discharge air temperature reset also has the following drawbacks:

- Lower fan energy savings because less volume flow can be reduced
- A higher space relative humidity during cooling mode operation

During part-load operation, because the reduction in space load and supply volume flow rate in the perimeter zone is different from that in the interior zone, a separate discharge air temperaure reset schedule for perimeter and interior zones is beneficial if separate AHUs or PUs for perimeter and interior zones are adopted, or a dual-duct system is used.

For a conditioned space that requires strict relative humidity control, discharge air temperature reset is not recommended. For a VAV system that uses the same AHU or PU to serve both perimeter and interior zones, discharge air temperature reset must consider the higher space cooling load at part load in the interior zone. In VAV systems using inlet vanes for duct static pressure and system supply volume flow control, discharge air temperature reset is preferable because it may save more energy. In VAV systems an analysis is often beneficial to determine whether discharge air temperature reset saves more energy and is cost-effective.

23.8 DUCT STATIC PRESSURE AND FAN CONTROLS

Duct Static Pressure Control

In a VAV system, if more than one half of the VAV boxes close their dampers at the same time to reduce the volume flow supplied to various zones at part-load operation, the static pressure of the supply duct will be raised. Such a high static pressure may cause objectionable noise and volume flow in some of the terminals and may result in unstable operation. At part load, the volume flow and the fan total pressure of the supply fan as well as the return fan should be reduced to achieve a balanced and optimized system operation if a supply and return fan combination is employed. The reduction of the volume flow of the return fan is discussed in Sec. 23.6.

The purpose of supply duct static pressure control (or simply duct static pressure control) is to maintain a nearly constant static pressure at a specific location in the main supply duct and, at the same time, to save fan energy use. A duct static pressure control system consists mainly of a static pressure sensor, a DDC controller, and an adjustable-frequency variable-speed drive, or an inlet vanes actuator, as shown in Fig. 23.8. When the static pressure sensed by the static pressure sensor at the predetermined location in the supply main duct is higher than a set point, the DDC controller actuates the ac inverter or positions the inlet vanes to reduce the fan volume flow and fan total pressure, so as to maintain a nearly constant static pressure at the location of the sensor.

Duct static pressure shows a small capacitance lag, or time constant. Therefore, duct static pressure responds quickly to changes in the supply fan output. Proportional-integral-derivative (PID) control action with a wider proportional band is more appropriate.

A high-limit pressure sensor should be installed after the supply fan outlet to prevent damaging the duct system by means of excessive pressure due to malfunction, or closing of the fire or smoke dampers. When these circumstances arise, the excessive static pressure buildup due to the blockage of the air passage can be sensed by the high-limit pressure sensor, and the DDC controller will send a signal to turn off the fan.

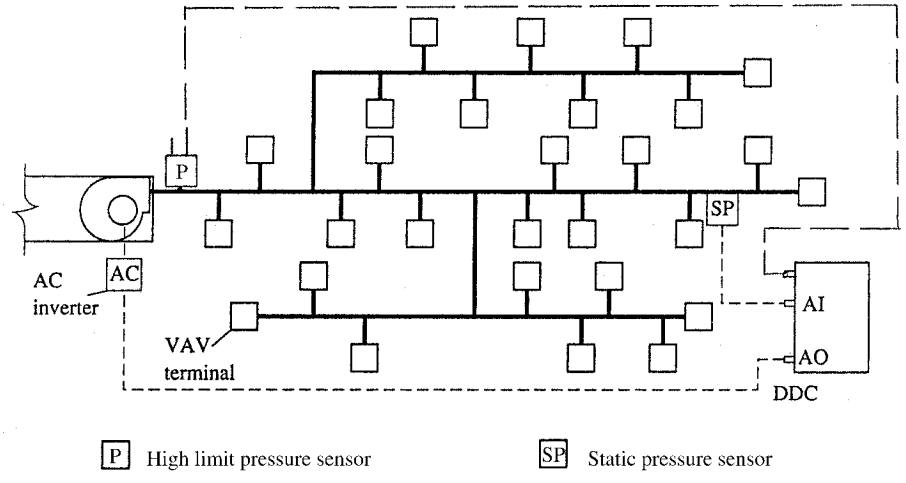


FIGURE 23.8 Duct static pressure control.

Set Point of Duct Static Pressure and Sensor's Location

The set point of the duct static pressure should be the minimum static pressure such that when added to its associated velocity pressure, their sum—the total pressure—should overcome the maximum total pressure loss in any of the branch takeoffs including VAV boxes, flexible ducts, diffusers, and dynamic losses under all operating conditions. The duct static pressure is more stable than the duct total pressure in a supply main duct when the airstream is flowing. More often, the set point of the duct static pressure in VAV systems ranges between 0.8 and 1.5 in. WG (200 and 375 Pag)

The static pressure sensor should be located in the supply main duct in a position such that the required supply volume flow rate is guaranteed for all VAV boxes with greatest energy savings during part load. The nearer to the farthest branch takeoff, the greater the energy saving because Δp_{fix} is nearly minimum. For most VAV systems, the duct pressure sensor is often located in the main duct near the farthest branch takeoff and at a location where steady static pressure can be measured. Duct static pressure control is discussed also in Sec. 29.5.

Warren and Norford (1993) recommended that the duct static pressure set point be reset based on messages from each VAV box collected by a poll manager control block during operation. Such a DDC control system is complicated and expensive.

ASHRAE/IESNA Standard 90.1-1999 specifies that duct static pressure sensors used to control variable-air-volume fan operating characteristics shall be placed in a position such that the controller set point is no greater than one-third of the total design fan (supply) static pressure except for direct digital control systems with zone reset capability where the sensor may be located at the fan discharge. For a supply duct system with major duct splits, multiple sensors should be installed in each major branch to ensure static pressure can be maintained in each of the major branches.

For air systems with direct digital control of individual zone VAV boxes connecting to the central system controller, the duct static pressure set point shall be reset based on the zone requiring the most pressure, that is, the set point is reset lower until one zone damper is nearly wide open.

Comparison between Adjustable-Frequency Drives and Inlet Vanes

In VAV systems, both adjustable-frequency variable-speed drives and inlet vanes are used to modulate the volume flow and fan total pressure of the centrifugal supply, return, relief, and exhaust fans, as described in Sec. 15.4. Generally, fans using adjustable-frequency variable-speed drive to vary

the fan speed are more energy-efficient at reduced volume flow than are positioning inlet vanes. However, adjustable-frequency variable-speed drive is more expensive than inlet vanes.

Lorenzetti and Norford (1992) reported that adjustable-frequency variable-speed drives consumed 46 to 66 percent less supply fan energy than inlet vanes, based on the measured performance of four AHUs during a retrofit project in two connected buildings. The brake horsepower of the supply fan ranged from 21 to 36 hp. Only one of the AHUs operated 24 h/day, while the rest operated 8 to 15 h/day. From the results of energy simulation programs, other researchers also found that adjustable-frequency variable-speed drives have an energy savings between 20 and 50 percent over inlet vanes depending on the type and size of fan, annual operating hours, and characteristics and location of the building.

Field measurements also showed that many fans installed with inlet vanes saved only 20 to 30 percent, even as little as 10 percent, when the volume flow rate was reduced to 0.3 of its design value, if inlet vanes are mounted too far away from the impeller. Theoretically, the energy savings should be greater than 50 percent.

Figure 23.9 compares the fan power consumption at fractions of the design volume flow rate when adjustable-frequency variable-speed drives and inlet vanes are used. Adjustable-frequency drives are often cost-effective under the following conditions:

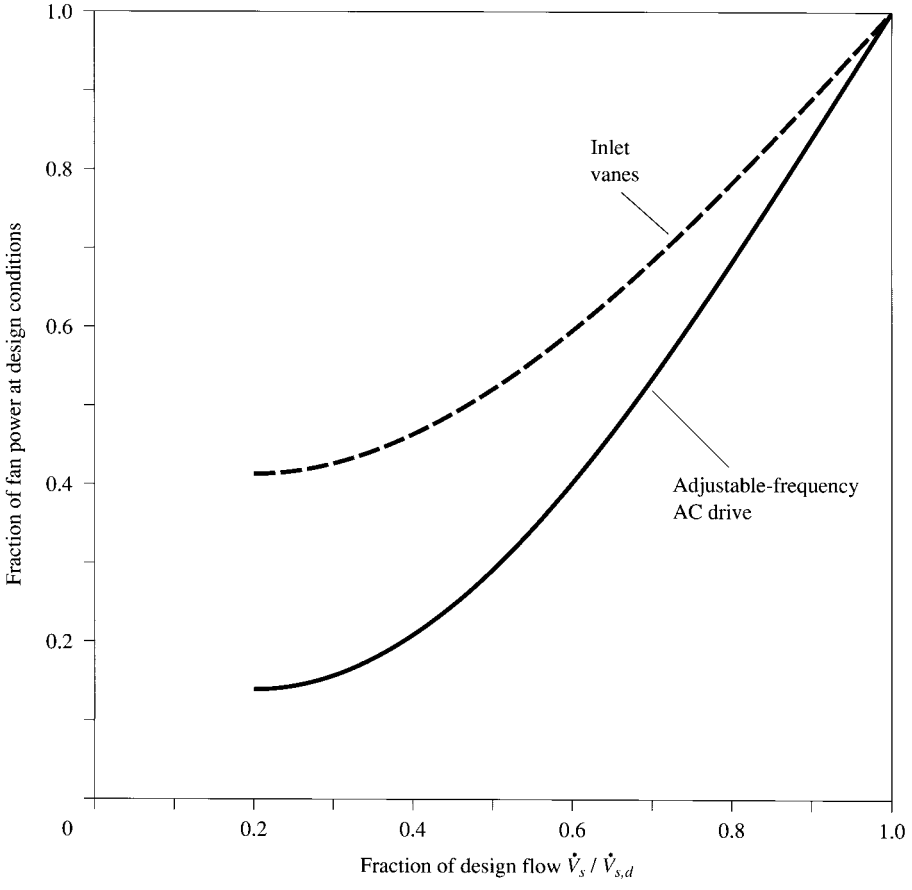


FIGURE 23.9 Comparison of fan power consumption between adjustable-frequency, variable-speed drives and inlet vanes.

- A variable supply volume flow is required during operation.
- The fan brake horsepower is 25 hp (19 kW) and greater.
- Backward-inclined and airfoil fans are the best candidates.
- The annual operating hours exceed 2500 h.

Mays (1998) suggested that the most common adjustable-frequency drive problem today is reflected wave voltages, which create two to three times overvoltage at the motor terminals with pulse-width-modulated (PWM) drives as the rise time of the pulses becomes shorter. These cause noise in motors. An external low-pass filter can be installed, or apply series reactors between the PWM output terminals and the cable to the motor to slow the rise time or to minimize overvoltages. The shorter the cable between the adjustable-frequency drive output and the motor, the lower will be the peak voltage.

For a vane-axial supply fan, blade pitch controls are often used at reduced volume flow for fan energy savings. Controllable blade pitch vane-axial fans have fan energy savings characteristics comparable to those of adjustable-frequency variable-speed drives. But they are also expensive.

23.9 FUNCTIONAL CONTROLS FOR VAV SYSTEMS

Nighttime Setback, Warm-Up, or Cool-Down Control

Many VAV systems are shut down during the nighttime unoccupied period. Only the heating system or the fan-powered VAV boxes are operated to maintain a nighttime setback temperature set point in order to save energy. The purpose of warm-up control is to recover the setback temperature and maintain an indoor environment between 70 and 72°F (21.1 and 22.2°C) during the occupied period in winter.

The purpose of cool-down control is to cool and dehumidify the space air down to predetermined limits following an increase in the space air temperature and humidity when the air system is turned off during the unoccupied period in locations where the outdoor climate is hot and humid in summer.

When the supply fan of the AHU or PU is turned on, if the difference between the space temperature T_r sensed by the temperature sensor and the set point T_{sp} ($T_r - T_{sp}$) is less than or equal to a predetermined limit, such as 3°F (1.7°C), then the air system enters the warm-up mode; and if ($T_r - T_{sp}$) $\geq$ a predetermined limit, such as 3°F (1.7°C), the air system enters the cool-down mode. Both the warm-up and cool-down controls close the minimum outdoor, economizer, and relief/exhaust dampers and open to 100 percent the recirculating damper. In addition, the warm-up control opens fully the two-way valve of the heating coil or the gas valve of the furnace, and the cool-down control opens fully the two-way valve of the cooling coil or starts all the refrigeration compressors that serve the DX coil, until the space temperature drops within predetermined limits.

One problem in warm-up or cool-down control is to determine the optimum start time. The optimum start time is affected by the following factors:

- Space temperature and relative humidity just prior to the warm-up or cool-down period
- Construction of the building envelope
- Capacity of the heating/cooling equipment
- Internal loads
- Outdoor temperature

These factors are used to calculate the thermal characteristics of a conditioned space served by an AHU or PU, and start the equipment prior to occupancy to reach the desirable space temperature. There are software packages available to calculate separate sets of heating and cooling rates for

spaces that have been unoccupied less than or greater than 24 h and provide an optimum start or stop time of the air system.

A simple and effective way that is worthwhile to try is as follows: Record the outdoor air temperature, space temperature, relative humidity, internal load, and equipment and damper operating conditions at suitable time intervals, say, 10 min to 1 h, during the warm-up or cool-down period; analyze and determine the optimum start time once sufficient data have been accumulated. Modify the results until they are verified by experience.

Some manufacturers also provide cycling capacity during morning warm-up to maintain the space temperature reached before the occupied period.

If both cool-down and purge mode or warm-up and purge mode operations are required, and the outdoor air is significantly hot and humid, or colder than the space air during the occupied period, then the purge operation must be performed first, and the cool-down or warm-up operation follows until the space temperature is maintained within predetermined limits. During purge mode, the minimum outdoor air, economizer, and exhaust dampers are opened 100 percent, and the recirculating damper is closed. During warm-up or cool-down mode, the minimum outdoor air, economizer, and exhaust dampers are closed, and the recirculating damper is opened 100 percent.

Only when the outdoor air can be used for free cooling, and to squeeze out the contaminated indoor air at the same time, are the cool-down and purge operations combined into a single operation.

Steam Humidifier Control

A steam humidifier is the most widely used humidifier when a warm air supply is required in winter in an AHU or PU. A humidity sensor is often located in the supply duct just after the supply fan discharge, to sense the relative humidity of the discharge air. The DDC controller receives the sensed signal and positions the two-way valve to modulate the steam flow rate entering the steam humidifier, as shown in Figs. 15.42 and 15.43, until the relative humidity of the discharge air meets the required set point.

When the fan is turned on, the humidifier control is energized. The steam valve closes when the fan is turned off.

If a steam humidifier is installed, it is critical that the humidifying capacity not be exceeded, so that a wet surface does not occur in the AHU as well as in the supply duct in order to maintain an acceptable indoor air quality.

Dew Point Control

For air systems serving conditioned space that needed both close control of the space temperature and relative humidity with a sensible ratio of the space conditioning line SHR_s above 0.9, a dew point control is often used. Dew point control involves using a water cooling coil, DX coil, or air washer to cause a specific amount of conditioned air to cool and dehumidify or occasionally cool and humidify to point *cc*, such as that on Fig. 23.10. Because of the high relative humidity of the departing air, the temperature T_{cc} is approximately equal to the dew point on the saturation curve.

When the indoor space temperature is to be maintained at 75°F (23.9°C), with a relative humidity of 50 percent, $T_{cc} = 55$°F (12.8°C), $\varphi_{cc} = 97$ percent, and $SHR_s > 0.95$, the state point of the conditioned space *r* must move approximately along the horizontal dew point line. Either the sensible cooling load is absorbed by the supply air or reheating is provided to the supply air by a reheating coil to compensate the sensible load at part load, if the space temperature is maintained at 75°F (23.9°C), the relative humidity must be at 50 percent. A temperature sensor senses the space temperature T_r, and a DDC controller modulates the capacity of the reheating coil according to the sensed space temperature to maintain $T_r = 75$°F (23.9°C) with $\varphi_{cc} = 50$ percent.

Dew point control is simple and effective. Its main drawback is its need for reheating which wastes energy. Dew point control is mainly used for industrial applications such as clean rooms.

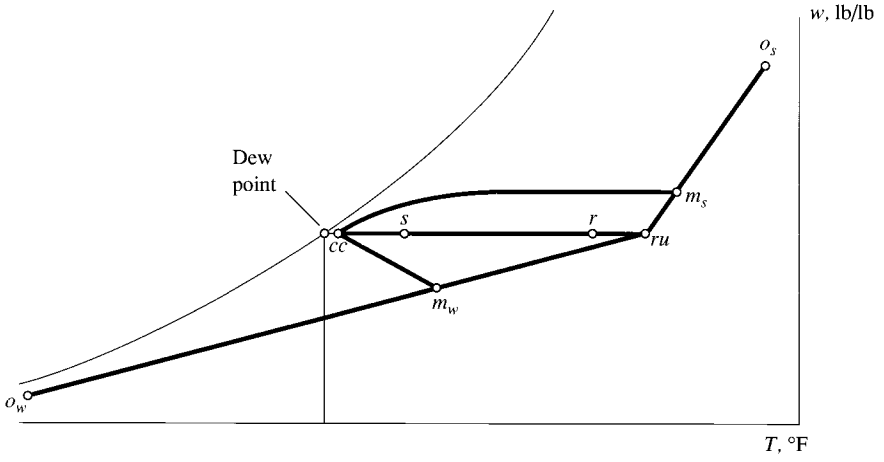


FIGURE 23.10 Dew point control.

Diagnostics

The system diagnostics automatically monitors the operation of all workstations, modems, LAN connections, control panels, and controllers. The failure of any device should be displayed in a diagnostic display module for which a legend is provided. In a recent model by a manufacturer, 49 different diagnostics can be read at the display module, and the last 20 diagnostics can be held in an active history buffer log in the module. The diagnostic module is also capable of displaying outputs of microprocessor-controlled run tests to verify operation of every thermistor, potentiometer, fan, and compressor before the air system is started.

A smoke detector is usually located in the return air passage. Once the smoke detector detects smoke in the return airstream, it sends a signal to the DDC controller and energizes the fire alarm system, as shown in Fig. 21.3*b*. Smoke controls and stairwell pressurization systems are discussed in Chap. 22.

A pressure sensor is usually used to sense the pressure drop across the filter. As soon as the pressure drop exceeds a predetermined limit, an alarm signal appears at the display module to indicate the need for replacement or maintenance.

Safety controls are often part of the system components in HVAC&R equipment, as discussed previously.

23.10 RECOMMENDATIONS AND INTERACTION BETWEEN CONTROLS

Recommendations for VAV Controls

- Equipment, control valves, and air dampers should be properly selected and sized. Oversized equipment and components degrade the control quality.
- Proportional-integral (PI) control mode is the widely used control mode in HVAC&R applications. PI control eliminates the offset, resulting in a better indoor environment and a lower energy

consumption than proportional control mode. A wider proportional band and properly selected proportional and integral gains for the controller provide a stable and effective operation.

Cold discharge air temperature reset for the perimeter zone is energy-efficient when PI or proportional-integral-derivative (PID) control mode is used.

- The use of an air economizer cycle is energy-efficient and provides adequate outdoor ventilation air for occupants.
- A sensor should be located where the value it senses represents the controlled variable of the whole zone to be controlled. A sensor should be exposed and located in a place that has sufficient air movement. However, it should not be directly under a supply air jet, in which the air temperature and relative humidity are quite different from those in the occupied zone. It also should not be located in a stratified airstream.
- To measure the duct static pressure, if a long section of straight duct is available (say, a length of duct section greater than or equal to 10 duct diameters), a single point, Pitot-tube type of duct static pressure sensor can be used. Otherwise, a multipoint Pitot-tube array of flow-measuring station with airflow straighteners should be used for velocity pressure or static pressure measurement. The small holes used to measure static pressure in a Pitot tube should never be directly opposite an airstream with a velocity pressure that can affect the reading. A reference pressure should be picked up at a point with low air velocity outside the duct, at a point served by the same air system, or in the ceiling plenum.
- A space pressure sensor should be located in an open area of the conditioned space where the air velocity is less than 40 fpm (0.2 m/s) and where its reading is not affected by the opening of doors. The reference pressure pickup is best located at the rooftop, at a level 10 ft (3.0 m) above the building and shielded from the influence of wind gusts.
- If a simpler control system can do the same job as a more complex system, the simpler system is always the first choice.

A sufficiently clear, well-followed operations manual and a well-implemented maintenance schedule are key factors for an effective control system.

Interaction between Controls

In a typical multizone VAV system, each of the zone temperature control, discharge air temperature control, minimum ventilation control, duct static pressure control, space pressurization control, and warm-up or cool-down controls may consist of a single closed control loop, or several closed control loops combined together. The interaction between these control loops can be categorized as follows:

- *Sequence control*. For this kind of control action, several individual control loops operate in sequence in order to maintain a particular controlled variable within predetermined limits to prevent simultaneous cooling and heating. For example, the discharge air temperature control in an AHU or PU consists of an air economizer control loop as the first-stage cooling, a cooling coil control loop, a mixed air temperature control loop, and a heating coil control loop in sequence. The two cooling control loops can be actuated simultaneously, whereas the cooling coil, mixed air, and heating coil control loops must be operated in sequence to prevent simultaneous cooling and heating.
- *Override*. If two control loops operate on the same control device, one control loop must override the control action of the other control loop. One must have a priority to actuate the control device. The predetermined precedence of one control signal over another in the DDC controller is called override. For example, the minimum ventilation control and the air economizer control both actuate the minimum outdoor air damper, economizer dampers, and recirculating dampers. When the DDC controller fully opens the economizer and minimum outdoor dampers, the air economizer control overrides the minimum ventilation control because the outdoor air intake during the 100 percent air economizer cycle is far greater than the minimum outdoor air intake during minimum ventilation control.

- Control loops are often operated independently of one another. For example, the cooling coil control loop that modulates the two-way valve of the chilled water supply is operated independently of the space pressurization control loop that modulates the speed of the return fan to maintain the space pressure within preset limits.
- *Control actions affect each other*. The control action of one control loop somtimes may affect the controlled variable of another control loop. For an example, the closing of the damper in a VAV box of a zone temperature control loop reduces its volume flow and raises the static pressure of the duct static pressure control loop. As the speed of the supply fan is reduced, the supply volume flow rate and the fan total pressure are reduced accordingly. The duct static pressure is then maintained within predetermined limits, and a new system balance is formed.

Stability Problems

- In VAV controls, instability may occur as the result of hunting because of a smaller proportional band and a fast opening-closing controlled device.
- Avery (1992) noted that interaction problems may occur because the stroke speeds of the valve actuator or damper actuator may be different, or the sensor response times may not be the same for system components from different manufacturers.
- Control loop interaction between pneumatic operated VAV boxes and the electric inlet vane actuators of the supply fan can be serious as the slow-operating electric inlet vane actuator cannot keep pace with the changes in duct static pressure caused by fast-acting pneumatic actuators.
- Control loops having control actions affecting each other sometimes may destabilize the system.

23.11 SEQUENCE OF OPERATIONS OF A VAV REHEAT SYSTEM WITH MINIMUM VENTILATION, DISCHARGE AIR TEMPERATURE, AND DUCT STATIC PRESSURE CONTROLS—CASE STUDY

HVAC&R System

Consider a multizone VAV reheat system using an AHU to serve a typical floor in an office building with perimeter and interior zones. The summer and winter design conditions are similar to those for a VAV reheat system described in Sec. 21.3, except the winter design temperature is 30°F (− 1.1°C) with a design outdoor humidity ratio of 0.003 lb/lb (0.003 kg/kg). Humidification is not required in the AHU during the winter heating mode.

In minimum ventilation control, the sequence of operations of both CO_2-based demand-controlled ventilation and the alternative mixed plenum pressure control at the AHU intake are included. The discharge air temperature control consists of a temperature air economizer cycle, a water cooling coil control loop, a mixed air temperature control loop, and a water heating coil control loop. The set point that changes over from minimum outdoor air recirculating mode to 100 percent outdoor air free-cooling mode is 75°F (23.9°C). There is a minimum outdoor damper, an economizer damper, a recirculating damper, and an exhaust damper.

In duct static pressure control, supply and return fan pressure and volume flows are modulated by varying the speeds of the supply and return fans.

Sequence of Operations

The recommended year-round sequence of operations of this VAV reheat system is as follows:

1. During the unoccupied period, the time-of-day clock shuts down the AHU and its associated supply and return fans. Outdoor minimum damper D1, economizer damper D2, recirculating

damper D3, exhaust damper D4, cooling coil valve V1, and heating coil valve V2 are all closed. Supply and return fan motors are off.

2. During the unoccupied period, an outdoor air temperature sensor (temperature transmitter) T1 measures the outdoor temperature T_o, and an indoor temperature sensor TT11 measures the indoor temperature T_r. Also an outdoor CO_2 sensor $CO_2 - 1$ measures the outdoor CO_2 concentration CO_{2o}, and an indoor CO_2 sensor $CO_2 - 2$ measures the concentrations of a representative floor space CO_{2r}.

In hot summer, if the summer indoor design temperature $T_{r,\text{des}} = 75°F$ (23.9°C) and the measured representative zone temperature $T_{rxn} \geq 3°F$, a cool-down mode is required. If the measured indoor-outdoor CO_2 concentration difference ($CO_{2r} - CO_{2o} \geq 200$ to 300 ppm, a purge mode is also required. If both cool-down and purge modes are required and if $T_o \geq 78°F$ (25.6°C), the purge mode must be performed first, then the cool-down mode.

- Based on the space temperature, space relative humidity, outdoor temperature, and temperature of representative structural member, or according to the computer software, the DDC unit controller starts supply and return fans at very low speed, such as 50 rpm through the adjustable-frequency variable-speed drives.
- The minimum outdoor damper, economizer damper, and exhaust damper are then opened fully, whereas the recirculating damper remains closed. The speeds of supply and return fans are gradually increased to design speeds.
- Once the measured $CO_{2r} - CO_{2o} < 100$ ppm, the purge mode is completed, and the DDC controller closes the minimum outdoor, economizer, and exhaust dampers, and opens the recirulating damper 100 percent.
- As $T_o > 75°F$ (23.9°C), the DDC controller sets the discharge air temperature set point at 52°F (11.1°C) and opens 100 percent the cooling coil valve.
- When the difference between the measured zone temperature and the summer indoor design temperature $T_r - 75 < 1°F$ ($T_r - 23.9 < 0.56°C$), the cool-down mode is completed.

If both cool-down and purge modes are required, and if the outdoor temperature $52°F \leq T_o \leq 75°F$ ($11.1°C \leq T_o \leq 23.9°C$), the DDC controller opens minimum outdoor, economizer, and exhaust dampers all to 100 percent, closes the recirculating damper, and operates the air economizer cycle in 100 percent outdoor air free-cooling mode. At the same time, it purges the conditioned space with outdoor air. When $T_r - 75 < 1°F$ ($T_r - 23.9 < 0.56°C$), the cool-down mode is completed; and when $CO_{2r} - CO_{2o} < 100$ ppm, the purge mode is completed.

3. As the temperature sensor measures the representative zone temperature $T_{rxn} > 75°F$ (23.9°C), the DDC controller calls for cooling. And if the outdoor temperature $T_o > 75°F$ (23.9°C), the VAV system is in minimum outdoor air recirculating mode, as shown in Fig. 23.7.

Based on the discharge air temperature reset schedule for cooling listed below, as the outdoor temperature $T_o > 75°F$ (23.9°C), the DDC controller resets the discharge air temperature $T_{\text{dis}} = 52°F$ (11.1°C).

T_o, °F	T_r, °F	φ_r, %	T_{cc}, °F	T_{mix}, °F	T_{dis}, °F	T_s, °F	$T_r - T_s$, °F
76–95	75	42–45	50	50–80	52	55	20
59–75	75	43–46	53	53–78	55	59	16
≤58	75 or less	48–51	56	56–78	58	63	12

Here T_r indicates zone temperature, φ_r zone relative humidity, T_{cc} off-coil temperature, T_{mix} mixed air temperature, T_{dis} discharge air temperature, and T_s supply temperature. For a VAV system using the same AHU that serves both perimeter and interior zones, T_{dis} only resets to 55°F (12.8°C) when outdoor temperature T_o drops belows 75° F (23.9°C) because a high space sensi-

ble cooling load in the interior zone during part-load operation in an office building needs cold air supply.

4. During minimum outdoor air recirculating mode, when the sensed zone temperature T_{rn} exceeds 75°F (23.9°C), DDC terminal controllers modulate the opening of single-blade dampers of the reheating VAV boxes in the perimeter zones and VAV boxes in the interior zones, to increase the zone, volume flow of supply air to maintain a nearly constant zone temperature $T_{rn} = 75$°F (23.9°C). During part-load operation, when T_{rn} drops below 75°F (23.9°C), because of a reduction in zone sensible cooling loads, DDC terminal controllers modulate the opening of single-blade dampers of the reheating VAV boxes in the perimeter zones and VAV boxes in the interior zones, to reduce the volume flow of the zone supply air to maintain a nearly constant zone temperature of 75°F (23.9°C), as discussed in Sec. 21.3.

5. During minimum outdoor air recirculating mode part-load operation, when reheating VAV boxes and VAV boxes close their single-blade dampers to smaller openings, the pressure drop across the dampers increases. This results in a higher static pressure in the supply main duct. As such, an increase in duct static pressure is sensed by a pressure sensor located at a distance of $0.3L_{\text{main}}$ from the farthest branch takeoff in the supply main duct, as discussed in Sec. 23.8; a DDC controller modulates and reduces the speed of the supply fan through an adjustable-frequency variable-speed drive to maintain a nearly constant duct static pressure set point, such as 1.2 in. WG (300 Pag), at the location of the duct static pressure sensor. Therefore, the volume flow rate of the supply fan is reduced to balance the reduction of zone supply volume flow rate during part load.

6. During the minimum outdoor air recirculating mode, if CO_2-based demand-controlled ventilation is used, the relationship between the supply flow density of minimum outdoor air (as well as the corresponding minimum outdoor damper opening) and the CO_2 concentration in the return duct (refer to Fig. 23.2) has already been determined during commissioning. Since as the supply fan is modulated at a speed to maintain a preset duct static pressure of 1.2 in. WG (300 Pag), the DDC controller opens the minimum outdoor damper to a setting to provide base ventilation $\dot{V}_{\text{bv}}$ first. According to the measured CO_2 concentration in the return duct, the DDC controller then resets the minimum outdoor damper opening setting according to the determined relationship at a predetermined time interval (such as 2 and 5 min).

Measured CO_2 concentration, ppm	Required minimum outdoor air, cfm/ft^2 (L/s·m^2)
≥ 800	$\dot{V}_{o,\text{sys}} = 0.14\ (0.7)$
≤ 400	$\dot{V}_{\text{bv}} = 0.07\ (0.35)$

The recirculating damper opens 100 percent, and the exhaust dampers open 30 percent.

For a VAV system with a supply and return fan combination, a pressure sensor is used to measure the representative space pressure p_r, in in. WG (Pag), and controls the space pressure in floating control mode. Only when $p_r > 0.03$ in. WG (+ 7.5 Pag) does the DDC controller increase the speed of the return fan, which raises the amount of exhaust air through the exhaust damper $\dot{V}_{eu}$ and maintains $p_r \leq + 0.03$ in. WG (7.5 Pag). Also only when $p_r < + 0.005$ in. WG will the DDC controller reduce the speed of the return fan to increase space pressure to $+ 0.005 \leq p_r \leq + 0.03$ in. WG ($+ 1.25 \leq p_r \leq 7.5$ Pag).

When the measured return duct CO_2 concentration is ≥ 800 ppm, the minimum outdoor air intake for a typical office building should be 0.14 cfm/ft^2 (0.7 L/s·m^2). Minus the exhaust through the restroom of 0.03 cfm/ft^2 (0.15 L/s·m^2), the remaining part is $0.14 - 0.03 = 0.11$ cfm/ft^2 (0.55 L/s·m^2). Such an amount of outdoor air is sufficient to maintain a space pressure of $+ 0.005 \leq p_r \leq + 0.03$ in. WG ($+ 0.005 \leq p_r \leq + 7.5$ Pag) and an associated exfiltration from an average leaky building as well as an amount of 0.015 cfm/ft^2 (L/s·m^2) of exhaust air through the exhaust damper.

Similarly, when the measured CO_2 concentration is $\leq$ 400 ppm, the minimum outdoor intake $\dot{V}_{o,\min} = \dot{V}_{bv} = 0.07$ cfm/ft^2 (0.35 L/s · m^2), and the remaining part is 0.07 − 0.03 = 0.04 cfm/ft^2 (0.2 L/s · m^2) which can provide a space pressure around + 0.005 in. WG (1.25 Pag).

7. During the minimum outdoor air recirculating mode, if a mixed plenum pressure control is used, either a pressure sensor located in the mixed plenum measures the static pressure difference between the mixed plenum and outdoors $\Delta p_{m,p}$, as shown in Fig. 23.4*a*, or a pressure sensor measures the pressure drop of the outdoor air damper and the louver, as shown in Fig. 23.4*b*. If the measured static pressure difference or pressure drop decreases and is less than a set point, such as − 0.4 to − 0.6 in. WC (− 100 to − 150 Pa), the DDC controller closes the recirculating damper until $\Delta p_{m,p}$ equals the set point and an amount of minimum outdoor air of $\dot{V}_{o,\text{sys}} = 0.14$ cfm/ft^2 (0.7 L/s · m^2) is extracted in the AHU. The minimum outdoor air damper is opened a fixed amount, such as 100 percent.

Mixed plenum pressure control provides an approximately constant minimum outdoor air with an amount of $\dot{V}_{o,\text{sys}}$. During minimum outdoor air recirculating mode, for VAV systems with a supply and return fan combination, the minimum outdoor damper opens 100 percent, the exhaust damper opens 30 percent, and the economizer damper is closed. A space pressure sensor measures the representative space pressure p_r, in in. WG (Pag). If it exceeds + 0.03 in. WG (7.5 Pag), a DDC controller increases the speed of the return fan through a variable-speed drive, and thus reduces the amount of minimum outdoor air intake as well as the space pressure and maintains p_r at less than + 0.03 in. WG (7.5 Pag).

When the zone load reduces, the speed and volume flow of the supply fan reduce accordingly. As the measured p_r drops below + 0.005 in. WG (1.25 Pag) because the return fan has not followed the supply fan to reduce its volume flow appropriately, the DDC controller then reduces the speed of the return fan, and therefore space pressure p_r increases and is maintained at a value not less than + 0.005 in. WG (1.25 Pag), as discussed in Sec. 23.6.

8. When the outdoor temperature 55°F $\leq T_o \leq$ 75°F (12.8°C $\leq T_o \leq$ 23.9°C), the AHU is operated in 100 percent outdoor air economizer cycle, as shown in Fig. 23.7*b*. The DDC controller opens 100 percent the minimum outdoor air damper, economizer damper, and exhaust damper. The recirculating damper is closed. When 100 percent outdoor air is used for free cooling, it overrides the minimum ventilation control.

The cooling and dehumidifying capacity of the chilled water cooling coil is also required at the same time. The DDC controller modulates the opening of the two-way control valve of the cooling coil to maintain a discharge air temperature $T_{\text{dis}} = 55$°F (12.8°C).

9. When the outdoor temperature $T_{o,\text{mix}} \leq T_o \leq 55$°F (12.8°C), the AHU is operated in the air economizer cycle with mixed air control. Here $T_{o,\text{mix}}$ is the outdoor air temperature at which the AHU or PU starts the mixed air control (such as $T_{o,\text{mix}} \approx 30$°F, or − 1.1°C) to provide a mixture of outdoor air and recirculating air at discharge air temperature T_{dis} directly, as discussed in Sec. 23.7. (Refer to Fig. 23.7.) So $T_{o,\text{mix}}$ depends on the percentage of outdoor air among the supply air and on the load ratio. In an air economizer cycle mixed air mode, a DDC controller opens 100 percent the minimum outdoor air damper and exhaust damper and modulates the economizer damper and recirculating damper to maintain a mixed air temperature equal to $T_{\text{dis}} = 55$°F (12.8°C) directly.

Since the amount of outdoor air intake in the AHU during mixed air mode is always greater than the amount of minimum outdoor air $\dot{V}_{o,\text{sys}}$ in recirculating mode, the mixed air control overrides the minimum ventilation control.

To provide cold air supply for the interior zone even in winter, the DDC controller resets $T_{\text{dis}} = 55$°F (12.8°C).

10. In the perimeter zone

- If the perimeter zone temperature T_{rxn} and interior zone temperature T_{rin} drop below 75°F (23.9°C), the terminal controller reduces the opening of the single-blade damper and thus the sup-

ply volume flow rate of cold conditioned air to the control zone and maintains a nearly constant zone temperature $T_{rxn} = T_{rin}$ 75°F (23.9°C).

- Further drop in the perimeter zone space sensible cooling load causes the damper in the reheating VAV box to close to a smaller flow passage until it reaches its minimum setting, such as 30 percent of the zone peak flow.
- When the temperature of perimeter zone n is 72°F $\leq T_{rxn} \leq$ 75°F (22.2°C $\leq T_{rxn} \leq$ 23.9°C), zone n is operated in the deadband mode, as discussed in Sec. 21.3. The damper in the reheating VAV box closes to its minimum setting, and neither cooling nor heating is provided except minimum ventilation air.
- When the temperature of the perimeter zone n is $T_{rxn} \leq$ 72°F (22.2°C), the zone temperature control is operated in heating mode operation. As discussed in Sec. 21.3, the DDC terminal controller actuates the reverse-action relay, and the cold primary air supply is still in minimum setting to provide minimum ventilation. Also the terminal controller modulates the opening of the two-way valve of the heating coil to maintain a preset zone temperature T_{rxn} = 72°F (22.2°C). The zone relative humidity may be between 25 and 50 percent. At winter design conditions, the warm air supply temperature should not exceed 87°F (30.6°C) in order to prevent the buoyancy effect.

Primary Considerations

- The sequence of operations of a VAV reheat system can be divided into two categories: sequence of operations for AHU or PU controls and for zone controls (including perimeter and interior zones).
- Sequence of operations for AHU or PU is usually varied based on outdoor temperature T_o and can be subdivided into

  Minimum outdoor air recirculating mode, cooling

  Air economizer cycle, 100 percent outdoor air free-cooling

  Air economizer cycle, mixed air control

  Minimum outdoor air recirculating mode, heating
- Sequence of operations for zone control is varied based on zone temperatures: perimeter zone temperature T_{rxn} and interior zone temperature T_{rin}. Zone temperature control can be subdivided into cooling, deadband, and heating mode operations.
- Minimum ventilation control is only valid to provide ventilation during minimum outdoor air recirculating mode between base ventilation $\dot{V}_{bv}$ = 0.07 cfm/ft^2 (0.35 L/s·m^2) and minimum outdoor air specified by ASHRAE Standard 62–1999, $\dot{V}_{o,sys}$ = 0.14 cfm/ft^2 (0.7 L/s·m^2) for a typical office building. During air economizer mode, outdoor air intake in the AHU or PU is always greater than $\dot{V}_{o,sys}$; there is no need of minimum ventilation control.
- For a VAV system with supply and relief fan combination operated in minimum outdoor air recirculating mode, if minimum outdoor air is provided by CO_2-based demand-controlled ventilation or mixed plenum pressure control, the space pressure is maintained between + 0.005 and + 0.03 in. WG (1.25 and 7.5 Pag) for a properly designed average leaky office building.

  For a VAV system with supply and return fan combination, when the speed of the supply fan is reduced during part-load operation, a DDC controller should reduce the speed of the return fan to maintain a space pressure between + 0.005 and + 0.03 in. WG (1.25 and 7.5 Pag).
- If a humidifier is used, be sure that the humidifying capacity is effectively controlled and the wet surface doesn't occur inside the AHU, PU, or supply ducts, to create indoor air quality problems.

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CHAPTER 24
IMPROVING INDOOR AIR QUALITY

24.1 IAQ PROBLEMS 24.1
24.2 VENTILATION, DILUTION OF CONCENTRATION OF CONTAMINANTS 24.2
Outdoor Air Requirements for Ventilation 24.2
Importance of Time of Operation and Ventilation Rate 24.2
Adopting an Air Economizer and Purge Operation 24.2
Select an Appropriate Minimum Ventilation Control System 24.3
Install a Minimum Outdoor Damper and an Economizer Damper 24.3
24.3 ELIMINATING MICROBIAL GROWTH 24.4
Basics 24.4
Contagious Respiratory Microorganisms 24.4
Microbial Growth 24.4
Eliminate Water Leaks, Prevent Dampened Surface and Material 24.5
Purge, Pressurization Control, and Ultraviolet Germicidal Irradiation 24.5
24.4 FILTERS TO REMOVE CONTAMINANTS 24.6
Remove Indoor Air Contaminants by Air Filters 24.6
Filter Selection for IAQ 24.6
Service Life of Air Filters 24.7
Filter Installation 24.7
24.5 REMOVING GASEOUS CONTAMINANTS BY ADSORBERS AND CHEMISORBERS 24.8
Indoor Gaseous Contaminants 24.8
Activated Carbon Adsorbers 24.9
GAC Performance 24.9
Simulating GAC Applications 24.10
Chemisorption 24.11
Chemisorber Performance 24.11
24.6 MAINTENANCE TO GUARANTEE IAQ 24.12
Inspection, Service, and Access 24.12
Monitoring of Operating Conditions in Air Systems 24.12
Coils and Ductwork 24.12
24.7 SPACE PRESSURIZATION CONTROL 24.13
24.8 LEGAL RESPONSIBILITY FOR IAQ CASES 24.13
SBS or IAQ Cases 24.13
Who Is Legally Responsible 24.13
HVAC&R Engineer 24.14
REFERENCES 24.15

24.1 IAQ PROBLEMS

The sick building syndrome (SBS) received public attention from 1970s after the energy crisis as a result of a tighter building and a reduced amount of outdoor ventilation air. Since Americans are spending more and more of their time indoors, they need a comfortable and healthy indoor environment and an acceptable indoor air quality. In an unhealthy building environment, uncomfortable employees do not perform well and productivity declines. Worker illness due to a poor indoor environment elevates absenteeism. A significant issue that is facing building owners, operating mangers, architects, consulting engineers, and contractors today is the possibility of legal suits filed by occupants or owners who feel that their health has been damaged by poor indoor air quality. In the 1900s, indoor air quality (IAQ) has become one of the primary concerns in air conditining (HVAC&R) system design, manufacturing, installation, and operation because of the following:

- IAQ is closely related to the health of the occupants inside a building, whether a building is a healthy building or a sick building.
- IAQ and thermal control (zone temperature and relative humidity control) indicate primarily the quality of the indoor environment in a building.
- IAQ and thermal control represent mainly the functional performance of an air conditioning (HVAC&R) system.

As described in Sec. 4.10, the indoor air contaminants are the basic pollutants that affect IAQ. According to the results of many field investigations and recent IAQ conferences, the primary IAQ problems and the appropriate mitigations and improvements are based on the three basic strategies: contaminated source control, removing contaminants by air cleaner, and ventilation air dilution, which are summarized and discussed in the following sections.

24.2 VENTILATION, DILUTION OF CONCENTRATION OF CONTAMINANTS

Outdoor Air Requirements for Ventilation

Persily (1989) took field measurements of the outdoor ventilation rates of 14 office buildings for approximately one year in 1983 and found that they are typically between 0.6 and 1.2 ach, with a mean value of 0.94 ach. Among these 14 office buildings 52 percent had a minimum level of outdoor air intake that was lower than the design level, 45 percent had less than 20 cfm/person (10 L/s·person), 8 percent had less than 10 cfm/person (5 L/s·person), and 1 percent had less than 5 cfm/person (2.5 L/s·person). ASHRAE Standard 62-1999 specified outdoor air requirements for ventilation for each occupant in various commercial facilities. Some of these requirements are listed in Table 4.5. Outdoor air requirements for ventilation per person $\dot{V}_{o,\text{req}}$, such as 20 cfm/person (10 L/s·person) in an office building, must be provided at both design flow and part-load reduced volume flow in VAV systems. From Eqs. (23.2) and (23.3), the calculated system outdoor air volume flow rate $\dot{V}_{o,\text{sys}} = N_{oc}\dot{V}_{o,\text{req}}$, in cfm (L/s). Here N_{oc} indicates number of persons, and $\dot{V}_{o,\text{sys}}$ is the minimum outdoor air intake at design flow in an AHU or a PU for the dilution of the concentrations of indoor air contaminants. For a typical office building with a ceiling height of 9 ft (2.7 m) $\dot{V}_{o,\text{sys}}$ is 0.93 ach, as listed in Sec. 23.2.

According to Mendell (1993), a statistically significant correlation between outdoor ventilation rate in offices and sick building syndrome complaints was found. Ventilation has been proved to be the most important factor to improve IAQ.

Importance of Time of Operation and Ventilation Rate

An often discussed topic is whether the type of HVAC&R system is of any significance that affects the aspect of ventilation and thus the SBS and IAQ. Sundell (1996), based on a study of 210 office buildings in northern Sweden, noted that the type of air system—whether it was a modern new HVAC&R system or an old simple system—had no significance on the aspect of ventilation and IAQ. Instead, the important aspects of ventilation were the time of operation and the ventilation rate. If the time of operation of the air system to provide ventilation was less than 10 h/day, the risk of SBS was increased. As the ventilation rate cfm/person (L/s·person) was reduced, the odds of SBS symptoms were raised accordingly.

Adopting an Air Economizer and Purge Operation

Select an AHU or a PU with an air economizer and purge mode control. As discussed in Sec. 23.11, if an air economizer is used, the amount of outdoor air intake in the AHU or PU during 100 percent

air economizer cycle and mixed air control mode is far greater than the calculated minimum air intake at design flow $\dot{V}_{o,\text{sys}}$. A 100 percent air economizer cycle significantly improves the IAQ.

If the difference in concentration of CO_2 between the conditioned space and the outdoors during morning time unoccupied period $\geq$ 200 to 300 ppm, or the difference in concentration of a specific indoor contaminant between outdoors and indoors is significantly greater than the normal operating period, a purge mode operation to exhaust all the space air with higher concentration of indoor contaminants and extract the outdoor air of lower concentrations of contaminants is beneficial to lower the concentration of the indoor contaminant.

Select an Appropriate Minimum Ventilation Control System

As discussed in Chap. 23, for a multizone VAV system that serves space with an obvious occupancy variation schedule and has low pollutants from nonoccupant-generated source, a CO_2-based demand-controlled ventilation (DCV) system should be used. In a DCV system:

- The amount of outdoor air intake during minimum outdoor air recirculating mode varies between $\dot{V}_{o,\text{sys}}$ at design occupancy and the base ventilation $\dot{V}_{\text{bv}}$ when occupancy reduces.
- Outdoor air requirements for ventilation per person, in cfm/person (L/s · person), meet the criteria specified in ASHRAE Standard 62-1999.
- Part of the cooling and heating energy to condition outdoor air is saved during reduced occupancy.

For a VAV system that serves space with a schedule of less occupancy variation, a mixed plenum pressure (MPP) minimum ventilation control system monitoring either plenum pressure or damper-louver pressure drop should be used. In an MPP control system, a fixed amount of outdoor air $\dot{V}_{o,\text{sys}}$ is extracted in an AHU or a PU for ventilation at both design flow and part-load volume flow during minimum outdoor air recirculating mode.

During commissioning, it is important to determine the relationship between the measured concentration of CO_2 in the return air duct (or on a representative floor space) and the minimum outdoor air intake expressed as the damper opening setting in a CO_2-based demand-controlled ventilation system, as discussed in Sec. 23.2.

Install a Minimum Outdoor Damper and an Economizer Damper

For better minimum ventilation control in a VAV system, it is recommended to install two outdoor dampers: a minimum outdoor damper and an economizer damper. The minimum outdoor damper is opened during minimum outdoor air recirculating mode. Its size is designed based on a volume flow rate of $\dot{V}_{o,\text{sys}}$, in cfm (L/s). The economizer damper is opened fully during 100 percent air economizer cycle. Its size is based on a volume flow rate $\dot{V}_{\text{ec},d}$, in cfm (L/s), and can be calculated as

$$\dot{V}_{\text{ec},d} = \dot{V}_s - \dot{V}_{o,\text{sys}} \tag{24.1}$$

where $\dot{V}_s$ = supply volume flow rate of the AHU or PU, cfm (L/s). The face velocity of either the minimum outdoor damper or the economizer damper is usually 1000 to 3000 fpm (5 to 15 m/s), and the ratio of the area of damper to the area of the duct of flow passage A_{dam}/A_d is often 0.5 to 0.9, as discussed in Sec. 5.7. The pressure drop of the damper when the damper is opened 100 percent Δp_{dam} can be calculated by Eq. (5.11) and often has a value between 0.2 and 0.5 in. WC (50 and 125 Pa) for minimum outdoor damper and economizer damper. During 100 percent air economizer cycle, both the minimum outdoor damper and the economizer damper are fully opened, and the pressure drops across the dampers are equal.

Outdoor air intakes should have wind shields and louvers, away from plumber stacks, and exhausts, with air filters or cleaners if the outdoor air quality is poor, and consider prevailing winds, as discussed in Sec. 16.2.

24.3 ELIMINATING MICROBIAL GROWTH

Basics

Microorganisms or microbes include fungi, bacteria, and viruses. Fungi and some bacteria form spores. Mold is minute fungi growing on moist organic substance. According to Kowalski and Bahnfleth (1998), disease-causing microorganisms transmit respiratory irritation and other serious health pathogens via the airborne route from two kinds of sources: Most contagious pathogens come from human hosts, and most noncommunicable pathogens come from the environment—the HVAC&R and building-related environment.

At the *ASHRAE Journal* Health Building/IAQ 1997 Conference, Dr. Bascom reported that "In the United States and Australia, asthma rates have nearly doubled in the past 15 years."

Contagious Respiratory Microorganisms

Kowalski and Bahnfleth (1998) noted that most contagious respiratory microorganisms induce their human host to aerosolize large quantities of infectious bioaerosols by irritation, which causes cough and sneezing. A single sneeze can generate 100,000 floating bioaerosols, and many carry viable microorganisms. A single cough typically produces only 1 percent of these bioaerosols but are ten times more frequent than sneezes. Keeping a distance from the infected person is a simple and useful method to reduce the risk of being infected.

Microbial Growth

Spores are characteristically larger and more resistant to impacts that will destroy viruses and bacteria. According to Kowalski and Bahnfleth (1998), noncommunicable diseases are almost entirely due to fungal or actinomycete spores and environmental and agricultural bacteria. Spores form the most important group in noncommunicable diseases. Microbes are commonly present in outdoor air. Normally, indoor air spore levels tend to be from 10 to 100 percent of outdoor spore levels when the species mix indoors reflects outdoors and are mostly less than 200 colony forming units (CFU) per m^3 because of transport from outdoors to indoors due to ventilation, infiltration, and people, and because of microbial growth indoors. In the California Healthy Building Study from Godish (1995), a naturally ventilated building had an indoor spore level of about 80 percent that of outdoors, mechanical ventilated building about 65 percent, and an air conditioned building about 15 percent.

Spores germinate and grow in the presence of water and nutrients—dirt and debris. The key factor is water. Water may exist inside a building or an HVAC&R system because of the following:

- Water leaks.
- Entrained water droplets.
- Wet coil face velocity exceeding 550 fpm (2.75 m/s).
- Wetted interior surface.
- Negative space pressure, due to improper airflow balance, extracts hot and humid outdoor air which contacts with the indoor cold surfaces and causes surface condensation.
- Clogged condensate pan and drain system.
- Oversized humidifier or improperly designed direct evaporative cooler.

The result is a "damp building" which is characterized by dampened materials, damp surfaces, mold, and microbial growth. *Legionnella* is a kind of pathogen transmitted via airborne route because the cooling tower water system is not properly treated and maintained.

Sundell (1996) noted that many studies have demonstrated a relationship between damp buildings and respiratory and allergic diseases in children and adults, and estimated that approximately 20 percent of asthma in children in Sweden can be attributed to damp housing.

Eliminate Water Leaks, Prevent Dampened Surface and Material

Always remember that water and dirt (dust and debris) are two necessities for microbial growth. Water leaks through the building envelope, especially through cracks and openings around outdoor air ducts, must be prevented. Pooling of water on the roof should be prevented.

In an HVAC&R system without a humidifier, the condensate flows over part of the outer surface of a water cooling coil or DX coil and collects in the condensate pan. This is often the only wetted surface inside the system that contacts the conditioned supply air.

As discussed in Secs. 15.10 and 15.11, to prevent dampened surfaces and damp material, the following is recommended:

- Choose a coil face velocity $v_a \leq 550$ fpm (2.75 m/s) for corrugated fins to prevent condensate droplets flying over.
- If the coil face has a vertical dimension exceeding 42 in. (1067 mm), an intermediate drain pan must be added to prevent condensate carryover.
- Install an individual drain pan and drain pipes for each coil. Drain pan may extend 1.5 to 3 times the coil depth in the direction of airflow.
- Condensate should be drained freely with a condensate trap, condensate collected in the condensate pan should be completely drained, and cleaned periodically. There should be no dirt and debris remaining in the condensate pan.
- Prevent surface condensation especially because of the infiltrated hot and humid outdoor air when the space pressure is negative.

In case a humidifier must be installed in an air conditioning system, the following is recommended:

- For most comfort air conditioning systems, select a steam humidifier or a heating element humidifier, instead of an atomizing or a supersonic humidifier, to prevent possible microbial growth.
- Modulation capacity control is essential. An accuracy $\leq +5$ percent relative humidity is recommended.

Purge, Pressurization Control, and Ultraviolet Germicidal Irradiation

Kowalski and Bahnfleth (1998) recommended the following HVAC&R measures to eliminate microbial growth and improve IAQ:

- If the concentration of a specific species of disease-causing microbe in outdoor air is substantially lower than that in the indoor air, a 100 percent outdoor air purge at a flow rate of 1 ach lowers the indoor concentration to 40 percent of the initial concentration level after 1 h of purge operation. If the purging airflow rate is 4 ach, the indoor concentration level drops to a few percent of the initial concentration level after 1 h of purge operation.
- Room pressurization control is often used in biohazard facilities and isolation rooms to prevent migration of microbes from one room to another. Room pressure is difficult to maintain and control, and is also expensive.
- The effectiveness of ultraviolet germicidal irradiation (UVGI) to inhibit fungal growth and kill the spores depends on the air velocity of the airflow passing the UVGI, airflow patterns, degree of

maintenance, resistance of the microbes, and ambient humidity. A single-pass airflow through a UVGI device may have only limited effect. However, recirculation airflow results in multiple exposures or chronic dosing.

24.4 FILTERS TO REMOVE CONTAMINANTS

Physically, indoor airborne contaminants are in solid, liquid, or gaseous form. Solid particulate contaminants include paper and fibrous particles, bacteria, fungus, spores, and viruses. Their size is between 0.003 and 100 μm. Liquid contaminants include mists, water, paints, cleaning sprays, and printing inks that typically have a size betweeen 1 and 50 μm. Gaseous contaminants such as CO, CO_2, and NO_2 and volatile organic compounds are fine particulates. Their size typically ranges from 0.003 to 0.006 μm.

Remove Indoor Air Contaminants by Air Filters

Basically, removing air contaminants by air filters, as discussed in Sec. 5.14, has the following primary purposes:

- To provide protection from respirable particulates, bioaerosols, and toxic and nuisance contaminants for occupants
- To protect the coils, ducts, and other air distribution flow passages from accumulated dust, dirt, and microbial growth
- To reduce cleaning and maintenance expenses

Airborne particulate matter is typically in the size range of 0.01 to 100 μm." Dust is defined as "an air suspension of particles (aerosol) of any solid matter, usually with particle size less than 100 μm."

However, particulates smaller than 10 μm in diameter may penetrate into the upper and lower parts of the respiratory passage and lungs. The indoor concentration level of particulates less than 10 μm must follow National Ambient Air Quality Standards as listed in Sec. 4.10. In addition, size of the particulates or indoor contaminants is an essential factor that affects the efficiency of air filters.

Filter Selection for IAQ

As discussed in Section 15.14, ANSI/ASHRAE Standard 52.2-1999 defines 16 minimun efficiency reporting values (MERV) for coarse-, low-, medium-, and high-efficiency air filters based on particle size efficiency E_1 in size range 0.3 to 1.0 μm, E_2 in size range 1.0 to 3.0 μm, E_3 in size range 3.0 to 1.0 μm and arrestance efficiency. Filters should be selected primarily based on particle size efficiency, the requirement of the conditioned space, pressure drop of the filter, service life, and the-life-cycle cost analysis. Considering IAQ, Burroughs (1997) and Liu and Huza (1995) select the following air filters to perform the specified duties:

- Employ medium-efficiency air filters MERV9 and higher with an $E_2 < 50$ percent up to $E_2 > 95$ percent in particle size 1 to 3 μm to control respirable particulates between 2 to 8 μm in size that may be breathed deeply into the lungs.
- Employ minimum air filtration level of MERV7 with a particle size efficiency of $50 \le E_3 < 70$ percent for a size range of 3 to 10 μm in an air system for the protection of the coil and air distribution system and to prevent the nutrition of microbial growth.
- Employ high-efficiency air filters MERV 13, 14, 15 with a low- or medium-efficiency prefilter; employ ultrahigh-efficiency air filters, HEPA and ULPA filters, with a medium-efficiency prefilter to extend service life of these filters.

Gill (1996) recognized that the use of a single low-efficiency filter in an AHU or PU has given way to a filter assembly with a prefilter and high-efficiency filter or at least medium-efficiency filters. Kowalski and Bahnfleth (1998) recommended that HEPA filters be used to remove microbes from the recirculating air. In a recirculating air system with an airflow rate of 1 ach, the microbial concentration level drops to 40 percent of its initial level after 1 h of operation. If the airflow rate is 4 ach, the microbial concentration drops to only few percent of the initial concentration level after 1 h of operation.

Service Life of Air Filters

When the media used in an air filter to remove air contaminants are clean, they are lower in efficiency. As the filter is loaded with more dirt, its efficiency gradually increases to a maximum. Then some particulates may have migrated through the media and be unloading back into the airstream for low- and medium-efficiency filters. At the same time, the pressure drop across the dirt-loaded air filter increases to a limit, and the amount of air passing through the air filter reduces to a point such that the filter doesn't pass through sufficient air for cooling or heating. This is often the end of the service life of the air filter, and it should be replaced by a new one. It is difficult to choose an optimum changing point between the increase of efficiency and the loss of airflow.

Ottney (1993) recommended the following service life of air filters as a reference for designers and operators.

Filter type	Typical change time	Max. pressure drop across air filter, in. WC (Pa)
Flat panel	30 to 60 days	0.5 (125)
2-in. (50-mm) pleated	3 to 6 months	0.9 (225)
4-in. (100-mm) pleated	10 to 14 months	0.9 (225)
6- to 12-in. (150 to 300-mm) cartridge	12 to 18 months*	1.5 (375)
21- to 36-in. bags	12 to 24 months*	1.5 (375)
HEPA	1 to 5 years*	2.0 (500)

*With prefilters

The following are useful measures to approach an optimum service life:

- As described in Sec. 23.9, a pressure sensor is used to measure the pressure drop Δp_{fil}, in in. WC (Pa), across each air filter. As soon as Δp_{fil} exceeds a predetermined limit, an alarm signal appears on the display module asking for replacement or maintenance. For a VAV system, Δp_{fil} should be measured when the airflow is approximately equal to the design airflow (such as the airflow during morning cool-down or purge mode operation).

  Sometimes the specified final pressure drop never reaches up because of (1) the blowout of the filters, especially holes developed in bag filters, (2) leakages from the filter hardwares, (3) leakages from the suction side of the supply fan, and (4) possible pressure sensor defects.
- Plan a preventive maintenance schedule, and change the air filters according to the planned schedule.
- Do not change the air filter just because it looks dirty on the surface. For panel filters, the filter should be replaced when the air leaving side shows dirt.

Filter Installation

In Ottney (1993), the following is emphasized:

- Air filters should be installed in gasketed channels. A fabric pile or polypropylene-finned gasket material provides a better seal. Poor-fitting and missing filler pieces are a primary factor that affects leakage.

- Many draw-through AHUs or PUs form a negative pressure region between the mixed plenum (including air filter) and the supply fan inlet. Any gaps in the casing, access panels, or doors downstream from the air filter are pathways for unfiltered air to enter the air system. All joints of the casing and access doors should be caulked or tape-sealed. Check all pipe and electric conduit penetrations for leaks. Improper fit and loose seating that allows a substantial amount of air leakage to bypass the filter are the most frequent problems observed.
- High-efficiency microfine fiberglass media of the bag filter have low tensile strength and are susceptible to punctures. When the media contact with sheet-metal screws or static pressure tips, it causes damage. If two adjacent pockets touch, abrasion results in a breach and they should be replaced. The bottom row of the bag filter may come in contact with condensate water in some filter banks. Wet filters grow microorganisms. These bags are either mechanically supported or changed to rigid-style filter to prevent wetness.
- Rigid extended surface filters are pleated in boxed or cartridge style. They are usually damaged by rough handling. The seal of the filter pack to the interior of its housing is essential.
- Panel-type, low-efficiency filters should be inspected to be sure that they are not installed backward. The air entering side is more open and dry, and the air leaving side is more dense and oily.
- An inspection of particulate pathways or leaks should be conducted each year and the observations recorded in a logbook.

24.5 REMOVING GASEOUS CONTAMINANTS BY ADSORBERS AND CHEMISORBERS

Indoor Gaseous Contaminants

Among the indoor air contaminants as discussed in Sec. 4.10, combustion products, volatile organic compounds (VOCs), smoke, objectable odors, and radon are all gaseous contaminants. Radon is an odorless, colorless naturally occurring radiative gas. Only about 6 percent of U.S. homes have annual average radon concentrations exceeding the acting level 147 Bq/m^3 (4 pCi/L) set by the U.S. EPA.

According to *ASHRAE Handbook 1997, Fundamentals*, for homeowners, a short-term charcoal canister is used as a screening technique to determine whether the long-term alpha track method is necessary. If long-term measurement (3 months to 1 year) finds that the average indoor radon concentration exceeding the acting level of 147 Bq/m^3 (4pCi/L), actions to reduce exposure to indoor radon concentrations must be taken. Exposures to indoor radon may be reduced by:

- Preventing radon entry into the building
- Removing or diluting radon and the radon progeny

An active subslab depressurization system is the most widely used effective method. This system uses a fan to draw soil gas containing radon and its progeny from pipework and exhausts the soil gas outdoors. Because of the depressurization of the soil gas containing radon beneath the floor slab, the indoor radon concentration may be significantly reduced.

Tobacco smoke is the most common indoor gaseous contaminant. There are two types of smoke: mainstream and sidestream. Mainstream smoke goes directly to the respiratory tract of the smoker. Sidestream smoke is the part exhaled by the smoker and smoke produced between puffs. Both mainstream and second-hand smoke may cause cancer. In many public spaces in the United States smoking is prohibited, and in many buildings, smoking is limited to special areas. Outdoor air requirements for ventilation in a smoking lounge are increased to 60 cfm/person (30 L/s·person) in ASHRAE Standard 62-1999.

Because gaseous indoor contaminants have very small molecules, gaseous adsorbers and chemisorbers are often used

- To eliminate or to reduce indoor gaseous contaminants such as combustion products, VOCs, and objectable odors
- According to the ASHRAE Standard 62-1999, to reduce the concentration of the known and specifiable contaminants in recirculating air to the required concentration level based on indoor air quality procedures to achieve acceptable air quality

Activated Carbon Adsorbers

Brown et al. (1994) reported that mean concentrations of VOCs in various office buildings ranged from 0.18 to 4.15 mg/m^3 with a new building having a total VOC concentration as high as 39.3 mg/m^3. For each individual compound, its mean concentration was about 0.05 mg/m^3. Granular activated carbon (GAC) is one of the most widely used adsorbents for indoor gaseous contaminants.

Adsorption (physical adsorption) is a surface phenomenona. In an adsorption process, the molecule of the gaseous contaminant

- Must be transported from the carrier airstream across the boundary layer that surrounds the adsorber granule
- Must diffuse into the pore of the adsorbent and occupy the surface
- Must be bound to the surface

Adsorption is different from absorption in which the gaseous contaminant is dissolved in or reacts with the absorbing medium, which can be either a porous solid or a liquid.

GAC Performance

As discussed in Sec. 15.16, removal efficiency, adsorption capacity, and service life are important performance parameters for GAC. Toluene has been used as a surrogate for indoor VOCs for the testing of GAC performance. An increase in relative humidity generally reduces the adsorption of VOC on GAC. This may affect some compounds more than others, and it becomes more significant at low VOC concentrations. When the concentration of the contaminant downstream of the adsorber rises until its concentration is the same as that of the upstream, the penetration at this time is called breakthrough penetration.

In addition to the adsorption capacity and the removing efficiency discussed in Sec. 15.16, the performance parameter breakthrough time t_b is defined as the elapsed time between the start of challenge and the time when the penetrating concentration reaches the specified breakthrough fraction, 10 percent breakthrough time $t_{10\%}$ or 50 percent breakthrough time $t_{50\%}$.

Another performance parameter is called capacity at t_b which is expressed as the ratio of the mass of contaminant collected at a particular breakthrough time to the mass of the GAC bed, in percentage.

Van Osdell and Sparks (1995) reported the performance of GAC challenged with toluene at various concentrations at 77°F (25°C) and 50 percent relative humidity as follows:

Challenge concentration ppm	0.44	1.1	9.2	71.7
Challenge concentration, mg/m^3	1.7	4.0	34	270
Carbon bed mass, g	24.1	23.1	23.4	25.8
$t_{10\%}$, h	625	344	72	11.9
$t_{50\%}$, h	750	422	88	15.0
Capacity at $t_{10\%}$, percent	6.6	9.2	16.2	18.8
Capacity at $t_{50\%}$, percent	7.9	11.3	19.7	23.8

Several test reports have estimated that the lifetime of GAC for indoor VOC control exceeds 1000 h depending on its concentration. The retentivity is the maximum amount of contaminant that the GAC can retain in clean air. The retentivity of GAC for toluene at 2860 mg/m^3 is 17 percent.

Simulating GAC Applications

Van Osdell and Sparks (1995) did a study based on an IAQ simulated model utilizing a computer program RISKBETA, which is a thoroughly mixed room model incorporating source/sink behavior that can predict concentrations as a function of time. The simulated model is a copy/storage room of 9.6 ft × 13.1 ft (3 m × 4 m) in a building that has eight rooms. The air system serving this building has a supply volume flow rate of 1177 cfm (2000 m^3/h) of 5 ach. It is assumed that 4.4 lb (2 kg) of toluene is spilled in the copy/storage room. One-half of the spilled solvent is cleaned up as a liquid; the other half vaporizes and is carried throughout the building by the air system. There is no other contaminative source in this building, and the outdoor air is assumed to be VOC-free. A GAC adsorber is installed in the air system to remove VOCs from the recirculating air.

Because of the buildup of the concentration of VOC contaminants in the indoor air, they are adsorbed into the sinks (paints, carpets, upholstery) and loaded into high levels. As the concentrations of toluene of indoor air drops and decays to below the 0.1 mg/m^3 level, the sinks become contaminative sources and reemit the toluene at low but significant rates.

Figure 24.1 shows the predicted concentrations within a copy room and the average concentration of the whole building with or without the GAC adsorbers based on the data and analysis of Van Osdell and Sparks (1995). The time between the beginning of the decay of the concentration of toluene in the copy room down to the 0.1 mg/m^3 level without a GAC adsorber is less than 10 days. The difference between the concentration decay curves with or without a GAC adsorber is comparatively small for the first two days and then begins to diverge because the GAC adsorber removes more contaminants than the sinks can reemit. Similar results can be found for the rest of the building.

As the concentration of contaminant continues to drop, the GAC will reach its retentivity limit for the incoming contaminant concentration at around 0.2 mg/m^3 and will cease to collect. In

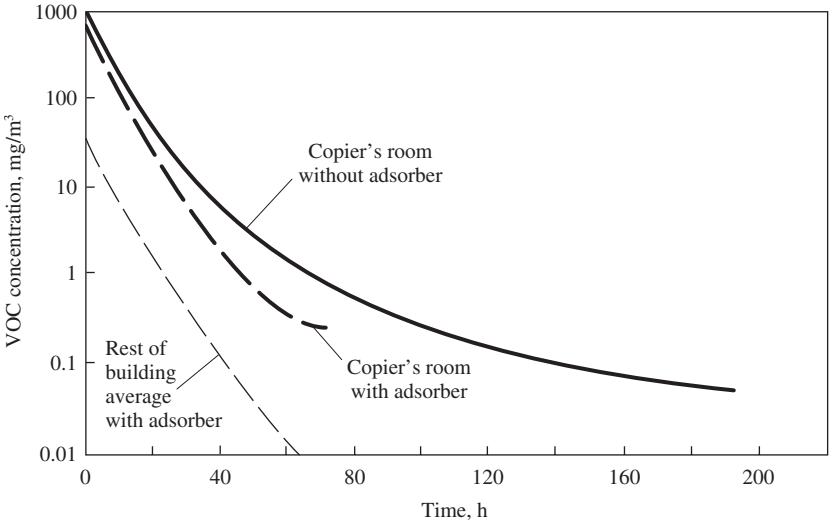

FIGURE 24.1 VOC concentration versus time decay curve for a simulated copy room.

Fig. 24.1, the sharp drop of the concentration decay curve will begin to parallel the curve that is without GAC adsorbers.

Because the building average concentration remains above 5 mg/m^3 for about 15 h, using GAC adsorber alone does not mitigate the spill. Better cleaning, short-term increased ventilation, a separate exhaust system for the copy room, or other similar measures should be taken to reach an acceptable level more quickly and to reduce the loading of the sources/sinks in the building.

Chemisorption

Chemisorption is a type of adsoption that occurs when the molecules of a volatile contaminant are first physically adsorbed onto the adsorbent. As soon as they are adsorbed, they react chemically with the chemical impregnant added to the surface of the adsorbent. The chemical impregnant makes the adsorbent more or less specific for a contaminant or a group of contaminants.

Chemisorption differs from the physical adsorption as follows:

- Chemisorption is generally irreversible whereas adsorption is reversible.
- Chemisorption improves when temperature increases; adsorption improves when temperature decreases.
- Chemisorption does not produce heat, but requires heat.
- Water vapor often helps chemisorption but usually hinders adsorption.

Most chemisorptive media are formed from a high porous support. Potassium permanganate-impregnated alumina (PIA) is the most widely used chemisorptive medium to remove gaseous contaminants. GAC does effectively remove most hydrocarbons, many aldehydes and organic acids, and nitrogen dioxides due to its high ratio of surface area to volume. PIA is effective against sulfur oxides, formaldehyde, nitric oxide, hydrogen sulfide, and lower-molecular-weight aldehydes and organic acids. PIA is often used in conjunction with GAC to cover a very broad spectrum of gaseous contaminant removal.

Chemisorber Performance

A chemisorber is an adsorber with chemically impregnated media. Most chemisorbers consist of chemisorptive media from a high porous support such as activated alumina or activated carbon coated or impregnated with a chemical reactant.

Three types of chemisorber cartridges are currently available: V-bank of large-mesh carbon trays, pleated dry composite media with fine-mesh carbon, and pleated nonwoven carbon-coated fabric. Removal efficiency, service life, and pressure drop are essential performance parameters.

Removal Efficiency. (See Section 15.16.) Removal efficiency depends on chemical impregnates, granular size, type of cartridge, and challenge concentration. Joffe (1996) reported that the removal efficiency of NO_2 measured at the beginning of a 2000 cfm (940 L/s) and 160-ppb challenge for a pleated dry composite medium with fine-mesh carbon chemisorber dropped from about 85 percent at the beginning to about 60 percent after 10 h of operation. For V-bank large-mesh carbon trays, the average removal efficiency is only about 30 percent in a period of 10 h of operation.

High removal efficiency is important for toxic and corrosive gases and becomes critical in odor control. According to *ASHRAE Handbook 1997*, *Fundamentals*, the human olfactory response S is nonlinear. The relationship between perceived intensities and the contaminant concentration C can be expressed as

$$S = k\,C^n \tag{24.2}$$

where k = a multiplying factor
n = exponential factor, varies from 0.2 to about 0.7

This relationship indicates that the concentration of some odorants must reduce substantially to make a perceived odor level change.

Upon exposure to contaminated air, the active surface is covered by absorbed pollutant molecules, or the product of a chemical reaction. This results in a decrease of removal efficiency over time.

Pressure Drop. There is always a compromise between higher performance and lower pressure drop across a chemisorber. At a face velocity of 500 fpm (2.5 m/s), the pressure drop is typically 0.4 in. WC (100 Pa).

24.6 MAINTENANCE TO GUARANTEE IAQ

Inspection, Service, and Access

Inspection, service, access, and maintenance to prevent system failure and component deterioration are essential factors to guarantee that air filters, coils, ducts, dampers, and other system components are properly and efficiently operated. Keep all the surfaces clean that surround the airflow passage and have contact with the outdoor air, recirculating air, and supply air, and without moisture except cooling coils, condensate pans, and evaporative coolers.

Access is the critical factor for inspection, service, and maintenance for an HVAC&R system. Many IAQ problems are the result of clogged coils or filters, blocked condensate pans, or carryover of the condensate droplets. If there is no convenient access to inspect the coil bank and filters, unless the AHU is partially dismantled, a small problem may become a disaster.

Hinged access doors to the filter section, fan sections, both sides of coil banks, condensate pans, and mixed plenum are essential. Standing water, mold and microbial growth, accumulated dust, and improperly operated sensors and instruments must be conveniently accessed, inspected directly and regularly, and promptly serviced.

The more difficult the access, the later the problem will be discovered, and less chance that the system component causing poor IAQ will be promptly found and improved.

Monitoring of Operating Conditions in Air Systems

The operating conditions that should be monitored and inspected include the following:

- Position of outdoor air dampers and minimum ventilation air controls
- Pressure drop across each filter
- Cleanliness of condensate drain pans in AHUs, PUs, and fancoils
- Cleanliness of coils
- Microbial growth in standing-water systems such as open-water system in condensate pans, evaporative coolers, and cooling towers
- Minimum setting in VAV boxes
- Optimum operation of exhaust or relief air system

Coils and Ductwork

Dust accumulated in coils and on the surface of the duct inner liner is best prevented by installing a medium-efficiency air filter (MERV7 with $50 \le E_3 < 70$ percent or higher) and without any air leakage bypassing the air filter. Ottney (1993) noted many hospitals installed a prefilter and a high-efficiency final filter on the air leaving side of the AHU. The coils and ductwork between the prefilter

and final filter in air systems with leak-free system components remained relatively free of visible particulate accumulation.

For a wet coil with condensate, a face velocity ≤ 550 fpm (2.75 m/s) should be selected for corrugated fins to prevent condensate carryover. If a fiberglass duct liner is used, a wet coil face velocity ≤ 500 fpm (2.5 m/s) should be selected to prevent microbial growth.

As discussed in Sec. 15.11, the condensate pan and condensate trap should be properly designed and installed. Condensate in the pan should be drained completely. Dirt, scale, and debris must be cleaned regularly.

For heating coils, if the coil leaks, the chemicals used in the hot water may be aerosolized and enter the air directly, causing IAQ problems.

24.7 SPACE PRESSURIZATION CONTROL

As discussed in Sec. 4.13, the purpose of space pressurization control is to maintain a positive pressure in an occupied space, a noncontaminated space, or a clean space to prevent untreated air, contaminated air, air with objectable odors, or toxic air entering or infiltrating this occupied space, and to maintain a negative pressure in contaminated or toxic area so that contaminated or toxic air (treated if required) exhausts outdoors directly. Air always flows from a place with a higher pressure to a place with a lower pressure. Space pressurization controls have been successfully applied to many laboratories, clean spaces and rooms, and industrial applications.

24.8 LEGAL RESPONSIBILITY FOR IAQ CASES

SBS or IAQ Cases

According to Eisenstein (1992), the 1980s saw litigation involving significant claims based on indoor air pollution, sick building syndrome (SBS), or IAQ. In the mid-1980s, there were homeowners and occupants of commercial buildings filing lawsuits claiming injuries from formaldehyde exposure. Cases filed by owners of mobile homes where large quantities of products containing formaldehyde are used have drawn much attention. There were also cases filed claiming personal injuries resulting from radon contamination and pesticide pollutants.

Who Is Legally Responsible

The most frequent cases of SBS or IAQ occurred in commercial office buildings with sealed windows in which occupants rely on the HVAC&R system for their thermal comfort control and ventilation requirements. A typical case is often caused by a combination of factors, including different toxic indoor contaminants from building materials, inadequate ventilation air supply due to defective ventilation control, and improper operation and maintenance of sophisticated air systems.

Eisenstein (1992) recognized that SBS cases have the potential of spreading a web of liability to a large number of professionals and industries. The prosecuting party, or plaintiff, is often an individual or group of individuals claiming personal injuries as a result of exposure to toxic substances in office suites. Another type of the SBS plaintiff is the commercial tenant who alleges that he or she cannot conduct business in the leased space because of the poor IAQ, that employees of the corporate tenant are frequently ill due to the poor IAQ, or that inadequate ventilation air causes business interruption and lost profit. Many of the SBS cases are multimillion-dollar lawsuits.

The list of defendants in SBS or IAQ cases varies with the type of buildings and may include some or all of the following:

- Building owner/managers
- Real estate developers
- Architects
- HVAC&R consultants, mechanical engineers, control engineers
- IAQ consultants
- General contractors, HVAC&R contractors
- Building and HVAC&R products manufacturers
- Leasing agents

Among them, the target defendants in SBS or IAQ cases are building owners who have the responsibility to provide safe premises, the building managers who must ensure that the building and HVAC&R system are maintained properly and the HVAC&R system is operated correctly, and all the entities that participate in the design, construction, and installation of the ventilation/HVAC&R system.

The theories of liabilities used against defendants depend to some degree on the law in various states. The following causes of actions have been successful in most states: contracts or breach of lease; professional malpractice or negligence; strict liability; fraud, mispresentation, and punitive damages.

HVAC&R Engineer

If the poor IAQ is determined to be due to defects in the ventilation/HVAC&R system, the mechanical or HVAC&R engineer who designs the ventilation/air system will be vulnerable to claims of negligence.

As discussed previously in this chapter, regarding HVAC&R, the primary tasks that must be done to ensure good IAQ are as follows:

- Provide adequate ventilation to dilute the indoor air contaminants.
- Prevent microbial growth.
- Use medium- and high-efficiency and HEPA filters to remove particulates, bacteria, and viruses.
- Remove specific gaseous contaminants by adsorbers and chemisorbers.
- Adopt pressurization control so that only filtered, noncontaminated air of acceptable air quality is supplied to the occupied zone, and contaminated air is exhausted directly from the contaminated area.
- Implement scheduled and proper HVAC&R maintenance for effective air system operation.

It is extremely important that the engineers' design follow the IAQ-related federal and local codes and the ASHRAE IAQ-related standards, especially ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality*. If an engineer fails to design an air system in comformity with appropriate ASHRAE standards, he or she will be unable to defeat a claim of negligence.

Although an engineer cannot be expected to foresee all the uses of the conditioned spaces during the life of the building, the changes in the occupancy rates and the alteration in the use of the space should usually be anticipated by the engineer, who must design a ventilation system to accommodate an adequate ventilation rate for high contamination load during remodeling of suites.

Recently, the courts have ruled that a ventilation system in a building is a product. Under strict liability, the designer, manufacturer, and installer of a defective product are considered liable regardless of whether their conduct was negligent. This theory emphasizes the product, not any individual, so that it is often easier for a plaintiff to succeed for a strict liability.

From the HVAC&R side, proper and careful consideration of the design, construction, and operation of an air system and the issues that may cause poor IAQ will greatly reduce the risk of SBS-related lawsuits. If an IAQ-related lawsuit still arises, the chance of a successful defense will be significantly increased.

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CHAPTER 25
ENERGY MANAGEMENT AND GLOBAL WARMING

25.1 ENERGY EFFICIENCY AND UNIT ENERGY RATE REDUCTION 25.1
- Energy Use and Energy Efficiency for HVAC&R Systems 25.1
- Energy Efficiency during Design, Construction, Commissioning, and Operation 25.2
- Reduction of Unit Energy Rates 25.2

25.2 GLOBAL WARMING 25.3
- Kyoto Protocol 25.3
- Total Equivalent Warming Impact 25.3
- Mitigating Measures 25.4

25.3 ENERGY EFFICIENCY 25.5
- Federal Mandates 25.5
- Energy Use Intensities 25.5
- Energy Audits 25.6
- Procedure for Energy Retrofit 25.6
- Performance Contracting 25.7
- Green Buildings 25.8
- Energy Star 25.10

25.4 ENERGY CONSERVATION MEASURES 25.10

25.5 CASE STUDY: ENERGY CONSERVATION MEASURES FOR AN OFFICE 25.12

25.6 RELATIONSHIP BETWEEN HVAC&R SYSTEM CHARACTERISTICS AND ENERGY USE 25.12
- System Characteristics and Energy Use Intensities 25.12
- Energy Use of Heating-Cooling Equipment and Fan 25.13

25.7 ELECTRICITY DEREGULATION 25.14
- Electric Utilities prior to Deregulation 25.14
- Electric Deregulation 25.14
- California Approach 25.15
- Real-Time Pricing 25.15
- Case Study: Automated Control of RTP 25.16

25.8 SYSTEM SIMULATION 25.17
- Energy Estimation and Energy Simulation 25.17
- Performance Equations 25.17
- Physical Modeling 25.18
- Steady-State and Dynamic Simulation 25.18
- Sequential and Simultaneous Simulation 25.19

25.9 ENERGY SIMULATION OF A CENTRIFUGAL CHILLER USING PHYSICAL MODELING 25.19
- System Model 25.19
- Operating Parameters Affecting Chiller Energy Performance 25.20
- Simulation Methodology 25.20
- Evaporator Model 25.20
- Condenser Model 25.22
- Cooling Tower Model 25.23
- Centrifugal Compressor Model 25.23

25.10 ENERGY SIMULATION SOFTWARE DOE-2.1E 25.25
- Energy Simulation Software 25.25
- Loads 25.26
- Systems 25.26
- Energy Efficiency Measures 25.27
- Plant 25.27

25.11 ASHRAE/IESNA STANDARD 90.1-1999 ENERGY COST BUDGET METHOD 25.28

REFERENCES 25.28

25.1 ENERGY EFFICIENCY AND UNIT ENERGY RATE REDUCTION

Energy Use and Energy Efficiency for HVAC&R Systems

Energy use or energy consumption indicates the amount of energy used or consumed. Energy efficiency indicates how efficiently energy is used. An energy-efficient HVAC&R system maintains a comfortable indoor environment with an acceptable indoor air quality (IAQ) and consumes optimum

energy resources. According to the ASHRAE definition, energy conservation indicates that energy is used efficiently. Energy management is the effort and measures taken to ensure that energy is used efficiently, and the unit energy rates (electric and gas rates) are reasonably low for the sake of reduction of energy cost. Energy management of HVAC&R systems consists of two areas: energy efficiency (the reduction of energy use or energy conservation) and the reduction of the unit energy rate.

As discussed in Sec. 1.7, the estimate of annual U.S. energy use of the HVAC&R systems in 1992 was approximately one sixth of the total national energy use. In addition, energy use for HVAC&R is closely related to the release of CO_2 to the outdoor atmosphere which causes the global warming effect. Energy efficiency is a challenge to every one of us in the HVAC&R industry now and for many years to come in the future.

According to EIA Commercial Building Characteristics 1992, energy sources used for the heating of the 67.8 billion ft^2 of commercial buildings in the United States in 1992 were electricity, 22.8 percent; natural gas, 51.8 percent; fuel oil, 6.6 percent; and district heating, 7.3 percent. Energy sources used for cooling U.S. commercial buildings in 1992 were electricity, 80.4 percent; natural gas, 2.8 percent; and district cooling, 3.0 percent. Electricity and natural gas are the two primary energy sources for HVAC&R systems.

Energy Efficiency during Design, Construction, Commissioning, and Operation

Energy efficiency must be achieved in every stage of HVAC&R system construction and operation: design, construction, commissioning, and daily operation. A well-designed and effectively functioning energy management direct digital control (DDC) system is necessary for an energy-efficient HVAC&R system.

- *Design*. Various alternatives should be compared and analyzed in terms of either payback years or life-cycle cost to determine which is most energy-efficient and cost-effective.
- *Construction and installation*. Roof and external walls should be well insulated. The efficiency of installed equipment should not be less than the minimum efficiencies specified by ASHRAE/IESNA Standard 90.1-1999. The amount of air leakages from the ducts, and the duct and pipe insulation all affect the energy use.
- *Commissioning*. The capacity of equipment, the air and water balance, and the coordination between various components and control systems should be carefully measured, adjusted, and commissioned. A poorly commissioned HVAC&R system will never function efficiently as specified.
- *Operation*. The energy use for chillers, compressors, fans, pumps, boilers, and furnaces should be monitored, periodically checked, investigated, reduced, and improved.

For HVAC&R, to reduce the emissions of CO_2 to the atmosphere by means of increasing energy efficiency in operation is the primary action to mitigate the global warming effect.

Reduction of Unit Energy Rates

Facility owners, facility managers, or the tenants as well as the designers and operators of the HVAC&R systems all are concerned about the unit energy rate E_r, as it affects the energy cost and the operating cost even if there is no saving in HVAC&R system energy use. For electricity, energy cost C_e is the product of the price of unit energy rate E_r and energy use, in kWh or therms. The lower E_r, the smaller C_e. The reduction of unit energy rate E_r is closely related to the following:

- The ratio of unit rate of kWh to therms which affects the designer in selecting electric cooling or gas cooling
- Use of thermal storage to provide off-peak conditioning to reduce E_r

- Choice of most favorable and optimum E_r rate from the electricity deregulation or gas deregulation

Energy cost is the dominant factor that affects the operating cost of an HVAC&R system.

25.2 GLOBAL WARMING

Kyoto Protocol

In December 1997, more than 150 countries met in Kyoto, Japan, and agreed to call for 38 developed nations to reduce the emissions of the greenhouse gases (GHGs) to about 5.2 percent below 1990 levels by 2008 to 2012, in order to mitigate the potential risk of global warming. The agreement is called the Kyoto Protocol and includes the following details:

- GHG emissions are to be reduced 8, 7, 6, and 6 percent below the 1990 level for the European Union, United States, Japan, and Canada, respectively.
- Six greenhouse gases are covered by the Kyoto Protocol: carbon dioxide (CO_2), methane (CH_4), nitrous oxide (N_2O), perfluorocarbons (PFCs), sulfur hexafluoride (SF_6), and hydrofluorocarbons (HFCs). And CO_2, CH_4, and N_2O will use the 1990 emission as the base value, whereas PFCs, SF_6, and HFCs can be compared with either the 1990 or 1995 emission as the base value.
- Currently, reduced emission controls are only applicable to developed nations. However, future negotiations will address the reduction of emissions and other issues for developing nations.
- Kyoto Protocol, using a basket approach, permits the limitation of emissions in any combination as long as its CO_2-equivalent target is met. The Kyoto Protocol also permits counting of benefits derived for "sinks" (forest, soil, and land used) for emissions trading. Details remain to be worked out.
- The Kyoto Protocol will be opened for signature from March 16, 1998, for one year and will enter into force after it has been ratified by at least 55 nations, representing 55 percent of the total emissions from the developed nations.

U.S. Ambassador Stuart Eizenstat, who led the U.S. delegation in Kyoto, said in the 9th Annual Energy Efficiency Forum, 1998, in Washington, "Kyoto is an insurance policy against the potentially devastating impacts of global warming. If we act now, the premium on this policy will be far more reasonable and less costly than if we delay and hope the problem goes away. The evidence shows that it will not."

In the 1998 ASHRAE winter meeting, an international panel of global climate experts concluded that the threat of global warming can be reduced if the world acts now to implement energy-conserving technologies in buildings and to reduce energy use. Experts also agreed that "energy efficiency" is technologically possible and economically viable.

Total Equivalent Warming Impact

The assessments of the global warming effect should be based on total equivalent warming impact (TEWI) concepts. In Baxter et al. (1998), total equivalent warming impact was defined as the combined global warming effects corresponding with the CO_2 released due to the indirect effect of energy use over the lifetime of a system and the direct effect resulting from lifetime refrigerant emissions. TEWI can be calculated as

$$\text{TEWI} = m_{\text{rt}}(\text{GWP})_{\text{rt}} + m_{\text{blow}}(\text{GWP})_{\text{blow}} + \alpha E_{\text{annual}} L \tag{25.1}$$

where m_{rt}, m_{blow} = mass of refrigerant and mass of blow agent released during its useful lifetime, lb (kg)

$(GWP)_{rt}$, $(GWP)_{blow}$ = equivalent global warming effect of CO_2 equivalent emisssion per lb (kg) of refrigerant or blow agent released, lb (kg)
α = conversion factor to convert energy use to global warming effect
E_{annual} = annual energy use
L = equipment lifetime, years

It is difficult to calculate an absolute value of TEWI because of the uncertainties of annual refrigerant leakage and GWP values. According to Baxter et al. (1998), for a typical supermarket in the United States with low- and medium-temperature DX refrigeration systems, estimated emissions due to refrigerant leakage are between 10 and 15 percent of the charge annually. If HFC-404A is used as the refrigerant, TEWI is about 8.6 million lb CO_2 equivalent. Of this, 57 percent is refrigerant emissions. For centrifugal chillers, the refrigerant leakage is between 0.5 and 4 percent of the charge annually. For a typical 1000-ton (3500-kW) centrifugal chiller in Atlanta, Georgia, where the refrigerant leakage is 2 percent of the charge annually, if HCFC-123 is used as refrigerant, the TEWI is about 5.1 million lb of CO_2 equivalent; of this, about 1 percent is due to refrigerant emissions. If HFC-134a is used as refrigerant, TEWI is about 5.9 million lb, and about 2.5 percent is due to refrigerant emissions.

Mitigating Measures

CO_2 Release due to Energy Use. David Gardner, asssistant administrator for the Environmental Protection Agency, who attended the International Climate Change Conference in June 1997, in Baltimore, reported that "85 percent of climate change problems result from combustion processes." For a combustion process using solid carbon C as the fuel, its reaction process and the released heat of combustion can be evaluated as follows:

Reaction	Heat of combustion ΔH (Q_{out}), Btu/lb (kJ/kg)	
$C + O_2 = CO_2$	− 14,100 Btu/lb (− 32,800 kJ/kg)	(25.2)
1 lb + 2.67 lb = 3.67 lb		

As expressed in Eq. (8.4), the combustion efficiency E_c, in percent, can be calculated as

$$E_c = \frac{100Q_{out}}{Q_{fuel}}$$

where Q_{out}, Q_{fuel} = heat output and heat content rate fuel consumed, Btu/h (W). For any combustion process and the successive energy transformation processes from heat to mechanical and then to electrical, and so on, in Eq. (8.4), if you require a specified Q_{out} for equipment in an HVAC&R system, for the sake of reducing the CO_2 in Eq. (25.2), if the fuel C must be reduced as well as the Q_{fuel} value in Eq. (8.4), there must be a corresponding higher combustion efficiency E_c. Not only higher E_c but all higher energy efficiencies reduce Q_{fuel} as well as the CO_2 and other greenhouse gases in the combustion process.

Refrigerant Emissions. HFCs are listed as one of the six greenhouse gases. Cox and Miro (1998) predict, by 2050, HFCs may account for 2 to 3 percent of GHGs. The measures to reduce the global warming effect because of the refrigerant emissions due to the leakage and purge are as follows:

- Refrigerant leakage in supermarket medium- and low-temperature systems should equal or not exceed the American Refrigeration Institute (ARI) target value of 6 percent of the charge annually. For chillers, a refrigerant leakage of 1 to 3 percent of the annual charge is appropriate.

- Secondary-loop systems using brine as the cooling medium and distributed systems in which compressors are close to the display case have far less refrigerant leakage than DX systems.
- The use of greenhouse gases as blow agents should be minimized.

The selection of environmentally-friendly refrigerant with low ozone depletion and global warming effect should be considered. From Table 9.1, compare the following refrigerants:

	Ozone depletion potential (ODP)	Halocarbon global warming potential (HGWP)
HFC-134a	0	0.28
HFC-404A	0	0.95
HFC-407A	0	0.47
HCFC-123	0.016	0.02

According to Baxter et al. (1998), the energy use of centrifugal chiller using HCFC-123a as refrigerant is only about 87 percent of that using HFC-134a as refrigerant. More research is required to determine a suitable alternative refrigerant for chillers when ODP, HGWP, and energy use are all considered at the same time.

25.3 ENERGY EFFICIENCY

Federal Mandates

According to Cox and Miro (1998), the U.S. government is the single largest energy user. In late 1990s, the federal government provided energy to approximately 500,000 buildings with a floor area of about 3.1 billion ft^2 (288 million m^2). Energy was needed in HVAC&R, lighting, and other building services. Approximately 77 percent of the floor area was used for housing, office, storage, and other purposes; the remaining 23 percent belonged to hospital, school, prison, and other uses.

Many federal buildings constructed before the energy crisis of 1973 are not energy efficient. The president's Executive Order 12759 in 1991 and the National Energy Policy Act of 1992 (EPAct) required federal buildings to reduce enery use by 20 percent from 1985 levels in the year 2000. In addition, Executive Order 12902 called for an energy efficiency improvement of 30 percent over the 1985 level by the year 2005.

The Federal Energy Management Program (FEMP) was established in 1974 to provide guidance and assistance to improve energy efficiency in implementing energy management plans. It has focused on the energy bill and potential for improvements. The Energy Policy Act established the fiscal year (FY) 1995 goal of improving energy efficiencies in federal buildings by 10 percent from 1985 levels on a Btu/gross ft^2 (W/m^2) of floor area. This goal is exceeded in FY 1994 with a total reduction in federal buildings of 11.2 percent. In addition, energy costs were $3.8 billion in FY 1994, and that was $1.5 billion less than in FY 1985. The federal government accumulated an energy savings of $9.8 billion and reduced federal building petroleum-based fuel use by 45.4 percent.

Energy Use Intensities

Energy use intensity (EUI) is the annual energy use per unit floor area, in $MBtu/ft^2 \cdot yr$ ($kWh/m^2 \cdot yr$). The EUI depends mainly on the locations of buildings, building characteristics, operating characteristics, and the HVAC&R system characteristics. According to DOE/EIA (1992), the abridged EUIs for commercial buildings in 1989 were as follows:

	MBtu/ft^2· yr			Electric, kWh/ft^2· yr				Natural gas, ft^3/ft^2· yr			
Building Type	Electric	Gas	Total	NE	MW	SO	WE	NE	MW	SO	WE
Assembly	27	39	64		7.1	9.5	7.3		57.5	25.9	45.8
Education	27	47	87	7.3	7.4	8.7	7.9	28.2	61.0	33.6	66.4
Health care	75	113	219	20.1	22.6	23.4	20.1	74.0	156.4	82.0	69.1
Mercantile/Serv.	44	46	85	15.1	12.5	12.1	13.4	62.9	62.1	29.4	31.1
Office	66	32	104	16.3	16.7	20.9	22.4	20.8	55.8	23.6	26.8

NE indicates northeast, MW midwest, SO south, and WE west.
Source: DOE/EIA (1992). Based on 4528 buildings. Reprinted by permission.

Energy Audits

Energy audit refers to a month-by-month accounting, survey, and analysis of energy use in a building. This energy use is checked against a budget or an energy estimate based on an hour-by-hour detailed system simulation in order to identify energy efficiency opportunities. According to *ASHRAE Handbook 1999, HVAC Applications*, following an energy use estimate, an energy analysis can proceed on three levels:

- Walk-through assessment analyzes energy bills and a brief survey of the building for the sake of identifying low-cost/or no-cost energy efficiency measures.
- Energy survey and analysis include a more detailed building survey and energy analysis with a breakdown of energy use within the building. Actual energy use can be compared with EUI of corresponding building types and locations. This level identifies and provides the savings and cost analysis of all practical energy efficiency measures that meet the owner's economic criteria.
- Detailed analysis of capital intensive modifications focuses on potential capital-intensive modifications and involves more detailed field data collection, system simulation, and engineering analysis.

In the detailed analysis level, an energy-efficient and cost-effective energy use estimate can be calculated according to the ASHRAE/IESNA Standard 90.1-1999 Energy Cost Budget (ECB) method. Refer to Sec. 25.11 for more information. The actual month-by-month energy bill can be compared against the ECB and energy cost breakdowns on the building envelope, load calculations, and HVAC&R system and components. Energy efficiency opportunities can then be determined.

Procedure for Energy Retrofit

Energy retrofit is a project whose purpose is energy efficiency, i.e., to convert an existing system to an energy-efficient system. Energy retrofit can often proceed as follows:

1. Organize a team. One individual should be chosen to be responsible for the team.
2. Draft a plan.
3. Make a field survey by walking through the building; talking with the operating manager, engineers, operators, and electric utility representative; take field measurements and tests, if necessary.
4. Proceed with an energy audit.
5. Justify the energy efficiency opportunities after cost analysis.
6. Implement the plan, starting with no-cost and low-cost opportunities.

7. After energy retrofit is complete, operate and control the HVAC&R system according to the requirements of the proposed energy efficiency opportunities. Check the energy costs against those from prior to the energy retrofit.

Performance Contracting

According to Mahoney and Weiss (1997), a performance contract legally guarantees energy and operating savings based on the performance of the contractor over a specified period of 5 to 25 years. Performance contracting also allows a facility to complete a major upgrade in energy efficiency funded by the saving of energy costs. The contractor must pay for savings that fail to materialize.

Energy Service Companies. During performance contracting, most often an energy service company (ESCO) initiates an HVAC&R system energy efficiency upgrade proposal to a facility owner tailored to the facility's needs. A performance contract includes design, installation, financing, project management, maintenance, and monitoring. Most of the components of a performance contract are the responsibility of the ESCO, and therefore, an experienced, technically competent person who has a good relationship with a financial party is the key factor for sucessful performance contracting.

There are three kinds of ESCOs:

- Engineering ESCOs are companies that always specialized to provide engineering services, including energy audits and system design. More often, engineering ESCOs also include project management, financing, and performance guarantees.
- Equipment ESCOs are manufacturers of energy users or controlling equipment. They establish their own energy efficiency service divisions to expand their marketing.
- Many utility ESCOs are owned by utilities which provide not only energy efficiency services, but also electric and natural gas power. There are also ESCOs aligned between utilities and consolidated HVAC&R contractors.

General Steps. Mahoney and Weiss (1997) noted that a performance contract works best when ESCOs and local contractor are well coordinated. The ESCO is reponsible for engineering, project management, guaranteed savings, and long-term project funding. The local contractor takes care of the installation, maintenance, and other services. All involved parties, such as facility owner, ESCO, local contractor, and equipment manufacturers, work well when the following general steps are taken:

- Conduct an energy use estimate. This involves assessing current conditions, collecting data to estimate potential savings, and determining factors affecting energy costs. Field surveys; interviews; review of building and HVAC&R system drawings, energy use of electric, gas, or oil for a number of years; occupancy schedules; and required indoor environmental parameters are important for an energy estimate.
- Perform a detailed engineering analysis. Computer modeling and simulation can be used to evaluate the current energy use as the baseline. Compare various alternatives, calculate and determine each possible energy-saving measure.
- Design the energy-saving measures for retrofit with reasonable payback time. Facility owner should approve an implementation plan prepared by the ESCO.
- Arrange project financing. After the approval of the implementation plan, the ESCO finds suitable third-party financing for the payment of installation and other costs. As the transaction costs are roughly the same for small, medium, and large loans and are expensive, a large or combination project often helps.

- Install approved equipment. The facility owner selects the equipment type and brand and decides whether a local contractor is needed or to use in-house staff for a small or medium-sized project.
- Commission the system and train operators. The ESCO verifies the operation characteristics of the new or modified system components and the HVAC&R system. The facility's staff learn how to operate new equipment, and the new or upgraded HVAC&R system.
- Monitor energy savings. The operation and control of the building's indoor environment and the energy efficiency-upgraded HVAC&R system remain with the facility's staff. The ESCO also monitors the building's indoor environment and the performance of the upgraded HVAC&R system remotely. The facility manager is notified if an abnormal condition occurs. On many occasions, the ESCO's monitoring service to ensure that energy savings are achieved should be included in the performance contract.
- Guarantee energy savings. If energy saving does not meet the projections, the ESCO pays the difference. The ESCO monitors and reviews energy use throughout the life of the contract. The contract also allows adjustments of the facility's baseline energy use when there is a change in the facility's size, occupancy, or equipment.
- Performance contracting provides the energy efficiency upgraded cost for the HVAC&R facility owners. Because the National Energy Policy Act of 1992 and Executive Orders 12759 and 12902 as well as utility companies offer fewer rebates, more and more federal building managers and facility owners in the private sector will rely on performance contracting as a viable way to upgrade the energy efficiency of HVAC&R systems. According to Mahoney and Weiss (1997), "Today, performance contracting projects in the United States have generated $2.3 billion in work and equipment. . . . The market is growing at a rate of 15 percent and has reached $750 million a year." There are more than 100 companies providing performance contracts. One of them, an engineering ESCO, has entered into more than 1000 agreements related to energy efficiency since 1974.

Case Study: Performance Contracting. *According to the Heating, Piping, and Air Conditioning* journal (January 1998, pp. 12–25), as the legislation permitted public school corporations to advertise for self-funding energy efficiency proposals from qualified performance contractors in 1995, the Evansville Vanderburg County School Corporation (EVCSC) in Evansville, Indiana, accepted a performance contract proposal from the Energy System Group (ESG), a utility ESCO, for energy efficiency upgrading of 35 school buildings. The ESG performance contract with EVCSC totaled more than $35 million.

The performance contract includes fully automatic summer/winter changeover two-pipe system in 19 buildings, pulse combustion gas boilers and DDC controls in 19 schools, and new lighting systems, windows, and doors. Based on the actual energy use records:

- The new boilers lowered the natural gas consumption 45 to 55 percent in the first year of use.
- Based on the original proposal, the energy saving program will save $369,000 per year. After 15 months of operation, it will actually save about half a million dollars annually.

Green Buildings

Basics. A green building implies a building, including HVAC&R systems, that is energy-efficient and environmentally friendly. It provides a healthy and comfortable indoor environment for its occupants, is friendly to the outdoor environment by releasing minimum pollutants into the outdoor atmosphere, and at the same time is considered sustainable, energy-efficient, and well maintained (refer to Sec. 1.8). Sustainable design considers resources consumed in a way that balances the current needs with those of future generations.

According to Cole and Larsson (1998), Green Building Challenge '98 (GBC '98) is a two-year international collaborative process of developing and demonstrating an improved method to assess

the enviromental performance of buildings. This process, initiated by Canada, has representatives from 13 other countries.

GBA Tool. The second-generation assessment method developed during the GBC '98 process is called the green building assessment (GBA) tool. GBA tool includes the following characteristics:

- It covers the following:

  Resource consumption: energy, land, water, and materials

  Enviromental pollutants: airborne emissions, solid waste, liquid waste, and others

  Quality of indoor environment: IAQ, thermal quality (such as indoor temperature and humidity), lighting quality, noise, and controllability of systems

  Longevity: sustainability, adaptability, and maintenance of performance

  Process: design, construction, commissioning, and operations planning

  Climate factors: locations, outdoor climate, sun, daylight, wind impacts
- It has a core set of detailed criteria and subcriteria that can be modified to suit national, regional, and building type variations. Assessments are based on applicable regulations or industrial norms in various regions. The structure can be used at various levels of detail.
- The tool reflects a consistency in terminology and a scoring system. All criteria and subcriteria are assessed on a -2 to $+5$ scale in which 0 is typical practice, 3 is best current practice, and 5 is a demanding, attainable goal.
- It comprises a weighting system that can be modified to emphasize regional priorities.

Case Studies of Green Buildings. The following is a description of the essentials of two green buildings: Gottfried et al. (1997) introduced San Diego's Ridgehaven Building (RB), San Diego, California, and Morrison (1998) reported on the Energy Resource Center (ERC), in Downey, California:

	RB	ERC
Building type	Office	Office-laboratory
Year constructed	1981	1950s, retrofit 1990s
Floor area, ft^2	73,000	45,000
HVAC&R system	WSHP (water-source heat pump)	Packaged system
Energy efficiency, $kWh/ft^2 \cdot yr$	9	22 percent less than code
IAQ OA, (outdoor air) cfm/person	20	
Filter efficiency, dust spot		40 percent
Daylighting		Maximized

Additional energy-efficiency and IAQ measures are as follows. For the Ridgehaven Building,

- Most material removed from the building had to be reused.
- Exterior foil, metallic film-coated foam, and exhaust systems are used to minimize glass fiber, VOC, and particulates entering the conditioned space.
- VAV systems and terminals are provided in variable-occupancy areas.
- Variable-speed drive for outdoor air fan and condenser water pumps are used.
- High-efficiency motors are utilized.

For the ERC:

- Direct-fired 30-ton double-effect absorption chiller-heaters
- Two dual-wheel desiccant dehumidification units

- AHU with evaporative coolers
- Variable-speed drive to save fan energy use
- Nontoxic products such as interior paints containing no petroleum derivatives, VOCs, and nontoxic floor sealers, padding, tile adhesives, and recycled carpeting material
- Linings in the ducts and plenums to prevent the growth of mold and bacteria
- Use of CO_2 -based demand-controlled ventilation

Energy Star

The U.S. Department of Energy (DOE) and the U.S. Environmental Protection Agency (EPA) are collaborating in rating to promote an energy-efficiency program for the recognization of

• EPA's Energy Star Benchmarking Tool assesses a facility's comparative energy performance against standardized indoor criteria and compares with similar-use buildings in United States according to Hoggard (2000). Each building is assigned a benchmark score from 0 to 100. All buildings scoring 75 or better qualify for an Energy Star label for buildings. Currently, the Energy Star label is only available for office buildings, and will be available for schools, retail stores, and other buildings later on.

- Equipment of the top 25 percent of products such as computers, printers, copiers, and HVAC&R equipment concerning energy efficiency with the Energy Star label.

Buildings with the Energy Star label are energy-efficient buildings, whereas green buildings are both energy-efficient and environmentally friendly.

25.4 ENERGY CONSERVATION MEASURES

In principle, adopting an integrated design approach by considering the building as a whole, minimizing heating and cooling loads, selecting high-efficiency equipment sized as closely to the design load as possible, emphasizing commissioning, and optimizing operation promote energy efficiency (energy conservation) in buildings. The following are possible energy conservation measures (ECMs) for the HVAC&R system in buildings:

- Turn off electric lights, office appliances, and other equipment when they are not needed.
- Shut down AHUs, PUs, fan coils, VAV boxes, chillers, fans, and pumps when the space they serve is not occupied or when they are not needed, except during warm-up, cool-down, or the purging period prior to the morning occupied period.
- The time chosen to start or to stop the AHUs, PUs, chillers, and exhaust fans daily should be optimum.
- The set point of space temperature, relative humidity, cleanliness criteria, and indoor-outdoor pressure differential should be optimum. Using different space temperature set points for summer and winter seasons for comfort air conditioning is often energy-efficient.
- The discharge air temperature from the AHU or PU and the temperature of water leaving the chiller or boiler should be reset for part-load operation based on the space temperature, system load, or outdoor temperature.
- Reduce air leakages from ducts, dampers, equipment, and HVAC&R system components. Use weather stripping to seal windows and revolving exterior doors to reduce infiltration. Ducts and pipes should be well insulated. Carefully design the layout of ducts and pipes to minimize their length, the number of duct and pipe fittings, as well as their pressure losses.

- Use more energy-efficient cooling methods such as free cooling, evaporative cooling, or groundwater cooling instead of refrigeration. Replace refrigeration with an air economizer, water economizer, evaporative cooler, and even desiccant dehumidification if doing so provides the same cooling results, is more energy-efficient and is cost-effective. An evaporative condenser is often more energy-efficient than a water-cooled or air-cooled condenser in many U.S. locations.
- Use heat recovery systems, waste heat from gas-cooling engines, or heat pumps to provide winter space heating when they are applicable and cost-effective.
- Use variable-air-volume systems instead of constant-volume systems, and variable-flow building loop water systems instead of a constant-flow one if the airflow or the water flow should be reduced during part-load operations. Large variable-speed aerofoil centrifugal fans are energy-efficient and often cost-effective in VAV systems.
- Use energy-efficient chillers, boilers, AHUs, PUs, and motors. Todesco (1996) recommended that chillers have an energy use index of 0.50 to 0.55 kW/ton, condensing boilers of 90 percent efficiency, and large supply and return fans of 65 percent total efficiency and higher.

ASHRAE/IESNA Standard mandates that minimum nominal full-load efficiency, in percent, for electric motors shall comply with the requirements of Energy Policy Act of 1992 as follows:

Motor horsepower, synchronous speed, rpm	Open motors, rpm			Enclosed motors, rpm		
	3600	1800	1200	3600	1800	1200
1		82.5	80.0	75.5	82.5	80.0
1.5	82.5	84.0	84.0	82.5	84.0	85.5
2	84.0	84.0	85.5	84.0	84.0	86.5
5	85.5	87.5	87.5	87.5	87.5	87.5
10	88.5	89.5	90.2	89.5	89.5	89.5
20	90.2	91.0	91.0	90.2	91.0	90.2
50	92.4	93.0	93.0	92.4	93.0	93.0
100	93.0	94.1	94.1	93.6	94.5	94.1
200	94.5	95.0	94.5	95.0	95.0	95.0

- Adopt a direct digital control (DDC) energy management and control system for large and medium-size HVAC&R new and retrofit projects.
- Adopt double-pane windows with low-emission coatings. Todesco (1996) recommended a U value of 0.25 Btu/h· ft^2· °F (1.4 W/m^2· °C) for exterior windows, also lower U values for external wall and roofs. Reduce the heat gain in the summer and heat loss in the winter in locations where the outdoor temperature is high in the summer and low in the winter.
- Use daylighting and controls for the perimeter zone. Also use energy-efficient fluorescent lamp and electronic ballasts with a goal to achieve connected lighting loads of 0.75 W/ft^2 (8 W/m^2) or better.

According to EIA's Commercial Building Characteristics (1994), in commercial buildings, the ratio of the floor area that had adopted the captioned energy conservation feature to the total conditioned floor area, in percentage, in 1992 in the United States was as follows:

Off-hours heating reduction	68.1 percent
Off-hours cooling reduction	63.0 percent
Energy management and control system (EMCS)	21.1 percent
Energy audit	21.8 percent
Variable-air-volume (VAV) systems	20.5 percent
Economizer cycle	27.0 percent
HVAC&R maintenance	72.4 percent

25.5 CASE STUDY: ENERGY CONSERVATION MEASURES FOR AN OFFICE

Parker et al. (1997) introduced an energy-efficient office building design for the Florida Solar Energy Center (FSEC) in the hot and humid climate of Cocoa, Florida. This office building consists of offices, a visitors' center, and laboratories and has a floor area of 41,000 ft^2 (3809 m^2). Laboratories were not included in this energy analysis. Eighteen energy conservation measures (ECMs) were considered during the energy analysis. Some of the ECMs are not effective for the hot, humid climate in Florida; e.g., the air economizer cycle showed little savings, and higher than code-mandated levels of wall and roof insulation resulted in little advantage. Only the following ten ECMs provided cost-effective energy savings.

- A high-efficiency lighting system of 0.9 W/ft^2 (9.7 W/m^2) with T-8 fluorescent lamps in a reflective troffer and with electronic ballasts saves energy use.
- Windows comprised of glazing units with high visible transmittance of 0.56 for daylighting and a low shading coeffficient of 0.33 reduce unwanted solar heat gain. Also a low U value of 0.31 $Btu/h \cdot ft^2 \cdot °F$ (1.76 $W/m^2 \cdot °C$) reduces conductive heat gain.
- Daylighting perimeter illumination coupled with solar control system on the south facade enables effective use of the perimeter office illumination. Dimming electronic ballasts are controlled by photometric sensors to adjust ballast output combined with the available daylight to maintain a constant desktop illumination level.
- Two high-efficiency screw chillers provide an IPLV of 0.65 kW/ton (COP = 5.41) and of 0.60 kW/ton (COP = 5.86) at 50 percent part load.
- Energy star personal computers, printers, and copiers are used to save energy.
- A reflective white single-ply roof membrane is chosen instead of a gray or black one, to reduce the solar irradiated heat gains through the roof.
- Use of occupancy sensors shuts off the VAV terminals when a room is not occupied.
- Using a DDC EMCS permits an increase in the cooling set point from 75 to 76°F (23.9 to 24.4°C) because of a finer control tolerance as well as provides optimal start and stop capability.
- Using variable-speed fans and variable-speed building pumps reduces the fan speed or pump speed during part load to save fan and pump energy use. Analysis showed that they are cost-effective.
- CO_2-based demand ventilation control is adopted for intermittently occupied spaces and zones.

In Florida, the average energy use of 160 state-owned office buildings in 1991 was 67 $MBtu/h \cdot ft^2 \cdot yr$ (211 $kWh/m^2 \cdot yr$). With these ten ECMs, the predicted energy use for the office building of FSEC will drop to 27 $MBtu/h \cdot ft^2 \cdot yr$ (85 $kWh/m^2 \cdot yr$).

25.6 RELATIONSHIP BETWEEN HVAC&R SYSTEM CHARACTERISTICS AND ENERGY USE

System Characteristics and Energy Use Intensities

As discussed in Sec. 25.3, for an HVAC&R (air conditioning) system, if the location of the building and the construction of the building it serves as well as its operating parameters are nearly the same, the energy use of this HVAC&R system varies as its system characteristics (types and configurations) change. According to the DOE/EIA 1998 nonresidential buildings energy consumption survey of commercial buildings' consumption and expenditures in 1995, the energy use intensity (EUI) of various HVAC&R systems is listed in Table 25.1.

TABLE 25.1 Energy Use of Various HVAC&R Systems

System characteristics	EUI, kBtu/ft^2 · yr (kWh/ft^2 · yr)
Heat pumps (including air-to-air and WSHPs)	28.1 (8.2)
Packaged systems (including VAV systems)	37.1 (10.9)
Evaporative systems	38.0 (11.1)
Individual systems	43.2 (12.7)
Single-zone, constant-volume packaged systems	44.0 (12.89)
Central systems (including VAV central systems)	44.7 (13.1)

In Table 25.1, WSHP represents water-source heat pump. Heat pump systems that include the air-to-air heat pumps and WSHP systems consumed the least energy because energy use in space heating was the lowest, as shown in the breakdown of heating, cooling, ventilation, lighting, office equipment, etc. below. Also, central systems consumed the most energy because the sensible cooling load of lighting, office equipment, etc. was about 50 percent higher in central systems than in packaged systems.

	Energy consumption, kBtu/ft^2 · yr					
	Heating	Cooling	Ventilation	Lighting	Office equipment	Others
Heat pumps	16.1	9.0	2.8	23.4	7.6	6.6
Packaged systems	26.0	7.9	3.2	24.6	6.7	6.4
Individual systems	36.0	5.5	1.7	16.5	3.9	6.6
Central systems	29.0	10.0	5.7	34.0	10.5	11.2

The energy consumption of cooling and ventilation in individual systems was the lowest. This indicated that the fan energy use was lowest with a lower fan total pressure and a poorer outdoor ventilation air supply than in other systems.

Energy Use of Heating-Cooling Equipment and Fan

As the heating energy used in an HVAC&R system mainly depends on the heating system characteristics, the efficiency of the direct-fired gas furnace and of the gas boiler is around 90 percent for the condensing furnace and boiler and 75 to 80 percent for conventional furnaces and boilers. The energy use of cooling equipment mainly depends on the characteristics of the refrigeration system.

For centrifugal chillers, it is assumed that energy-use for the centrifugal compressor is 0.55 kW/ton (6.39 COP), and the energy use of water pumps and cooling tower fans is 0.2 kW/ton. For packaged systems, the energy use of reciprocating or scroll compressors including the air cooled condenser fans is from 1.1 to 1.2 kW/ton.

The energy use of ventilation depends mainly on the ventilation rate and the fan characteristics. From Eq. (15.6), fan energy use P, in hp (kW), can be calculated as

$$P_f = \frac{\Delta p_f \dot{V}}{6356 \eta_f \eta_m \eta_d} \tag{25.3}$$

where Δp_f = system total pressure, in. WC (Pa)
$\dot{V}$ = volume flow rate, cfm (L/s)
$\eta_f \eta_m \eta_d$ = combined fan, motor, and drive efficiency

According to ASHRAE/IES Standard 90.1-1989, typically Δp_f and $\eta_f \eta_m \eta_d$ have the following values:

	Δp_f, in WC (Pa)	$\eta_f \eta_m \eta_d$
Constant-volume single-zone packaged system	2.5 (625)	0.40
Multizone packaged system	4.0 (1000)	0.45
VAV reheat packaged system	4.0 (1000)	0.45
VAV reheat central system	5.0 (1250)	0.55
Dual-duct VAV central system	6.0 (1250)	0.55
Fan-coil system	0.5 (125)	0.25
Water-source heat pump system	0.5 (125)	0.25

For both fan-coil and water-source heat pump systems, for each ton of refrigeration capacity there is about a corresponding 320 cfm (151 L/s) of volume flow for the small fan in the fan-coil unit and 80 cfm (38 L/s) of volume flow in the dedicated outdoor ventilation system. The dedicated outdoor ventilation system has often a total pressure loss of 5 in. WC (1250 Pa).

For VAV systems, the system total pressure often has a 1 in. WC (250 Pa) fixed pressure loss. The variable part will drop to an average of its 65 percent design system total pressure loss during part-load operation.

25.7 ELECTRICITY DEREGULATION

Electric Utilities prior to Deregulation

An electric utility is vertically integrated from three main components: generation, transmission, and local distribution network, as shown in Fig. 12.11*b*. There are usually four kinds of generation plants: high-efficiency coal and nuclear plants and inefficient diesel and gas-turbine plants.

Prior to the electricity deregulation, large facilities purchased electricity often based on the time-of-use (TOU) electric rate schedule that divides a 24-h working day into two or three periods, on-peak (such as from 9 a.m. to 10 p.m. weekdays) and off-peak periods (10 p.m. to 9 a.m. weekdays plus weekends and holidays) or on-peak, partial-peak, and off-peak periods. Higher rates are charged for on-peak hours than for off-peak hours. In addition, there is a demand charge and a monthly service charge based on the largest on-peak electricity power demand. In the 1990s prior to the electricity deregulation, unit electric rates ranged between \$0.02 and \$0.16 per kWh with an average around \$0.07 per kWh in the United States.

By using a time-of-use electric rate schedule, electric utilities tend to shift the electricity from on-peak hours to off-peak hours, and they shift the daytime peak operation of inefficient diesel and gas-turbine plants to nighttime base-load high-efficiency coal and nuclear plants to reduce operating costs.

Electric Deregulation

Electric utilites were originally regulated to prevent duplicated investments in electricity generation, transmission, and distribution networks to serve the same community. Electricity deregulation encourages open markets and competition and is generally considered as a means to reduce the cost of electricity and services. According to Warwick (1997), in the United States, deregulation began 20 years ago with telecommunications, gas, banking, airlines, and trucking. For electric deregulation, the Public Utility Regulatory Policies Act (PURPA) of 1978 required retail utilities to buy power from small generation plants developed by independent producers. This formed a competitive market in power generation and power wholesalers. In 1995, the Federal Energy Regulatory Commission (FERC) required utilities with bulk power transmission lines to provide open access to wholesale power buyers and sellers. As of 1997, two of the three major operations, generation and transmission, are open to competition as the result of the federal reforms. The next

phase of electric deregulation requires direct access to retail markets based on the condition that a competitive power market is provided.

California Approach

In late 1990s, several states followed California's lead in having legislation in place by setting a specific date for electric deregulation, such as Arizona in 1999, California in 1998, Massachusetts in 1998, New Hampshire in 1998, Pennsylvania in 1999, and Rhode Island in 1997.

In 1999, according to Warwick (1998), Waintroob (1998), and Gottfried (1997), the following holds:

- States in which electric utilities are deregulating typically adopt one of the following two approaches. The first approach is to provide electric consumers with an immediate rate cut but to retain regulatory control over power rates for the incumbent utility. Power-supplying competitors have to offer power below the new, discounted rates. In most states, the new rate is accompanied by a rate freeze or cap lasting for several years. The second approach is to separate the power cost from the old utility rate and to allow electric users to buy from alternate competitors.
- Power supply is the major electric utility service for competition under electric utility deregulation. California established a power exchange pool which acts as a power auction house and an independent (transmission) system operator, or ISO, which manages all power transfers and settles all power sales accounts when utilities continue to own and maintain their transmission lines.

  The experience of direct access in New Hampshire using bilateral trading allows individuals to negotiate for the best deal they can get. Those who do well have lower-cost good-quality electricity. Bilateral trades are contracts with receive and delivery terms. The buyer has to arrange transmission, if the contract is for power only.
- To implement competition without involving lawsuits by incumbent electric utilities and other jurisdictions that rely on the tax dollars generated from power plant valuations, electric utilities were offered means to recover their stranded costs within a certain period. In California, the state's degeneration bill AB 1980, enacted in September 1996, calls for a nonbypassable stranded cost surcharge of approximately 2.5 to 6 cents per kWh payable over 4-year period until 2002 by rate payers. In 2002, California consumers will realize the benefits of deregulated rate. Stranded cost is the cost difference between a plant cost carried on its books and the real market value.

Real-Time Pricing

Recently, the use of the computer and data handling software, the improvements in telecommunications, as well as the demand for better energy management following electric deregulation have made it possible for electric utilities to offer to customers an electric rate structure that more precisely represents the actual cost of providing the electric service. In addition, this electric rate structure also depends on the climate and demand; varies hourly, 24 h daily and 8760 h annually; and is known as real-time pricing (RTP). It began in 1985 in California. Many states allowed electric utilities to offer RTP electric rate structure to commercial and industrial customers in the late 1990s. Other rate structures will also be available in a deregulation environment as the power providers maneuver to offer the most competitive plans possible.

According to Bynum (1998), the components of a typical bill, in percentage, in the late 1990s are as follows:

Cost of generation	40 percent
Transmission, local distribution, service, profit, etc.	60 percent

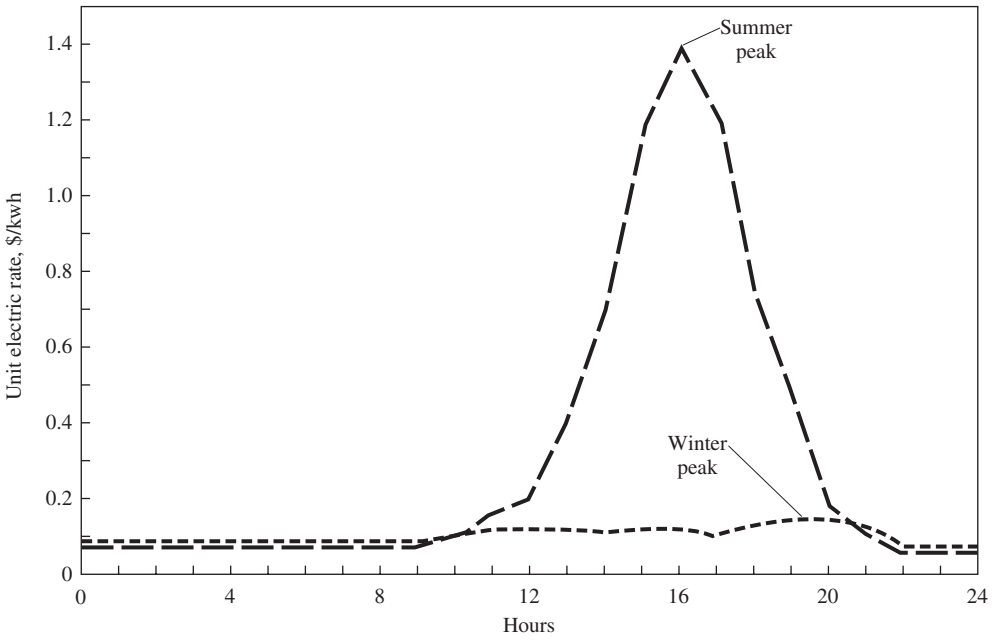

FIGURE 25.1 Real-time pricing (RTP) unit electric rate daily range at summer and winter peaks.

Figure 25.1 shows the daily range of RTP electric rates at summer and winter peaks. Because of the use of the RTP electric rate structure instead of the TOU rate structure following electric deregulation, Norford et al. (1998) noted the following:

- The need for such crude price signals as the demand charge is diminishing.
- Commercial and industrial facility operators have the opportunity to rationally evaluate the tradeoffs and the possible reduction or shifts in building services, such as lights, heating, cooling, and process loads. The general strategy of shifting loads from high-priced to low-priced hours is common to both TOU and RTP approaches; however, RTP has a much greater range of price variation than TOU.
- Knowledge-based system computer programs are used for RTP cost optimization control. RTP control strategies can be summarized thus: use of thermal storage systems; utilization of fuel switching of on-site auxiliary or emergency generation or cogeneration, as well as gas cooling such as absorption chillers or gas-engine-driven chillers; general load shedding of lights and fans; and batch scheduling of production processes.

Case Study: Automated Control of RTP

According to Gabel et al. (1998), the first automated RTP control was installed in the 51-story Marriott-Marquis Hotel with 1.8 million ft^2 (167, 200 m^2) in New York City. The mechanical systems include three 900-ton (3170-kW) centrifugal chillers and many constant-volume and VAV systems. Heating is provided by purchased steam. The 1900 guest rooms were excluded from RTP control for primary room comfort. About 70 air-handling, ventilation, and lighting systems for the remaining 55 percent of floor space were placed under RTP control.

The control strategy for each system or equipment has criteria that must be met before its load can be shed, reduced, or rescheduled. Here is an example of control logic. If the RTP price is greater than $0.10 per kWh, if the AHU AC-2 is in an occupied period, and if the space temperature AC-2 serves is not over 80°F (27°C), then command AHU AC-2 off. If the above control parameters are not met, AC-2 continues to operate.

The electric utility provided day-ahead notice of upcoming hourly RTP rates downloaded via dial-up phone line. The RTP control system responds to the rate schedule hour by hour, first by analyzing its preprogrammed control strategies and then by executing the most appropriate events in sequence. The control system also reevaluates the control logic for each controllable load each hour, making adjustments if necessary. About 70 air system loads were configured for RTP control.

During the three most recent years with TOU electric rates, the electric energy use averaged approximately 38 million kWh per year, with a peak electric load of 6 MW. The electric bill averaged $3 million per year. With automated control response to RTP rates:

- The electric bill saved $1 million in 4 years.
- Average energy use dropped more than 2 million kWh each year.
- Peak electric load reduced 1500 kW during high-price periods.

25.8 SYSTEM SIMULATION

Energy Estimation and Energy Simulation

Annual, seasonal, or short-term estimation of energy use in an air conditioning or HVAC&R system is an important tool to assess energy performance and cost in both design and daily operations. The annual energy use of various types of air, water, refrigeration, and heating systems can be predicted and analyzed. Based on the results of energy estimation, the most energy-efficient system can be selected.

For the daily operation of air conditioning systems in existing buildings, detailed simulation reveals the system's negative and positive effects on energy use and performance by comparing the estimated evaluation against the actual measured results. The energy use of many alternative improvements can be predicted and assessed. These estimates provide great potential for reducing the energy use and operating cost of existing air conditioning systems.

The detailed simulation method uses a comprehensive computer program through a powerful personal computer (PC) to simulate the thermal behavior of the building, and the performance of the air, water, refrigeration and heating plants at various outdoor weather conditions, to estimate the energy use of an air conditioning system. Because of the powerful capability of the computer program, detailed simulation methods can include most of the important factors that affect system energy use. The most widely used program is the year-round, hour-by-hour energy simulation software. Short-term simulation is sometimes used in dynamic modeling to evaluate the influence of the thermal storage effect of system and components on their performance.

Energy simulation is the representation of energy use of an actual system or component by a model of analogous characteristics, and it is used to predict the performance and operating parameters of a system or component. Mathematical equations are used to describe the operating characteristics of the working substance and work and energy transfer of the system or component being simulated. Energy use simulation can be represented by either performance equations or physical modeling.

Performance Equations

Generally, a system component's performance with regard to one or two independent operating variables is often available from the manufacturer's catalog. A polynomial expression is used to fit

the catalog data or actual performance into a regression equation, to mathematically relate the dependent variable z to independent variable x in steady-state simulation as follows:

$$z = a_0 + a_1x + a_2x^2 + a_3x^3 \tag{25.4}$$

To relate a dependent variable with two independent variables x and y, the polynomial expression is

$$z = b_0 + b_1x + b_2x^2 + b_3y + b_4y^2 + b_5xy + b_6x^2y + b_7xy^2 + b_8x^2y^2 \tag{25.5}$$

where $a_0, a_1, \ldots, a_3, b_0, b_1, \ldots, b_8$ = coefficients. Computer programs are available to solve the coefficients according to the data from the manufacturer or actual performance data.

Physical Modeling

Modeling. Setting up a component or a system model is the first step of energy simulation by physical modeling. Modeling includes the following:

- Description of system or component configuration whether it is an air, water, refrigeration, or heating system, or whether it consists of many components or only contains a single device.
- Description of the operating characteristics of the system or component, and the interaction between system components, whether it can be simplified to a steady-state model or a dynamic model. A simplification of the physical model that results in an error of only few percent of the final result is recommended, in order to simplify the calculation and analysis.

Developing Mathematical Equations. Mathematical equations are developed to describe the operating characteristics of the working substance, and work and energy transfer.

Solving for Outputs. Computer programs are used to solve equations simultaneously, in sequence, or by iteration. The required operating parameters, the outputs, can thus be obtained. Sometimes performance equations are used to link the required outputs with one or two operating parameters.

Verification. The results of predicted performance during energy simulation can be verified against actual measured readings of similar models and operating conditions. According to *Scientific Computing* (1997), using performance equations to simulate the energy use of an HVAC&R system or component is simple, and lumps the capacity, energy use, or other required operating parameters in one performance equation. The disadvantage is that the performance data must be provided to create the performance equations. Physical modeling allows more freedom in configuring the characteristics of the system or equipment.

Steady-State and Dynamic Simulation

According to its operating characteristics, energy simulation can be classified as steady-state or dynamic simulation.

Steady-State Simulation. In steady-state simulation, the relationship between various operating parameters within a certain time increment is described by mathematical equations independent of time, such as

$$z = F(x, y, \ldots) \tag{25.6}$$

where x, y, z = operating variables that are not a function of time. Steady-state simulation always simulates relatively long-term system or component characteristics, such as annual or seasonal energy simulation. A time increment of 1 h is typically used for analysis. Within the time increment, the operating parameters are independent of time. However, the magnitude of the same operating parameter may be different in successive hours.

In steady-state simulation, the heat capacity of the working substance is often far greater than the heat being absorbed and released from the equipment, pipes, ducts, and surroundings when the temperature of the working substance fluctuates. Therefore, heat absorbed by the equipment, pipes, and ducts and the heat transfer to or from the surroundings are often ignored. An hour-by-hour, year-round energy simulation computer program is an example of steady-state simulation.

Dynamic Simulation. In dynamic simulation, the relationship between various operating parameters is described by a mathematical equation that is a function of time

$$z = F(x, y, t, \ . \ . \ .) \tag{25.7}$$

where t = time variable. Dynamic simulation is mainly used to simulate short-term characteristics. Its purpose is to analyze the interaction between the control actions and the dynamic response, and the step change of an operating parameter on system performance. A time increment of minutes or even seconds is required to analyze the rapid response of the system during simulation. In dynamic simulation, the heat being absorbed and released from the equipment, instrument, pipe, and ducts is often taken into consideration. The variation of zone temperature and supply volume flow caused by a step change of the zone lighting load is an example of dynamic simulation.

Sequential and Simultaneous Simulation

When a system comprises many components linked together by a working substance and forms an open circuit, the calculation can be started at a component whose output is the input of the next linked component, and can progress in sequence to the component where the final result can be obtained. Such an approach is called sequential simulation.

In sequential simulation, the output of a successive component usually does not affect the output of the former component already calculated. If the output of a component does affect the output of the former component already calculated, a computing loop is used until the calculated output is equal to the assumed value.

The fuel input rate to a direct-fired gas heater is an example of sequential simulation. Calculation starts from the heating requirement at the warm air heater. The next calculation is mass flow and the temperature drop of the combustion gas, and finally the rate of fuel gas supplied to the burner, which is the required result, is then calculated.

When the working substance that links the components together forms a recirculating closed loop and many operating parameters are interrelated, all unknown operating parameters should be solved simultaneously. A simultaneous simulation is required. The power input to a compressor in a reciprocating compression refrigeration cycle is an example of closed-loop simultaneous simulation.

25.9 ENERGY SIMULATION OF A CENTRIFUGAL CHILLER USING PHYSICAL MODELING

The following illustrates the detailed energy simulation processes of a centrifugal chiller for the sake of calculating its power input and thus its accumulated annual energy use using physical modeling.

System Model

The centrifugal chiller has a two-stage compressor using HCFC-123 as refrigerant. Its refrigeration cycle is the same as shown in Fig. 9.7. The centrifugal compressor is driven by a hermetic motor of variable-speed drive. After compression, the refrigerant is discharged to a shell-and-tube water-cooled condenser. A flash cooler or economizer is installed between the condenser and the shell-and-tube flooded evaporator. Multiple orifice plates are used as the throttling device between

condensing pressure p_{con} and intermediate pressure p_{int}, and between p_{int} and evaporating pressure p_{ev}. Water from the cooling tower is used as the condenser water. A proportional-integral (PI) DDC panel is used to maintain a nearly constant temperature of chilled water as it leaves the evaporator.

During part-load operation, the DDC unit modulates the speed of the compressor to reduce both the mass flow rate of refrigerant $\dot{m}_r$ entering the compressor and its corresponding system head, to create a new mass and energy balance.

The energy simulation of this centrifugal chiller is considered at steady state within a time increment of 1 h. Sequential simulation, iterations, and perfomance equations are used to calculate the final results. Many simplifications are adopted to reduce the calculations during simulation.

Operating Parameters Affecting Chiller Energy Performance

Operating parameters that affect the energy performance of this centrifugal chiller are as follows:

- Refrigeration load ratio R_{load} is defined as the ratio of the operating refrigeration load Q_{ev} to the design full-load $Q_{ev,d}$. Ratio R_{load} affects the rate of heat transfer at the evaporator and condenser. It also affects the efficiency of the two-stage compressor.
- Temperature of condenser water entering the condenser T_{ce} is a function of outdoor wet-bulb temperature T_o' and the performance of the cooling tower. Temperature T_{ce} is closely related to the condensing temperature T_{con} in the condenser, and therefore the pressure lift $\Delta p = p_{con} - p_{ev}$, the temperature lift $\Delta T = T_{con} - T_{ev}$, and the power input to the compressor P_{in}.

Simulation Methodology

From Eq. (13.5), power input to the compressor motor $P_{in,m}$, in hp, can be calculated as

$$P_{in,m} = \frac{\dot{m}_r W_{isen}}{2545 \eta_{cp} \eta_{mec} \eta_{mot}} \tag{25.8}$$

where $\dot{m}_r$ = mass flow rate of refrigerant, lb/h
$\eta_{cp}\eta_{mec}\eta_{mot}$ = compression efficiency, mechanical efficiency, and motor efficiency of compressor

Work input W_{in} or W_{isen}, in Btu/lb, is directly proportional to the enthalpy difference of the refrigerant between condensing and evaporating pressure. To find W_{in}, it is necessary first to determine the condensing pressure p_{con}, the condensing temperature T_{con}, the evaporating pressure p_{ev}, and the evaporating temperature T_{ev}. So the following information is required:

- An evaporator model is required to determine T_{ev} at various operating load ratios R_{load}.
- A condenser model is required to determine T_{con} at various R_{load} and outdoor wet-bulb temperatures T_o'.
- Calculation of isentropic work input W_{isen} to the compressor is required based on the two-stage refrigeration cycle at various R_{load} and T_o' values.
- The actual power input to the compressor motor based on the compressor model and the annual energy use is finally calculated.

Evaporator Model

From Eq. (13.7), the rate of heat transfer at the tube surface of the evaporator or the refrigeration load Q_{ev}, in Btu/h (W), can be calculated as

$$Q_{ev} = A_{ev} U_{o,ev} \Delta T_{ev} \tag{25.9}$$

From Eq. (13.8), the overall heat transfer coefficient based on the outer surface area of the copper tubes $U_{o,\text{ev}}$ in Btu/h·ft^2·°F (W/m^2·°C), is given as

$$U_{o,\text{ev}} = \frac{1}{1/(\eta_f h_o) + A_o R_f / A_i + A_o/(A_i h_i)} \tag{25.10}$$

The boiling coefficient for HCFC-123 is slightly lower than that for refrigerant CFC-11. From Eq. (13.9), for a shell-and-tube flooded-type evaporator, the boiling coefficient of HCFC-123 can be evaluated as

$$h_o = C_b \left(\frac{Q_{\text{ev}}}{A_{\text{ev}}} \right)^n$$

According to the experimental results in Webb and Pais (1991) and Jung and Radermacher (1991), for HCFC-123 in copper tubes with integrated fins of 26 fins/in. (0.98 mm), constant C_b can be taken as 2.5 and exponential index n is approximately 0.7. For enhanced surfaces, according to the results of Webb and Pais (1991), h_o is 35 percent higher.

The copper integrated-fin tubes currently used for evaporators usually have a fin spacing of 19 to 35 fins/in. (1.3 to 0.73 mm), typically 26 fins/in. (0.98 mm). The outside diameter of copper tubes varies from $\frac{5}{8}$ to $\frac{3}{4}$ in. (15.9 to 19.1 mm). The ratio of outer surface area to inner surface area A_o/A_i is often between 3 and 4.

For a closed-circuit chilled-water system for evaporators with conventional water treatments, the fouling factor R_f can be taken as 0.00025 h·ft^2·°F/Btu (0.000044 m^2·°C/W).

From Eq. (10.9), water-side heat transfer coefficient h_i, in Btu/h·ft^2·°F (W/m^2 °C), can be calculated as

$$Nu_D = \frac{h_i D_h}{k} = 0.023 \, \text{Re}_D^{0.8} \text{Pr}^{0.4} \tag{25.11}$$

From Eq. (13.11), the log-mean temperature difference between refrigerant and chilled water ΔT_{ev} can be calculated as

$$\Delta T_{\text{ev}} = \frac{T_{ee} - T_{\text{ev}} - (T_{el} - T_{\text{ev}})}{\ln[(T_{ee} - T_{\text{ev}})/(T_{el} - T_{\text{ev}})]} \tag{25.12}$$

The mass flow rate of chilled water $\dot{m}_{w,\text{ev}}$, in lb/min (kg/min), flowing through the copper tubes in the evaporator usually remains approximately constant during operation. Temperature T_{el} is often set and reset according to the requirement of the air system and the energy efficiency of the system.

From Eq. (13.13)

$$\frac{T_{ee} - T_{el}}{Q_{\text{ev}}/(A_{\text{ev}} U_{o,\text{ev}})} = B$$

The evaporating temperature can then be determined as

$$T_{\text{ev}} = \frac{e^B T_{el} - T_{ee}}{e^B - 1} \tag{25.13}$$

At design load, Q_{ev} is equal to the design refrigeration load $Q_{\text{ev},d}$, in Btu/h (kW). In part-load operation,

$$Q_{\text{ev}} = R_{\text{load}} Q_{\text{ev},d} \tag{25.14}$$

and the evaporating temperature at part load $T_{\text{ev},p}$ can be similarly calculated.

Condenser Model

For a shell-and-tube water-cooled condenser, from Eqs. (10.16a) and (13.7), the rate of heat transfer at the condenser Q_{rej}, in Btu/h (kW), or the total heat rejection, can be calculated as

$$Q_{rej} = Q_{ev} + \frac{2545P_{com}}{\eta_{mot}}$$
$$= A_{con}U_{o,con}\Delta T_{con} \tag{25.15}$$

From Eq. (25.8) power input P_{com} is known only when condensing temperature T_{con}, work input W_{in}, and mass flow rate of the refrigerant $\dot{m}_r$ all have been calculated. Therefore, from Eq. (10.17) $Q_{rej} = F_{rej}Q_{ev}$; assume a heat rejection factor F_{rej} first. Then T_{con}, W_{in}, and P_{com} can be calculated. If the calculated P_{com} and Q_{rej} do not equal the assumed values, try another F_{rej} until the assumed and calculated values are equal to each other. For comfort air conditioning, F_{rej} usually varies from 1.20 to 1.35.

As in the evaporator model, from Eq. (13.8), the overall heat-transfer coefficient based on the outer surface area of the condenser $U_{o,con}$, in Btu/h · ft² · °F (W/m² · °C), is

$$U_{o,con} = \frac{1}{1/(\eta_j h_{con}) + A_{con}R_f/A_i + R_g + A_{con}/(A_i h_i)} \tag{25.16}$$

From Eq. (13.11), the log-mean temperature difference between the condensing refrigerant and the condenser water ΔT_{con}, in °F (°C), is

$$\Delta T_{con} = \frac{T_{cl} - T_{con} - (T_{ce} - T_{con})}{\ln[(T_{cl} - T_{con})/(T_{ce} - T_{con})]} \tag{25.17}$$

From Eq. (13.10), the condensing coefficient h_{con}, in Btu/h · ft² · °F (W/m² · °C), is

$$h_{con} = C_{con}\left(\frac{1}{Q_{rej}A_{con}}\right)^{1/3}$$

In a filmwise condensation shell-and-tube water-cooled condenser having $^5/_8$-in or $^3/_4$-in.- (15.9- or 19.1-mm-) diameter copper tubes with integrated fins and using HCFC-123 as refrigerant, constant C_{con} can be taken as 10,500.

In a condenser using a well-maintained cooling tower water with proper water treatment, a fouling factor $R_f = 0.00025$ h · ft² · °F/Btu (0.000044 m² · °C/W) is recommended. In industrial areas, if a brush cleaning system is installed, $R_f = 0.0002$ h · ft² · °F/Btu (0.000035 m² · °C/W).

The operating pressure of HCFC-123, like CFC-11, is lower than atmospheric pressure, so air and other noncondensable gases leak into the evaporator. The compressor transports them to a higher level and then they accumulate in the condenser. Noncondensable gases reduce the condensing area and raise the condensing pressure. Their effect is similar to that of a gas-side resistance R_g, in h · ft² · °F/Btu (m² · °C/W), at the condenser as follows:

$$R_g = 0.00778 - 0.0173R_{load} + 0.0114R_{load}^2 \tag{25.18}$$

As in the evaporator model, C is calculated as follows:

$$\frac{T_{cl} - T_{ce}}{Q_{rej}/(A_{con}U_{o,con})} = C$$

Condensing temperature is therefore calculated as

$$T_{con} = \frac{e^C T_{cl} - T_{ce}}{e^C - 1} \tag{25.19}$$

Cooling Tower Model

The approach of condenser water leaving the cooling tower $T_{ce} - T_o'$ is mainly influenced by the load ratio R_{load} of the condenser, the configuration of the cooling tower, the outdoor wet-bulb temperature T_o', and the number of transfer units (NTU) of the fill.

In a counterflow induced-draft cooling tower using PVC packing fill, with a condenser water flow rate of 3 gpm per ton_{ref} (0.054 L/s · kW) of heat rejection and a water-air ratio of 1.2, if the outdoor wet-bulb temperature 65°F $< T_o' <$ 78°F (18.3°C $< T_o' <$ 25.6°C), the temperature of condenser water entering the condenser T_{ce}, in °F (°C), can be roughly estimated by following performance equations:

$$\begin{aligned} T_{ce} &= T_o' + \Delta T_{\text{ap},d} K_{\text{load}} K_{\text{wet}} \\ K_{\text{load}} &= 0.1 + 0.9 R_{\text{load}} \\ K_{\text{wet}} &= 4.8 - 0.0475 T_o' \end{aligned} \tag{25.20}$$

where $\Delta T_{\text{ap},d}$ = approach of cooling tower at design condition, °F (°C)
K_{load} = load factor
K_{wet} = factor considering drop of outdoor wet-bulb temperature

Centrifugal Compressor Model

From the refrigeration cycle shown in Fig. 9.7*b* and Eq. (9.33), isentropic work input $W_{\text{isen}} = W_{\text{in}}$, in Btu/lb (kJ/kg), for a two-stage compressor can be calculated as

$$W_{\text{in}} = (1 - x)(h_2 - h_1) + h_4 - h_3 \tag{25.21}$$

From Eq. (9.35), the mass flow rate of the refrigerant $\dot{m}_r$ at the condenser, in lb/h (kg/h), is given as

$$\dot{m}_r = \frac{Q_{\text{ev}}}{(h_1 - h_9)(1 - x)} \tag{25.22}$$

From Eq. (9.30), the fraction of refrigerant evaporated in the flash cooler x can be evaluated as

$$x = \frac{h_{5'} - h_8}{h_7 - h_8} \tag{25.23}$$

On a pressure-enthalpy *p-h* diagram, the specific enthalpies of saturated liquid and vapor are functions of temperature and pressure only. In centrifugal chillers used in comfort air conditioning systems, the range between evaporating temperature T_{ev} and condensing temperature T_{con} is usually 20 to 120°F (11 to 66°C), and a simple polynomial representation can be used to calculate the specific enthalpy from the known T_{ev} and T_{con} with acceptable accuracy.

At the flash cooler, because the saturated temperature and pressure are interrelated, for simplicity, the intermediate saturated temperature between condensing and evaporating temperature can be estimated as

$$T_{\text{int}} = \frac{T_{\text{ev}} + T_{\text{con}}}{2} \tag{25.24}$$

According to the thermodynamic properties of HCFC-123 and its *p-h* diagram, the polynomial expression to calculate the specific enthalpy of the saturated vapor of refrigerant HCFC-123 h_v, between 20°F (11°C) and 70°F (21°C), in Btu/lb (kJ/kg), is

$$h_v = 89.7 + 0.145(T_{\text{ev}} - 20) \tag{25.25}$$

The polynomial expression used to calculate the specific enthalpy of the saturated liquid of HCFC-123 h_l, between 60°F (15.6°C) and 110°F (43.3°C), in Btu/lb (kJ/kg), is

$$h_l = 20.2 + 0.24(T_1 - 60) \tag{25.26}$$

Specific enthalpy differences $h_4 - h_3$ and $(1 - x)(h_2 - h_1)$ to determine the isentropic work can be calculated more simply and with acceptable accuracy according to the corresponding saturated temperature of the gaseous refrigerant at condensing, interstage, and evaporating pressures along the constant-entropy lines of similar profiles in the superheated region. The polynomial expression used to calculate the enthalpy difference of the gaseous refrigerant HCFC-123 between 60 and 120°F (15.6 and 48.9°C) along the constant entropy line Δh_s, in Btu/lb (kJ/kg), is

$$\Delta h_s = 0.135(T_{\text{int}} - T_{\text{ev}}) = 0.135(T_{\text{con}} - T_{\text{int}}) \tag{25.27}$$

where T_{int} = temperature of saturated gaseous refrigerant HCFC-123 at intermediate pressure, °F. In Eq. (25.8) η_{cp} is the compression efficiency of the centrifugal compressor, which is defined as the ratio of isentropic work to actual work delivered to the gaseous refrigerant during compression.

Also, mechanical efficiency η_{mec} is defined as the ratio of work delivered to the gaseous refrigerant to the work input to the compressor shaft. The difference in these two work inputs is mainly due to the loss in the bearings, and gear train, and during transportation of refrigerant in the centrifugal chiller. For centrifugal compressors operated at a certain speed, η_{mec} is considered a fixed value.

Mechanical efficiencies η_{mec} for centrifugal chillers manufactured after 1973 can be taken as

Without gear train	0.87
With gear train	0.85

Motor efficiency η_{mot} is a function of motor size and the load ratio of the chiller. Normally, for motor size greater than 125 hp (93 kW), ASHRAE Standard 90.1-1999 recommends an efficiency level equal to or greater than 93.0 percent as listed in Sec. 25.4. If the motor is to be operated more than 750 h annually, a high-efficiency motor is most cost-effective. The efficiency of a 200-hp (150-kW) motor should be 94.5 percent or greater. If a hermetic motor is used, another 2 to 4 percent of power input is required to provide the refrigeration capacity to cool the hermetic motor.

Compression efficiency η_{cp} is often the most influential parameter during annual energy simulation of a centrifugal compressor. Compression efficiency η_{cp} is a function of the volume flow rate of refrigerant $\dot{V}_{\text{rf}}$ and the system head or pressure lift Δp_t. Volume flow $\dot{V}_{\text{rf}}$ is closely related to load ratio R_{load}, and system head is closely related to temperature lift $\Delta T = T_{\text{con}} - T_{\text{ev}}$. In most centrifugal chillers, the chilled water leaving the chiller T_{el} is usually set at a constant value when the outdoor wet-bulb temperature is high. As the outdoor wet-bulb temperature T_o' drops, the condenser water entering the condenser T_{ce} and, therefore, the condensing temperature T_{con} fall accordingly. It is assumed that when T_o' drops below 70°F (21°C), the reset of T_{el} offsets the fall of T_o'.

Figure 25.2 shows the compressor map of a variable-speed centrifugal compressor (see Fig. 13.10). In Fig. 25.2, point A is the operating point of the centrifugal chiller at design load conditions ($R_{\text{load}} = 1$) and at a condenser entering water temperature $T_{ce} = 85$°F (29.4°C). If T_{ce} remains at 85°F and R_{load} drops to 0, then the operating point is at point B. At point B, temperature lift ΔT is calculated as follows:

$$(T_{\text{con}} - T_{\text{ev}})_B = (T_{\text{con}} - T_{\text{ev}})_A - (T_{el} - T_{\text{ev}})_A - (T_{\text{con}} - T_{cl})_A - (T_{cl} - T_{ce})_A \tag{25.28}$$

In a typical centrifugal chiller, if $(T_{el} - T_{\text{ev}})_A = (T_{\text{con}} - T_{cl})_A = 8$°F, $(T_{cl} - T_{ce})_A = 10$°F, and a typical $(T_{\text{con}} - T_{\text{ev}})_A = 67$°F (similar to scheme A in Sec. 13.6), the operating curves, or required system head, of the compressor at other T_{ce} values can then be plotted on the compression map, as shown in Fig. 25.2.

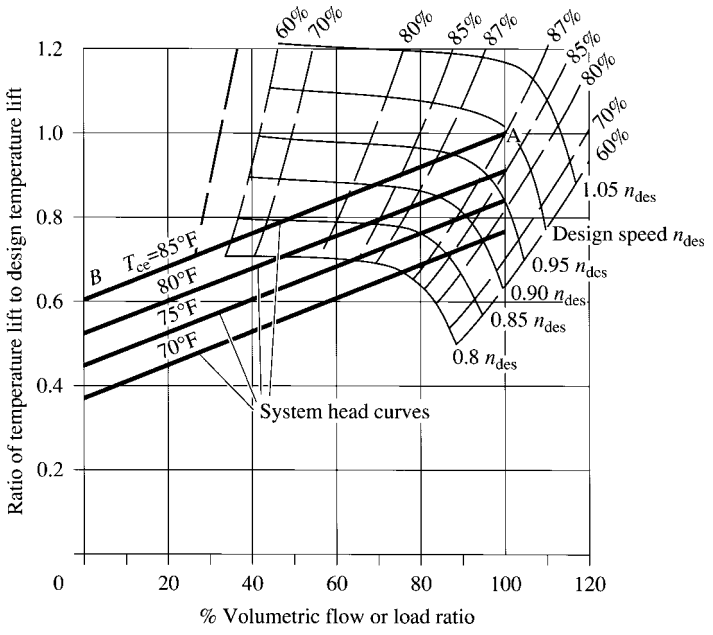

FIGURE 25.2 Compressor map of a centrifugal chiller at various load ratios R_{load} and T_{ce}.

The compression efficiency η_{cp} can therefore be calculated by a polynomial regression (performance equation) based on two variables, R_{load} and T_{ce}, as follows:

$$\eta_{cp} = C_1 + C_2R_{load} + C_3R_{load}^2 + C_4T_{ce} + C_5T_{ce}^2 + C_6R_{load}T_{ce} + C_7R_{load}^2T_{ce} + C_8R_{load}T_{ce}^2 + C_9R_{load}^2T_{ce}^2 \quad (25.29)$$

where C_1, C_2, . . . ,C_9 = coefficients. From the manufacturer's compressor map and the added operating curves at various outdoor wet-bulb temperatures T_o', the coefficients can be determined by a computer program. In a two-stage centrifugal compressor, the mean value of compression efficiency of two individual stages is assumed to equal the overall compression efficiency η_{cp} for simplification.

25.10 ENERGY SIMULATION SOFTWARE DOE-2.1E

Energy Simulation Software

For energy audit, energy efficiency improvements, and the selection of energy-efficient HVAC&R systems, Lawrence Berkeley Laboratory and Hirsch Associates, sponsored by the Department of Energy (DOE) collaboratively developed energy simulation computer program DOE-2.1E. It is the most widely used because it includes various types of systems, equipment, and energy-efficiency measures and is friendly to the user. DOE2.1E is an hour-by-hour energy simulation software. The other two widely used energy simulation software programs are TRACE 600, developed by The Trane Company, and HAP E 20-II, developed by Carrier Corporation. Building loads analysis and systems thermodynamics (BLAST) is often used in energy research works.

According to Scientific Computing (1996), another DOE-sponsored energy simulation software DOE-2.1E 100 or DOE-2.2 has similar system and equipment capacities to DOE-2.1E. The following table shows their differences:

	DOE-2.1E	DOE-2.2
Equipment time step	1 h	15 min
Sequential-iterative	Sequential	Sequential-iterative
Zone volume flow automatically calculated	Supply	Outside
Heat recovery	Sensible	Sensible and latent
Heat recovery variations during cooling		As a function of T_o and T_o'

Loads

DOE-2.1E has the following primary simulation programs: loads, systems, energy efficiency measures, and plant. The LOADS programs calculate the sensible and latent components of the hourly heating or cooling loads for each designated conditioned space (zone) in a building under the following conditions:

- The software uses 8760 h of actual weather data annually.
- Indoor temperature is assumed to be kept at a constant value within the hour.
- The LOADS program is reponsive to weather and solar conditions; to schedules of people, lighting, and equipment; to infiltration; to the heat storage effect of massive walls and roofs; and to the effect of building shade due to solar radiation.
- The LOADS program considers both external and internal loads. External loads are due to heat transfer through walls, heat transfer through windows, infiltration through walls and windows, and solar heat gain through windows. Internal loads are due to heat gains from people, lights, and equipment inside the conditioned space.
- The LOADS program performs calculations in a hierarchical order: building (system), space (zone), wall, window, or door.The program calculates the external loads for all windows and doors on a wall, then for all walls in a space. For the space, LOADS calculates the internal loads, combined with external loads, and gives the total space loads. When all space (zones) loads are summed at a consistent time, this gives the heating or cooling load for the hour of the building (system). During heating and cooling load calculations, the typical floor of a building is usually divided into many zones, such as five or nine zones; each often has external walls and windows facing one, two, or even three orientations except the interior zone.
- The LOADS program performs load calculations by using weighting factors (transfer function coefficients). Custom weighting factors for various room configurations are also used as an option instead of a preset group of weighting factors. Detailed calculations of each external or internal load using weighting factors (transfer function method) can be found in Sec. 6.6.

Systems

The energy use simulation of the following air and water systems is included:

- Single-zone constant-volume central systems
- Multizone constant-volume reheat central systems
- Multizone variable-air-volume (VAV) reheat central systems
- Multizone fan-powered VAV central systems

- Multizone dual-duct VAV central systems
- Multizone VAV reheat packaged systems
- Multizone fan-powered VAV packaged systems
- Desiccant dehumidification packaged systems
- Evaporative cooling systems
- Two-pipe fan-coil system
- Four-pipe fan-coil systems
- Packaged terminal air conditioner
- Water-source heat pump systems
- Ground-source heat pump systems
- Gas-engine-driven heat pump systems
- Thermal storage systems

Energy Efficiency Measures

Based on Scientific Computing (1996), the energy use is calculated and analyzed with the following energy efficiency measures and controls:

- *Control of the air leaving coil temperature*. Constant, coldest/warmest, reset schedule based on outdoor temperature, and user-defined reset temperature schedule.
- *Air economizer control*. No economizer, air economizer actuated by a fixed temperature, air economizer actuated by comparing outdoor air enthalpy with return air, air economizer actuated by comparing outdoor air temperature with a fixed temperature, and supply air contains fixed amount of outdoor ventilation air.
- *Type of air-to-air heat recovery*. Heat exchanger, rotary wheel, and run-around coils.
- *Fan operating schedule and control*. Fan is operating according to user-defined schedule; fan control includes constant-volume, variable-speed fan inlet vanes, discharge damper, cycles on and off, high and low speeds, as a function of part-load ratio.

Plant

Energy simulation of the following heating and cooling plant equipment can be performed either by performance equations or by physical modeling:

- Boilers include multistage resistance-type electric boilers, gas/oil-fired hot water boilers, and gas/oil-fired steam boilers. Boilers can be operated only when load is passed from the system, or in standby operation. Boilers can also be operated at a modified energy input/heat output as a function of the part-load ratio.
- Single-effect or double-effect direct-fired absorption chiller is used in which the cooling capacity can be modified as a function of refrigeration load required, or as a function of chilled water temperature entering the chiller.
- Open and hermetic centrifugal chillers modify the capacity and electric power input as a function of chilled water temperature entering the chiller, or their part-load ratio.
- Centrifugal chillers are used with double-bundle condensers in which condenser water is used as the hot water for winter heating in perimeter zone.
- Open and hermetic reciprocating chillers are used with the type of condenser specified, including air condenser or evaporatively-cooled condenser.

- Reciprocating chiller driven by a diesel engine has its COP modified as a function of the part-load ratio when the engine is at minimum speed, and COP is modified as a function of chilled water temperature entering the chiller when the engine is operated above minimum speed at part load.
- Cooling towers use the following controls of the condenser water temperature leaving the tower: fixed set point; or wet-bulb temperature reset as outdoor wet-bulb temperature drops; or control of the number of cells operating, either actual number of cells needed or all cells. The control of the temperature of condenser water leaving the tower comprises the following devices and setups:

  Fluid bypass, a three-way valve to bypass condenser water around the cooling tower

  Cycling, cycling on and off of the tower fans

  Two-speed fans, that cause the tower fans to cycle between off, low, and high speeds

  Variable-speed fans to vary the speed of the tower fans

25.11 ASHRAE/IESNA STANDARD 90.1-1999 ENERGY COST BUDGET METHOD

The energy cost budget method is an alternative to the prescriptive provisions in Standard 90.1-1999 (which is similar to the building envelope trade-off option to the prescriptive building envelope option in Sec. 3.13). Using the energy cost budget method compliance will be achieved if

- All mandatory provisions are met
- The design energy cost does not exceed the energy cost budget when evaluated in accordance with specified provisions
- The energy efficiency level of components specified in the building design meet or exceed the efficiency level used to calculate the design energy cost

Compliance shall be documented and submitted to the authority having jurisdiction as follows:

1. The energy budget for the budget building design and the design energy cost for the proposed design
2. A list of energy-related features that are included in the design
3. The input and output reports from the simulation program including a breakdown of energy use of components
4. An explanation of any error messages noted in the simulation program output

The simulation program shall be a computer-based program to analyze the energy consumption in buildings (such as, DOE-2, BLAST). The simulation shall be performed using hour values. The simulation program shall include the calculation methodologies of the building components being modeled. For details, including calculation of design energy cost, calculation of the energy cost budget, and HVAC&R systems, refer to Standard 90.1-1999.

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CHAPTER 26

AIR CONDITIONING SYSTEMS: SYSTEM CLASSIFICATION, SELECTION, AND INDIVIDUAL SYSTEMS

26.1 CLASSIFICATION OF AIR CONDITIONING SYSTEMS 26.1
- Basic Approach 26.1
- Air Conditioning Systems Classification 26.2
- Air, Heating, and Cooling Systems Designation 26.2

26.2 AIR CONDITIONING SYSTEM, SUBSYSTEM, AND MAIN COMPONENTS SELECTION 26.3
- Selection Levels 26.3
- Requirements Fulfilled during Selection 26.4
- Applications and Building Occupancies 26.4
- System Capacity 26.5
- Indoor Air Quality 26.5
- Zone Thermal Control and Sound Problems 26.6
- Energy Efficiency 26.7
- Fire Safety and Smoke Control 26.7
- Space Limitations 26.8
- Maintenance 26.8
- Initial Costs 26.8

26.3 INDIVIDUAL AIR CONDITIONING SYSTEMS 26.8
- Basics 26.8
- Advantages and Disadvantages 26.9

26.4 ROOM AIR CONDITIONING SYSTEMS 26.9
- Equipment Used in Room Air Conditioning Systems 26.9
- Configuration and Cooling Mode Operation 26.11
- Energy Performance and Energy Use Intensities 26.11
- Controls 26.12
- Features 26.12
- System Characteristics 26.12

26.5 PACKAGED TERMINAL AIR CONDITIONING SYSTEMS 26.13
- Equipment Used in Packaged Terminal Air Conditioning Systems 26.13
- Heating and Cooling Mode Operation 26.13
- Minimum Efficiency Requirements 26.14
- System Characteristics 26.15
- Applications 26.15

REFERENCES 26.15

26.1 CLASSIFICATION OF AIR CONDITIONING SYSTEMS

Basic Approach

Because of the variation in building occupancies as well as the outdoor weather in various locations, and because of different operating requirements from facility developers, owners, and users, air conditioning or HVAC&R systems are usually designed, installed, and operated in different types and configurations, with different system characteristics to meet these requirements.

The purpose of classifying air conditioning systems is to distinguish one type from another and to provide a background for the selection of an optimum air conditioning system based on requirements. As each air condition system consists of air, water, heating, and refrigeration systems, the classification of air conditioning systems and is often mixed with the classification of air systems and water and refrigeration systems. If a designer cannot properly classify an air conditioning system and distinguish it from others, it will be difficult for him or her to select an appropriate system for the client.

During the classification of air conditioning systems, the following points should be considered:

- The classification of air conditioning systems, or HVAC&R systems, should include the primary aspects of air systems as well as heating and cooling systems if possible, as the air system directly affects the control of the indoor thermal environment and indoor air quality.
- The system and the primary equipment used should be compatible with each other. For example, the primary equipment in a unitary packaged system is the packaged unit.
- System classification should mainly be based on practical applications. For example, as the indoor air quality becomes one of the primary criteria used to select an air conditioning system, it must be considered whether an "all-water system" without outdoor ventilation air supply can exist.
- System classification should be simple, and each of the air conditioning systems should be clearly different from the others.

Air Conditioning Systems Classification

As discused in Chap. 1, air conditioning systems can be classified currently into eight categories according to their configurations and operating characteristics:

- Individual room air conditioning systems or simply individual systems
- Evaporative cooling air conditioning systems
- Desiccant-based air conditioning systems or simply desiccant systems
- Thermal storage air conditioning systems or simply thermal storage systems
- Clean room air conditioning systems or simply clean room systems
- Space conditioning air conditioning systems or simply space systems
- Unitary packaged air conditioning systems or simply packaged systems
- Central hydronic air conditioning systems or simply central systems

First, as discussed in Sec. 1.5, the individual, space, packaged, and central systems together had more than 98 percent of floor area both in commercial buildings in 1992 and in air conditioned homes in 1991 in the United States. Clean room and desiccant-based air conditioning systems are often processed air conditioned systems. Except where deep-well water or an air economizer is available, evaporative cooling systems are the cheapest cooling systems and are widely used in arid southwestern areas of the United States. Thermal storage systems have a quite different water system and operating characteristics from a central system. Second, there are many specific air conditioning systems used in many industrial applications such as air conditioning used for textile mills with air washers, and also new air conditioning technology will be invented and developed. As soon as their importance and their distinct system characteristics are recognized by the HVAC&R industry, a new category of air conditioning system should be added.

Air, Heating, and Cooling Systems Designation

The title of air conditioning systems listed in later sections of this chapter and succeeding chapters, such as VAV reheat rooftop packaged systems, also designates the air, heating, and cooling systems:

- A packaged system always has a cooling system that uses the DX coil to cool air directly. A desiccant-based system usually uses the DX coil as supplementary cooling. An individual system also uses a small, self-contained, factory-assembled refrigeration system that has a DX coil to cool air.
- A central system has a cooling system that uses chilled water as a cooling medium to cool air indirectly. A thermal storage system is always a central system that uses chilled water or brine as the cooling medium.
- As discussed in Sec. 9.18, for DX coils in individual systems, rotary compressors are the most widely used. For DX coils in packaged systems, reciprocating and scroll compressors are most widely used. Scroll compressors are gradually replacing the reciprocating compressors in packaged systems because of their higher energy efficiency. For chillers in central and space systems, centrifugal and screw compressors are widely used in large chillers whose cooling capacities are greater than 75 tons (264 kW); and screw, scroll, and reciprocating compressors are widely used in medium-size and small chillers. Screw and scroll types are also gradually replacing the reciprocating compressors in small and medium-size chillers.
- As discussed in Sec. 8.1, hot water, heat pump, and direct-fired warm air furnace heating systems were installed in commercial and residential buildings with a floor area of both about 75 percent in the early 1990s. In packaged systems, direct-fired warm air furnaces and heat pumps are often used. For individual systems, either heat pumps or electric resistance heaters are often used. In central and clean space systems, hot water heating systems are most widely used.
- As discussed in Sec. 20.9, air systems can be classified into three categories:

  Constant-volume (CV) outdoor recirculating air mixing systems

  Variable-air-volume (VAV) outdoor recirculating air mixing systems

  Dedicated ventilation and space recirculating systems

  Only air systems of the space air conditioning systems are dedicated ventilation and space circulating systems. Individual, packaged systems in residentials and clean room air conditioning systems are mostly CV air systems. All the other air conditioning systems may be either CV or VAV systems and will be so designated.

26.2 AIR CONDITIONING SYSTEM, SUBSYSTEM, AND MAIN COMPONENTS SELECTION

Selection Levels

During the design of a new or retrofit system, air conditioning system, subsystems, and main components selection is performed mainly on three levels:

- *Air conditioning system level*. This level deals with such aspects as whether a VAV packaged air conditioning system or a fan-coil air conditioning system should be selected.
- *Air system, water system, central plant cooling and heating system, and control system level*. For a VAV packaged system, it must be determined whether the system is to be a single-zone VAV air mixing system, a multizone VAV reheat air mixing system, or a fan-powered VAV air mixing system. In a packaged system, its cooling system is always a DX system with either reciprocating or scroll compressors. Its heating system is either a direct-fired warm air heating system or, in locations where the unit electric rate is low, a heat pump system or an electric heating system.

  For a space air conditioning system, its air system is always a dedicated ventilation and space recirculating system. For the water system, the choice is between a plant-building loop with variable flow in the building loop and a plant-through-building loop. Either a hot water heating system or an electric heating system can be selected for a fan-coil system. A fan-coil system

requiring large tonnage of refrigeration load often uses centrifugal or screw chillers, and a small-tonnage fan-coil system usually uses screw and scroll chillers.

Except for room air conditioning systems and packaged systems of small size, usually an energy management and control system (EMCS) with direct digital control (DDC) is cost-effective. For a specific project, the control zones, control points, and selection of generic and specific control functions should be optimum.

- *Main system components level*. For air systems this involves the selection of main components, such as:

  A forward-curved centrifugal, an airfoil centrifugal, and a vane-axial supply fan.

  A supply-relief fan combination or a supply and return fan combination.

  A CO_2-based demand control ventilation, or a mixed plenum pressure ventilation control.

  An air economizer or a water economizer, a thermal wheel, run-around coils, or a heat pipe air-to-air heat recovery system.

  An adjustable frequency variable-speed drive, inlet vanes, or an inlet cone duct-static pressure control.

  A mixing flow, stratified mixing flow, or displacement flow pattern and supply outlets and return inlets layouts.

For water systems, the main system components level involve selection of

A plant-building loop or a plant-through-building loop

In a plant-building loop, whether balance valves are necessary

In a plant-through-building loop, whether a bypass throttling flow, distribution pumping, or variable flow

An open expansion tank or a diaphragm expansion tank

For refrigeration systems, main system components level involves the following:

For heat rejection systems, whether an air-cooled, water-cooled, or evaporatively cooled condenser is selected

For a centrifugal chiller, whether a two-stage or a three-stage impeller, and whether a variable-speed drive or an inlet vane's capacity modulation is selected

For reciprocating or scroll refrigeration systems, whether a variable-speed, two-speed, cylinder unloader, or on/off capacity control is selected

Requirements Fulfilled during Selection

To properly, effectively, and energy-efficiently design an air conditioning (HVAC&R) system, the designer, facility owner, or developer should collaborate to select the system according to the following requirements:

Applications and Building Occupancies

When air conditioning systems are used for different applications or building occupancies, they need different design criteria, operating hours, and different system characteristics. For example, a constant-volume central system is always used for a class 10 clean room to fabricate semiconductor wafer. When the clean room is in operation, adequate clean air must be provided to maintain unidirectional flow to prevent the contamination of semiconductor wafers by submicrometer-size particulates. A case study of a clean-space air conditioning system for a class 10 clean room is discussed in Chap. 30.

A four-pipe fan-coil system is the most widely used air conditioning system for guest rooms in luxury hotels. This is because a four-pipe fan-coil system can provide individual temperature and fan speed controls as well as a positive supply of adequate outdoor ventilation air. The fan coil can be turned off conveniently when the room is not occupied. It is isolated acoustically from adjacent rooms. The most annoying space maintenance tasks, such as changing the filters and periodic inspection and maintenance of the fan coil, can be done when the guest room is unoccupied.

Specific design criteria are always related to applications. Specific design criteria usually dictate the type of air conditioning system that should be selected. For instance, a high-precision constant-temperature room maintained at a temperature of 70 ± 0.2°F (21 ± 0.1°C) within a limited working space needs a constant-volume central system with electric terminal reheat. Electric reheat can be modulated more precisely than any other type of heating system. A central system can maintain a more uniform discharge air temperature at the AHU than DX coil packaged units. Usually, such a high-precision constant-temperature room is a clean space. A constant-volume system is always preferable to a VAV system for such rooms.

System Capacity

System capacity is closely related to the selection of an air conditioning system. For a single-story small retail shop, a constant-volume packaged system is often chosen. If the conditioned space is a large indoor stadium with a seating capacity of 70,000 spectators, a single-zone VAV central system with many AHUs using peripheral nozzles at different levels to supply conditioned air is often selected.

Indoor Air Quality

Indoor air quality (IAQ) depends mainly on the minimum ventilation control; the removal of bacteria, particulates, irritating vapors, and toxic gases; preventing dampness; and proper maintenance of the HVAC&R system. Among various minimum ventilation controls, the dedicated ventilation system with CO_2-based demand-controlled ventilation guarantees the minimum outdoor ventilation rate specified by ASHRAE Standard 62-1999 and is energy-efficient.

The dedicated ventilation system, the mixed plenum pressure control, and the outdoor air injection fan are good minimum ventilation control systems with satisfactory performance. All guarantee the specified constant-volume flow of minimum ventilation rate at the outdoor air intake in the AHU or PU.

The supply and return fan tracking systems, and the direct measurement of outdoor air intake in the AHU and PU are considered as poor minimum ventilation control systems. As the volume flow rate of the outdoor ventilation air is usually less than one-fourth that of the total supply air, it is difficult to measure it correctly by measuring the volume flow rate of supply and return air. This may result in an insufficient outdoor ventilation intake, especially during part-load operation in VAV systems.

Air economizer cycle and purging operation prior to the occupied period on weekday mornings both improve IAQ because of the introduction of all outdoor air to the conditioned space.

As discussed in Secs. 15.16 and 24.5, to remove particulates, bacteria, viruses, irritating vapors, and toxic gases from the conditioned space, air filtration and gaseous adsorption and chemisorption in the AHUs or PU can be rated into the following four grades:

- Low-efficiency air filters are used to remove dusts of size 3 to 10 μm, such as, spores, pollens, and textile fibers. Panel filters made of wire mesh and synthetic and glass fibers are often used as the filtration medium.
- Medium-efficiency filters are used to remove dusts of size 1 to 3 μm, such as, bacteria, automotive-emissions, and fine dust for the protection of coils and air distribution systems and to prevent

the nutrition of microbial growth. Pleated filters with synthetic and glass fibers as the filter's media, are also used.

- High-efficiency filters are used to remove dusts of size 0.3 to 1 μm, such as, bacteria, cooking oil fumes, and tobacco smoke. Bag or cartridge filters with synthetic and glass fibers of submicrometer diameter are used as filter media.
- In ultrahigh-efficiency filtration, HEPA has a DOP efficiency of 99.97 percent, and ULPA filters have a DOP efficiency of 99.999 percent. HEPA and ULPA filters are used to remove viruses, carbon dust, combustion smoke, and radon progeny. They are widely used in clean rooms and clean space. To remove objectionable odors, irritating vapors, and toxic gases of molecular size between 0.003 and 0.006 μm, activated carbon adsorbers and chemisorbers are often used. When high-efficiency filters, HEPA and ULPA filters, activated carbon adsorbers, and chemisorbers are used, they must be protected by medium-efficiency filters to extend their service life.

Zone Thermal Control and Sound Problems

Zone thermal control comprises the temperature and humidity control of the control zone, which affects directly the thermal comfort of the occupants in the conditioned space. In addition to the heating and cooling capacities needed to meet the space load requirements, the scope of control and control quality can be assessed in the following three areas:

Scope. Today zone control is the basis of thermal individual control. Usually, a room or a conditioned space with a floor area often between 100 and 1000 ft^2 (10 and 100 m^2) is considered as a controlled zone. For each control zone, there is often a sensor, a controller, and a corresponding terminal device (a VAV box, a fan coil, or a water-source heat pump), to control the space temperature and space humidity according to the requirements.

For a large open office, additions of cubical partitions, and use of supply outlets that can adjust the volume flow or the flow direction to regulate are means to control the space temperature and humidity that cover a smaller control zone.

Recently, a desktop task conditioning system has been developed from an underfloor upward flow space air diffusion system which can provide zone control for an individual worker in the conditioned space, as discussed in Sec. 18.9.

Volume Flow Control and Supply Temperature Control. For cooling, the zone control quality depends on the volume flow control of cold supply air of a VAV box to meet the space sensible load reduction at part-load operation or the reduction of the chilled water flow to a cooling coil. For heating, the zone control quality depends on the capacity control of a reheating system or a perimeter heating system.

Modulating the volume flow of cold supply air during part-load operation only slightly reduces the space relative humidity. However, reducing the chilled water flow to a cooling coil may significantly increase the space relative humidity especially when the fan coil is oversized.

Control Modes. More often, the zone control mode is a DDC proportional-integral (PI) modulation control without offset. It can be also a two- or three-fan speed control (small fan in fan coil or in water-source heat pump) plus on/off control of the water flow. A DDC with PI or proportional-integral-derivative (PID) control always has a better control quality because it has no offset except due to instrument inaccuracy.

Sound problems often are the result of improperly designed systems and poorly selected fans, pump, and compressors. Noise always is very annoying. Some air conditioning systems such as individual systems and space conditioning systems have inherent sound problems because the room air conditioners and water-source heat pumps are often installed directly in the conditioned space. Many fan-coil units are hung in the ceiling plenum directly above the conditioned space where the recirculating air inlet provides a short and convenient sound transmission path. The estimated sound levels in the occupied zone for various air conditioned systems are as follows:

Condition	Air conditioning system	Equipment	Estimate
Poor	Room air conditioning system	Room air conditioner	45 to 50 dBA
Acceptable	Four-pipe fan-coil system	Fan-coil unit (400 cfm, 189 L/s)	35 to 40 dBA
Good	VAV reheat central system	AHU, VAV-box	25 to 35 NC
Excellent	Single-zone VAV central system	AHU	15 to 20 NC

Energy Efficiency

As discussed in Sec. 25.6, each type of air conditioning system has its own system characteristics and energy use intensity (EUI), expressed in $kBtu/h \cdot ft^2 \cdot °F$ ($kWh/m^2 \cdot °C$), as shown in Table 25.1. Boilers, furnaces, compressors, fans, and pumps are energy users. To properly select an air conditioning system, the designer should know the energy efficiency of main energy users as well as the estimated EUI of the selected air conditioning system.

Clearly distinguish the type and configuration of energy users in an air conditioning system:

Heating equipment

Direct-fired warm air furnace

Hot water boilers, or condensing hot water boilers

Refrigeration compressors: reciprocating, scroll, screw, or centrifugal

Water pumps: centrifugal hot water pumps, centrifugal chilled water pumps, or centrifugal condenser water pumps

Fans: forward-curved/aerofoil centrifugal supply fans, centrifugal/plug/axial return fans, or axial relief fans, air-cooled axial condenser fans, evaporatively-cooled centrifugal condenser fans, or propeller cooling tower fans, and exhaust fans

Motors: high-efficiency motors

Estimate the combined efficiency of various energy users and the EUI of the selected system.

Consider energy conservation measures (ECMs), such as VAV systems, air or water economizers, heat recovery devices, variable-speed drives, and occupancy sensors, for the selected air conditioning system to improve its energy efficiency.

Fire Safety and Smoke Control

In case of a building fire in highrise buildings, as discussed in Sec. 22.12, the water flow indicator of the automatic sprinkler system actuates smoke control systems and stairwell pressurization systems, and the building automation system (BAS) coordinates various air systems to provide a smokefree escape route through the emergency exits and stairwells to the outdoors, as follows:

- Stairwell pressurized systems are actuated.
- Smoke is exhausted to outdoors from the fire floor (smoke zone) through the smoke exhaust system.
- Supply outdoor air to floor or floors above and below the fire floor or adjacent to the smoke zone to pressurize them to prevent smoke contamination.

For air systems with air economizers, the exhausted volume flow from the fire floor and the volume flow to pressurize floors or control zones adjacent to the smoke zone are approximately equal to the supply volume flow rate of the air economizer cycle.

For air systems with dedicated outdoor ventilation systems, or with a local outdoor ventilation air intake (such as air systems in the fan-coil, water-source heat pump systems, or room air conditioning

systems), a smoke exhaust system of about 6 ach (air changes per hour) should be installed; and the dedicated outdoor ventilation system can be used to pressurize the adjacent control zones according to local and federal fire code requirements.

Space Limitations

Space limitations specified by the architect or facility owner also influence the selection of the air conditioning system. For instance, for a high-rise building of more than 30 stories, if rooftop space is not available for the penthouse of AHUs and other mechanical equipment, or if there is no space left for supply and return duct shafts, a floor-by-floor AHU central system may be the practical choice. Because of the available ceiling space, low-velocity duct systems are often used in high-ceiling industrial buildings to save air-transporting energy.

Maintenance

A central system with AHUs, a few water-cooled centrifugal chillers, and cooling towers needs less maintenance work than a packaged system with many rooftop air-cooled reciprocating packaged units. A VAV reheat central system needs less maintenace work in the fan and plant rooms than a fan-coil system, which often requires much maintenance work in the ceiling space directly above the conditioned space.

Initial Costs

Initial cost and operating costs (mainly energy cost) are always primary factors that affect the selection of an air conditioning system, especialy for a developer who sells buildings with installed air conditioning systems. The initial cost of the air conditioning (HVAC&R) system in a building, expressed in $\$/ft^2$ ($\$/m^2$), depends on the building occupancies, system configurations, size of the building, and capabilities of specific systems. According to Gladstone and Humphreys (1995), the initial costs of an HVAC&R system for school buildings are as follows:

System type	$\$/ft^2$
Single-zone, constant-volume packaged systems	5.25
Multizone reheat constant-volume central system	6.60
Multizone dual-duct central systems	8.25

Generally, the more complex an air conditioning system becomes and the more features it has, the higher will be the initial cost. Among all these, the Class 1 clean room central system is the most expensive.

26.3 INDIVIDUAL AIR CONDITIONING SYSTEMS

Basics

As discussed in Sec. 1.3, an individual room air conditioning system or simply an individual system uses a self-contained, factory-made packaged air conditioner to serve an individual room. It is ready to use after electric cable and necessary water drainage are connected. Individual systems always use a DX coil to cool the air directly. Individual systems can be subdivided into the two following air conditioning systems:

- Room air conditioning (RAC) systems
- Packaged terminal air conditioning (PTAC) systems

As discussed in Sec. 1.3, individual air conditioning systems always use a DX coil for cooling. They are normally used in perimeter zones for the convenience of local outdoor ventilation air intake.

Advantages and Disadvantages

Individual air conditioning systems have the following advantages:

- There are no supply, return, or exhaust ducts.
- Individual air conditioning systems are the most compact, flexible, and lower in initial cost than others, except portable air conditioning units.
- Building space is saved for mechanical rooms and duct shafts.
- It is easier to match the requirements of an individual control zone.
- They are quick to install.

Individual systems have the following disadvantages:

- Temperature control is usually on/off, resulting in space temperature swing.
- Air filters are limited to coarse or low-efficiency filters.
- Local outdoor ventilation air intake is often affected by wind speed and wind direction.
- Noise level is not suitable for critical applications.
- More regular maintenance of coils and filters is required than for packaged and central systems.

Individual room and unitary packaged air conditioning systems are both self-contained, factory-made, packaged systems using DX coils for cooling. However, individual systems are single-zone units, whereas a packaged system can be either a single-zone or multizone unit. Individual systems have no supply and return ducts. Also, individual systems provide far smaller heating-cooling capacities, poorer outdoor ventilation intake, lower-efficiency filters, simpler control system, and far less ECMs than packaged systems. Room air conditioners are often window-mounted. Most of the packaged units are rooftop units.

As discussed in Sec. 1.5, individual air conditioning systems were found in about 26 percent of 66 million air conditioned homes and packaged systems in 74 percent in 1991 in the United States. Also, individual systems were found in 22 percent of commercial buildings and packaged systems in 48 percent in 1992 in the United States.

26.4 ROOM AIR CONDITIONING SYSTEMS

Equipment Used in Room Air Conditioning Systems

Room air conditioning systems are the most widely used individual air conditioning system in the United States. Window-mounted or through-the-wall types of room air conditioners; room heat pumps (RHPs) to provide both heating in winter and cooling in summer; and split air conditioners (SACs) with an outdoor condensing unit and an indoor air handler, for the sake of greater flexibility in location of the air handler as well as isolation of compressor noise outdoors are three kinds of equipment used in room air conditioning systems. A room heat pump has a similar configuration to a room air conditioner except a four-way reversing valve is added to change from cooling mode to heating mode, and vice versa. According to Mahoney (1996), annual shipments of window-mounted

and through-the-wall room air conditioners increased from 2.8 million units in 1991 to 4.3 million in 1995 in the United States.

The cooling capacity of currently available room air conditioners ranges between 5000 and 34,000 Btu/h (1.5 and 10 kW). Refrigerant HCFC-22 and alternative refrigerants HFC-407C and HFC-410A are the refrigerants used now and after 2020. Electric resistance heating is often used in room heat pumps to supplement the winter heating when the outside weather is cold. According to Rosenquist (1999), the average service life of a room air conditioner is 12.5 years.

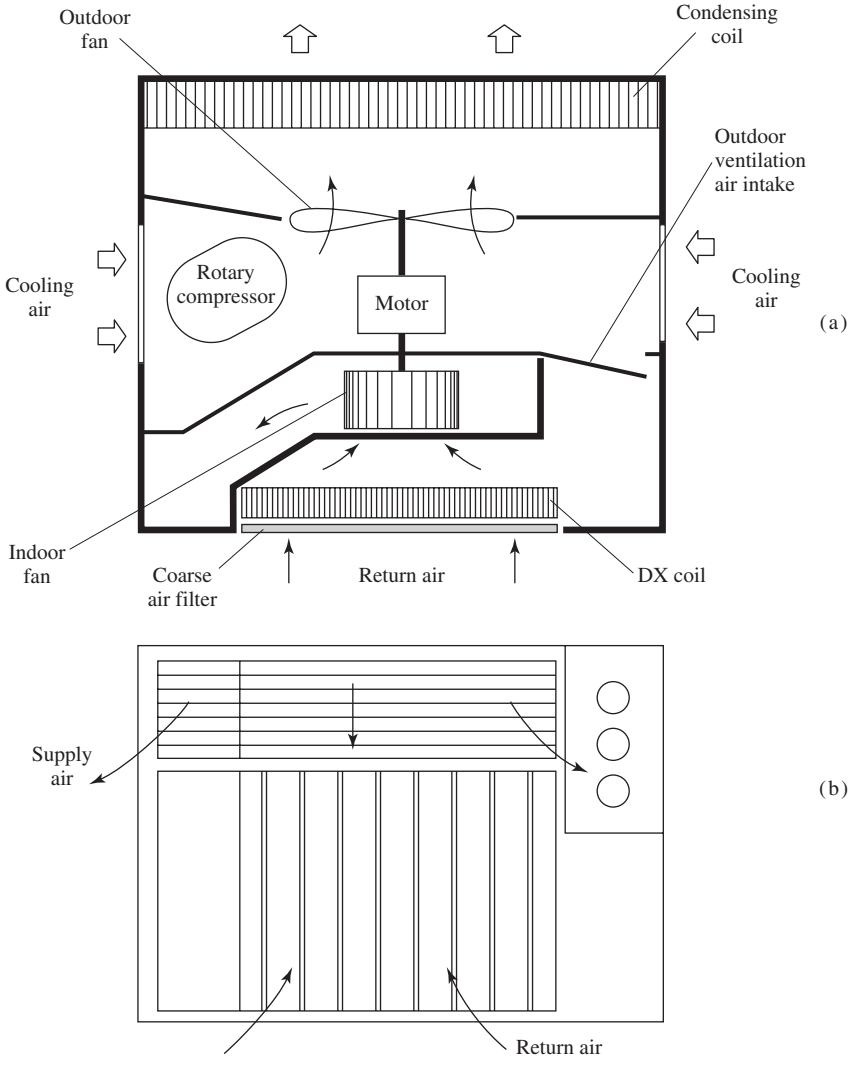

FIGURE 26.1 A window-mounted room air conditioner: (a) Sectional view; (b) front view.

Configuration and Cooling Mode Operation

Figure 26.1 shows a window-mounted room air conditioner. The cabinet is divided into indoor and outdoor compartments which are separated by insulated wall to reduce the heat transfer. The DX coil and indoor fan are in the indoor compartment. The outdoor compartment contains the compressors, condensers, outdoor fan, capillary tube, and fan motor. The fan motor often has a double-ended shaft which drives both fans.

Return air from the conditioned space flows through a coarse air filter, is cooled and dehumidified in a DX coil, and then enters the inlet of the indoor fan. In a room air conditioner, the indoor fan is a forward-curved centrifugal fan. The conditioned air is pressurized in the impeller and forced through the air passage that leads to the supply grille. The conditioned air is then supplied to the conditioned space to offset the space cooling load.

Outdoor air is extracted by the propeller fan (outdoor fan) and forced through the condensing coils, in which hot gaseous refrigerant is condensed to liquid refrigerant. During condensation, condensing heat is released to the outside through the cooling air. A portion of outdoor ventilation air is extracted by the indoor fan and mixed with the return air. The opening of the outdoor ventilation air intake is adjustable.

According to *ASHRAE Handbook 1996, HVAC Systems and Equipment*, RAC is rated under the following standard conditions:

Evaporating temperature	45°F (7.2°C)
Compressor suction temperature	55°F (12.8°C)
Condensing temperature	130°F (54.4°C)
Liquid temperature	115°F (46.1°C)
Ambient temperature	95°F (35°C)

The schematic diagram and refrigeration cycle of a room heat pump system are shown in Fig. 12.1. The heating and cooling modes operation are similar to those in a rooftop heat pump packaged system, discussed in Sec. 12.2 and shown in Fig. 12.3.

Energy Performance and Energy Use Intensities

According to Rosenquist (1999), the room air conditioners built in the early 1980s often had an EER of 7.3 and greater. In 1999, the maximum rotary compressor efficiencies range from 10.7 to 11.1 EER. Many RAC manufacturers are using slit-type aluminum fins and grooved or rifled copper refrigerant tubing in the heat exchange coils. Incorporating a large surface area to capacity ratio, it is able to yield rotary compressor efficiencies of 11.1 to 11.3 EER (3.25 to 3.31 W/W).

Permanent split capacitor (PSC) motors are the predominant fan motor used in RACs. PSC motors have an efficiency between 55 and 70 percent. Some RAC fan motors use shaded pole motors with an efficiency of 30 to 40 percent.

ASHRAE/IESNA Standard 90.1-1999 specifies the minimum efficiency requirement for room air conditioners with louvered sides and room air conditioner heat pumps with louvered sides as follows:

Equipment type	Capacity Q_{rc} Btu/h	Minimum efficiency	Efficiency as of 10/29/2001
Room air conditioner	$Q_{rc} < 6000$	8.0 EER	9.7 EER
	$6000 \le Q_{rc} < 8000$	8.5 EER	9.7 EER
	$8000 \le Q_{rc} < 14{,}000$	9.0 EER	9.8 EER
	$14{,}000 \le Q_{rc} < 20{,}000$	8.8 EER	9.7 EER
	$Q_{rc} \le 20{,}000$	8.2 EER	8.5 EER
Room air conditioner heat pump with louvered sides	$Q_{rc} < 20{,}000$	8.5 EER	9.0 EER
	$Q_{rc} \ge 20{,}000$	8.5 EER	8.5 EER

For other types of room air conditioners and heat pumps, refer to Standard 90.1-1999.

According to DOE/EIA Commercial Buildings Consumption and Expenditures 1995, individual systems using mainly room air conditioners in commercial buildings in 1995 in the United States had an annual energy consumption of 43.2 kBtu/ft$^2 \cdot$yr (12.7 kWh/ft$^2 \cdot$yr or 137 kWh/m$^2 \cdot$yr), which is only less than the EUI for central systems and single-zone constant-volume packaged systems, as shown in Table 25.1. Higher energy use in RAC systems is mainly due to the lower efficiency of the compressor and fan and the motor and drive combined efficiencies.

Controls

RACs usually use a two-stage thermostat to separate cooling mode and heating mode operations. One of the RAC controls selects the operation mode. Another control adjusts the set point and cycles the compressor on and off to maintain the required set point. The third one changes the fan speed if required.

There is another alternative. The thermostat reduces the indoor fan speed when the space temperature approaches the set point by using a two- or three-stage speed operation. If the space temperature drops further, it cycles the compressor off. If the space temperature drops still further, the indoor fan is finally cycled off.

For a room heat pump system, a two-stage thermostat often selects the heating or cooling mode operation manually. Then it cycles the compressor and evaporator fan separately or simultaneously to maintain a space temperature set point. Usually, there is a 2 to 5°F (1.1 to 2.8°C) difference between heating and cooling set points for room air conditioning systems.

If room air conditioners use a two-speed (high-low) or three-speed (high-medium-low) indoor fan and if the fan speed is regulated only a few times a day manually, then the supply volume flow of this RAC is often steady within an hour, and its air system is constant-volume in nature. If the thermostat in an RAC reduces the fan speed when the space temperature approaches its set point automatically, the air system is actually a VAV system.

Features

Many RACs have the following features:

- Most RACs are designed to extract outdoor ventilation air and mix it with the recirculating air. Many RACs also can exhaust a portion of the return air through the condenser fan.
- Many RAC condenser fans have a slinger-ring which splashs the condensate that has been collected in the pan under the DX coil where it flows onto the condenser. The water drops hit the outside surface of the air-cooled condenser, evaporate, and increase the air-side heat-transfer coefficient of the air-cooled condenser. However, in coastal areas, the corrosive effect of water with dissolved salts must be considered.
- The supply velocity from most RACs ranges from 300 to 1000 fpm (1.5 to 5 m/s). RACs have adjustable vanes mounted on the supply outlet to alter the direction of the air jet as well as its throw, to prevent drafts.
- Some RACs incorporate subcoolers to subcool the hot liquid refrigerant discharge from the condenser. Subcoolers are often added after the condenser outlet and submerged in the condensate before the capillary tube inlet. Subcoolers improve the refrigeration effect and thus the COP of the RAC system.

System Characteristics

System characteristics define system capabilities, functions, and the differences between a specific system and other systems. System characteristics of room air conditioning systems are listed in Table 26.1.

TABLE 26.1 System Characterisics for RAC, PTAC, and Their Heat Pumps

	RAC, RHP, SAC	PTAC, PTHP
Zone thermal and sound control		
Control zone	Single	Single
Control methods	Electric, two-stage thermostat or DDC, HI-LO, or HI-MED-LO fan speed	Electric, two-stage thermostat or DDC
Control modes	On-off	On-off
Heating-cooling mode changeover	Manual	Manual, automatic
Sound control	45 to 50 dBA	45 to 50 dBA
Indoor air quality (IAQ)		
Minimum ventilation air control	Constant-volume flow, affected by wind direction and speed	Constant-volume flow, affected by wind direction and speed
Filters	Coarse or low-efficiency filters	Coarse or low-efficiency filters
Air systems		
Types	Constant-volume or VAV air mixing	Constant-volume air mixing
Indoor fan	Forward-curved centrifugal	Forward-curved centrifugal
Indoor fan (IF) total pressure	0.6 in. WC	0.6 in. WC
Combined IF-motor drive efficiency	25%	25%
Volume flow control	HI-LO or HI-MED-LO fan speed	Single-speed or HI-LO speed
Outdoor fan	Propeller	Propeller
Cooling systems		
Refrigeration compressor	Rotary	Rotary and reciprocating
Refrigerants	HCFC-22, HFC-407C, HFC-410A	HCFC-22, HFC-407C, HFC-410A
Evaporator	DX coil	DX coil
Condenser	Air-cooled	Air-cooled
Refrigerant flow control	Capillary tube, RHP with four-way reversing valve	Capillary tube, PTHP with four-way reversing valve
Heating systems		
Type	Heat pump, or electric heating	Hot water, electric heating, or heat pump
Energy use, cooling	8.0 to 10.0 EER	8.5 to 10.0 EER

26.5 PACKAGED TERMINAL AIR CONDITIONING SYSTEMS

Equipment Used in Packaged Terminal Air Conditioning Systems

Packaged terminal air conditioners (PTACs) and packaged terminal heat pumps (PTHPs) are two main types of equipment used in packaged terminal air conditioning systems. Packaged terminal heat pumps add a four-way reversing valve and provide winter heating through the release of condensing heat from the outdoor coil and supplementary electric heater.

Figure 26.2 shows a packaged terminal air conditioner. The main chassis contains an air conditioner to provide heating and cooling. A cabinet sleeve is used to slide the chassis through the wall. A front panel with a return air grille provides a neat appearance. There is also an exterior louver to show a more attractive appearance on the outer building facade. Heating can be provided by heat pumps, electric heating, gas heater, or hot water heating system, especially an existing hot water system. According to *ASHRAE Handbook 1996*, all-electric PTACs are dominating the current market. In the mid-1990s, PTACs with electric resistance had a share of 49 percent of packaged terminal air conditioning systems, and PTHPs a share of 45 percent. PTACs are available with cooling capacity of 6000 to 18,000 Btu/h (1.8 to 5.3 kW).

Heating and Cooling Mode Operation

In a typical PTHP, the return air enters from the return grille on the front panel. It then flows through a coarse filter and an indoor coil (DX coil) in which it is cooled and dehumidified during

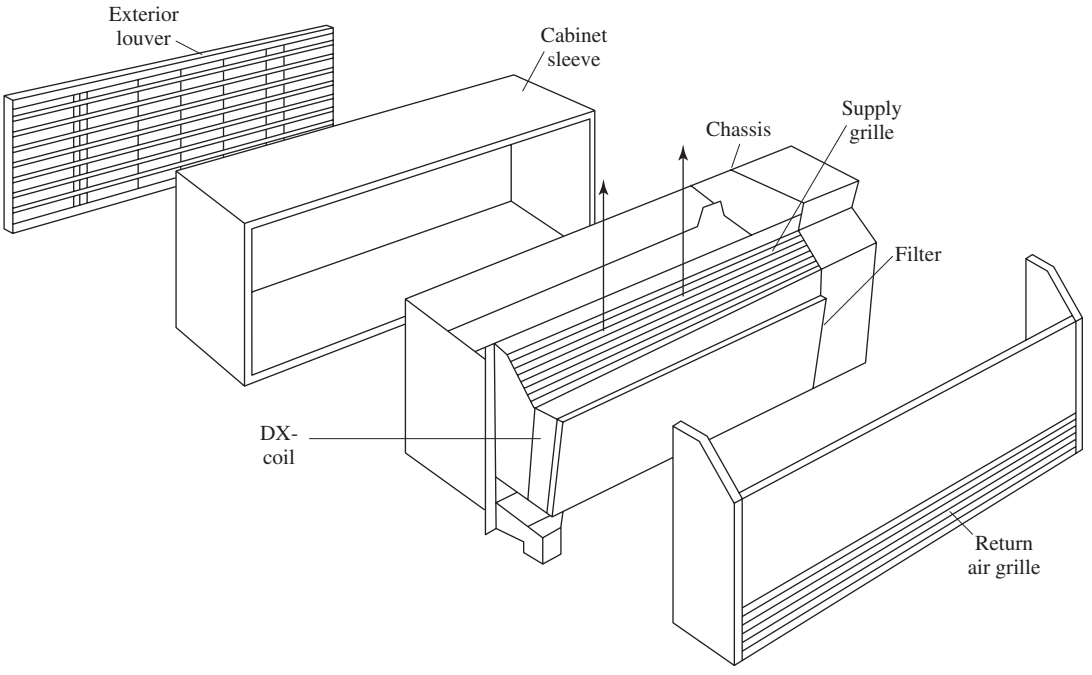

FIGURE 26.2 A packaged terminal air conditioner.

cooling mode operation. The conditioned air is then mixed with the outdoor ventilation air extracted by the indoor fan and discharged through the adjustable supply grilles upward. A thermostat senses the zone temperature and controls the refrigerant supplied to the DX coil in on/off mode to maintain a preset zone temperature.

During heating mode operation, the indoor coil acts as a condenser. The return air is then heated by the indoor coil to maintain the required zone temperature.

As in an RAC, spray the condensate on the outdoor coil (condensing coil) where it evaporates and enhances the release of condensing heat. In coastal areas, if corrosive saltwater is found in the condensate sump, consider installing a condensate draining system.

Minimum Efficiency Requirements

ASHRAE/IESNA Standard 90.1-1999 specifies that the minimum efficiency requirements for new installed packaged terminal air conditioners and packaged terminal heat pumps in cooling mode operation can be calculated as:

$$\text{EER} = 10.0 - \frac{0.16Q_{rc}}{1000} \tag{26.1}$$

Efficiency as of October 29, 2001 for packaged terminal air conditioners:

$$\text{EER} = 12.5 - \frac{0.213Q_{rc}}{1000} \tag{26.2}$$

Efficiency as of October 29, 2001 for packaged terminal heat pumps:

$$\text{EER} = 12.3 - \frac{0.213Q_{rc}}{1000} \tag{26.3}$$

For new installed packaged terminal heat pumps in heating mode operation, their minimum efficiency requirements can be calculated as:

$$\text{COP} = 2.9 - \frac{0.026Q_{rc}}{1000} \tag{26.4}$$

Efficiency as of October 29, 2001:

$$\text{COP} = 3.2 - \frac{0.026Q_{rc}}{1000}$$

where Q_{rc} = cooling capacity, Btu/h. For factory-labeled replacement of packaged terminal air conditioners and heat pumps, only their required minimum EER or COP will be slighly lower as of October 29, 2001. Refer to ASHRAE/IESNA Standard 90.1-1999 for details.

System Characteristics

System characteristics of PTACs and PTHPs are listed in Table 26.1. Compare the system characteristics of RACs with PTACs, also RHPs with PTHPs; they are quite similar, except for the following differences:

- RACs and RHPs are often window-mounted, whereas PTACs and PTHPs are mounted through the wall.
- PTACs and PTHPs have a better appearance on the building facade outdoors as well as on the front panel indoors.
- Some PTACs and PTHPs offer more control functions such as automatic changeover from cooling mode operation to heating mode operation, and vice versa, evaporator freeze-up protection, limited operation when zone temperature limits are exceeded, thermostat set points adjustments during the unoccupied period, and fault detection and diagnostics.

Applications

According to *Air Conditioning, Heating & Refrigeration News* on April 15, 1996, PTAC shipments in 1994 in the United States comprised 75,000 units and PTHPs 100,000 units. PTACs and PTHPs are widely used in commercial buildings. They are mostly adopted in relatively small zones in the perimeter zones of buildings such as hotels, motels, apartments, nursing homes, and offices.

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CHAPTER 27

AIR CONDITIONING SYSTEMS: EVAPORATIVE COOLING SYSTEMS AND EVAPORATIVE COOLERS

27.1 EVAPORATIVE COOLING AND EVAPORATIVE COOLING SYSTEMS 27.1
27.2 DIRECT EVAPORATIVE COOLING AND DIRECT EVAPORATIVE COOLERS 27.2
Direct Evaporative Cooling Process 27.2
Saturation Efficiency 27.2
Direct Evaporative Coolers 27.4
Operating Characteristics 27.6
27.3 INDIRECT EVAPORATIVE COOLING AND INDIRECT EVAPORATIVE COOLERS 27.6
Indirect Evaporative Cooling Process 27.6
Indirect Evaporative Coolers 27.7
Heat-Transfer Process 27.7
Effectiveness 27.10
Operating Characteristics 27.11
Part-Load Operation and Control 27.12
27.4 INDIRECT-DIRECT TWO-STAGE EVAPORATIVE COOLING SYSTEMS 27.13
Indirect-Direct Two-Stage Evaporative Cooler 27.13
Operating Characteristics Using Outdoor Air as Cooled and Wet Air 27.15
Operating Characteristics Using Return Air as Wet Air and Outdoor-Return Air Mixture as Cooled Air 27.15
Energy Efficiency Ratio and Energy Use Intensities 27.16
Case Study: A Two-Stage Evaporative Cooling System in Nevada's College 27.16
System Characteristics 27.18
27.5 ADD-ON EVAPORATIVE COOLERS 27.18
Add-on Indirect-Direct Evaporative Cooler to a DX Packaged System 27.18
Tower Coil and Rotary Wheel Combination 27.20
Tower, Plate-and-Frame Heat Exchanger, and Coil Combination 27.22
Plate-and-Frame Heat Exchanger 27.23
27.6 DESIGN CONSIDERATIONS 27.24
Scope of Applications 27.24
Beware of Dampness, Sump Maintenance, and Water Leakage 27.24
Selection of Summer Outdoor Design Conditions 27.24
REFERENCES 27.26

27.1 EVAPORATIVE COOLING AND EVAPORATIVE COOLING SYSTEMS

Evaporative cooling is an air conditioning process that uses the evaporation of liquid water to cool an airstream directly or indirectly so that the final dry-bulb or dry- and wet-bulb temperatures of the airstream being cooled are lower than those before undergoing the evaporative process.

An evaporative cooling air conditioning system, or simply evaporative cooling system, is an air conditioning system in which more than 50 percent of the total cooling provided annually is evaporatively cooled. It consists of mainly evaporative coolers, fans, pumps, filters, heat recovery devices, heat exchangers, a mixing box, dampers, controls, and other components. An evaporative cooler is a piece of equipment in which the evaporative cooling process proceeds. A central or a packaged air

conditioning system may be comprised of an add-on evaporative cooler which provides 50 percent or less of the total cooling annually.

There are three types of evaporative cooling processes: (1) direct evaporative cooling, (2) indirect evaporative cooling, and (3) indirect-direct evaporative cooling, as shown in Fig. 27.1.

27.2 DIRECT EVAPORATIVE COOLING AND DIRECT EVAPORATIVE COOLERS

Direct Evaporative Cooling Process

In a direct evaporative cooling process, the airstream to be cooled comes directly into contact with the wetted medium or water spray. The direct evaporative cooler in which the direct evaporative cooling process proceeds is shown in Fig. 27.1*a*. Air enters the direct evaporative cooler at point 1 and leaves at point 2. The release of the latent heat of evaporation from the directly cooled airstream lowers the airstream temperature; the airstream's humidity ratio increases because of the added water vapor.

Saturation Efficiency

Saturation efficiency is an index used to assess the performance of a direct evaporative cooler. As discussed in Sec. 15.21, saturation efficiency ε_{sat} is defined in Eq. (15.54) as

$$\varepsilon_{\text{sat}} = \frac{T_{ae} - T_{al}}{T_{ae} - T_{ae}^*} \tag{27.1}$$

where T_{ae}, T_{al} = temperature of air entering and leaving direct evaporative cooler, °F (°C)
T_{ae}^* = thermodynamic wet-bulb temperature of entering air, °F (°C)

For a direct evaporative cooler, if T_{ae} and ε_{sat} are known, T_{al} can be determined from a psychrometric chart. The value of ε_{sat} depends on the following factors:

1. Face velocity v_a of the air flowing through the direct evaporative cooler, in fpm [m/(60 s)]. For a specific cooler (water dipping or water spraying) with fixed face area A_a, in ft^2 (m^2), and a given water flow rate $\dot{V}_w$, in gpm (L/s), a higher v_a yields the following results:

- Higher volume flow rate $\dot{V}_a$ of the cooled air, in cfm (m^3/s)
- Greater evaporative cooling effect $Q_{\text{ev},c}$, in Btu/h (W), which can be calculated as follows:

$$Q_{\text{ev},c} = 60 v_a A_a \rho_a c_{pa} (T_{ae} - T_{al}) \tag{27.2}$$

where ρ_a = air density, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of moist air, Btu/lb · °F (J/kg · °C)

- Lower saturation efficiency ε_{sat}, mainly due to a smaller water flow rate for each ft^3 (m^3) of cooled air.

For most direct evaporative coolers used for comfort air conditioning, the face velocity should usually be no greater than 600 fpm (3.0 m/s) in order to prevent the carryover of water droplets. Otherwise, a water eliminator should be installed, which significantly increases the air-side pressure drop.

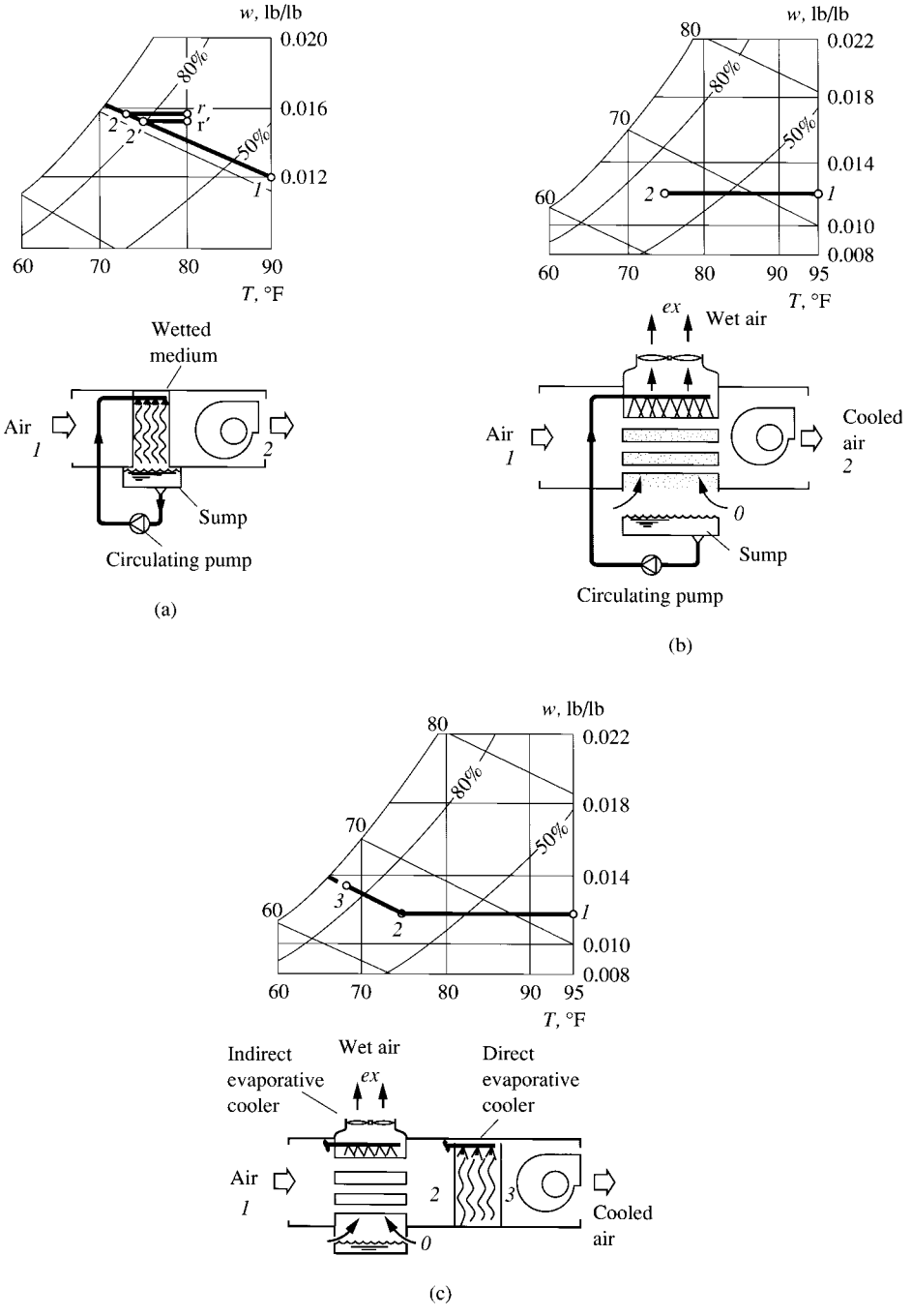


FIGURE 27.1 Types of evaporative systems: (*a*) direct evaporative cooling; (*b*) indirect evaporative cooling; (*c*) indirect-direct evaporative cooling.

2. Water-air ratio $\dot{m}_w/\dot{m}_a$. This is the ratio of the mass flow rate of spraying water to the mass flow rate of cooled air, both in lb/min (kg/min). A greater $\dot{m}_w/\dot{m}_a$ indicates a comparatively greater contact area between air and water and, therefore, a higher ε_{sat}.

3. Configuration of the wetted surface. Wetted media that provide a greater contact surface and a longer contact time between the air and water yield higher value of ε_{sat}.

When a direct evaporative cooler is used to supply cooled air to maintain a space temperature of 80°F (26.7°C) during the summer, it is important to realize that a higher ε_{sat} always means a point nearer to the saturation curve, point 2. For a predetermined space temperature $T_r = 80$°F (26.7°C), it means a higher space relative humidity φ_r, as shown in Fig. 27.1*a*.

In a direct evaporative cooling process, recirculating water is usually used in order to save water, which is often more economical. The temperature of the recirculating water always approaches the wet-bulb temperature of the cooled air.

Because the air is sprayed or in contact with dipped water, direct evaporative cooling provides a certain degree of air cleaning. However, if the cooled air contains a great deal of dirt or particulate matter, an additional filter should be used to prevent clogging of the wetted medium or nozzles.

Other parameters should be considered to assess the performance of a direct evaporative cooler including

- Use of freshwater or makeup water, usually expressed in gal/h per 1000 cfm (L/h per m^3) of cooled air
- Air-side pressure drop, in. WC (Pa)

Direct Evaporative Coolers

A stand-alone self-contained direct evaporative cooler that can provide cooled air for a conditioned space independently consists of mainly the following: a wetted medium, a fan (which is usually a centrifugal fan to provide the required total pressure loss and a lower noise level), dampers, a control system, and a sump at the bottom. For water-spraying systems, a circulating pump and piping connection are needed to distribute water evenly. To drip water on the medium from the top (except for rotary evaporative coolers), air filters, dampers, and an outer casing are necessary. Provisions should be made to bleed off the water in order to prevent mineral buildup.

Direct evaporative coolers can be categorized according to the characteristics of the wetted medium as follows:

1. *Air washers*. An air washer, or water-spraying chamber, is itself a direct evaporative cooler. The characteristics and performance of air washers are discussed in detail in Sec. 15.21.

2. *Evaporative pads*. These media are generally made of 2-in.- (50-mm-) thick aspen wood fibers with necessary chemical treatment and additives to increase wettability and to prevent the growth of microorganisms, as shown in Fig. 27.2*a*. Evaporative pads are mounted in removable galvanized steel or plastic frames. Because evaporative pads require comparatively lower face velocities, in a self-contained direct evaporative cooler integrated with a centrifugal fan, three sides of the fan cabinet are often mounted with evaporative pads to increase the surface area.

3. *Rigid media*. These are sheets of rigid and corrugated material made from plastic, impregnated cellulose, or fiberglass, as shown in Fig. 27.2*b*. Air and water typically flow in a crossflow arrangement so that horizontal channels for airflow and vertical channels for water flow meet between two corrugated sheets. The depth of the rigid medium is typically 12 in. (300 mm) in the direction of airflow but may vary from 8 to 16 in. (200 to 400 mm). Rigid media need no supporting frame. They have lower air pressure drops and can easily be cleaned by water flushing.

4. *Rotary wheel*. A wetted medium in the shape of a rotary wheel is made of corrosion-resistant materials such as plastic, impregnated cellulose, fiberglass, or copper alloy, as shown in Fig. 27.2*c*.

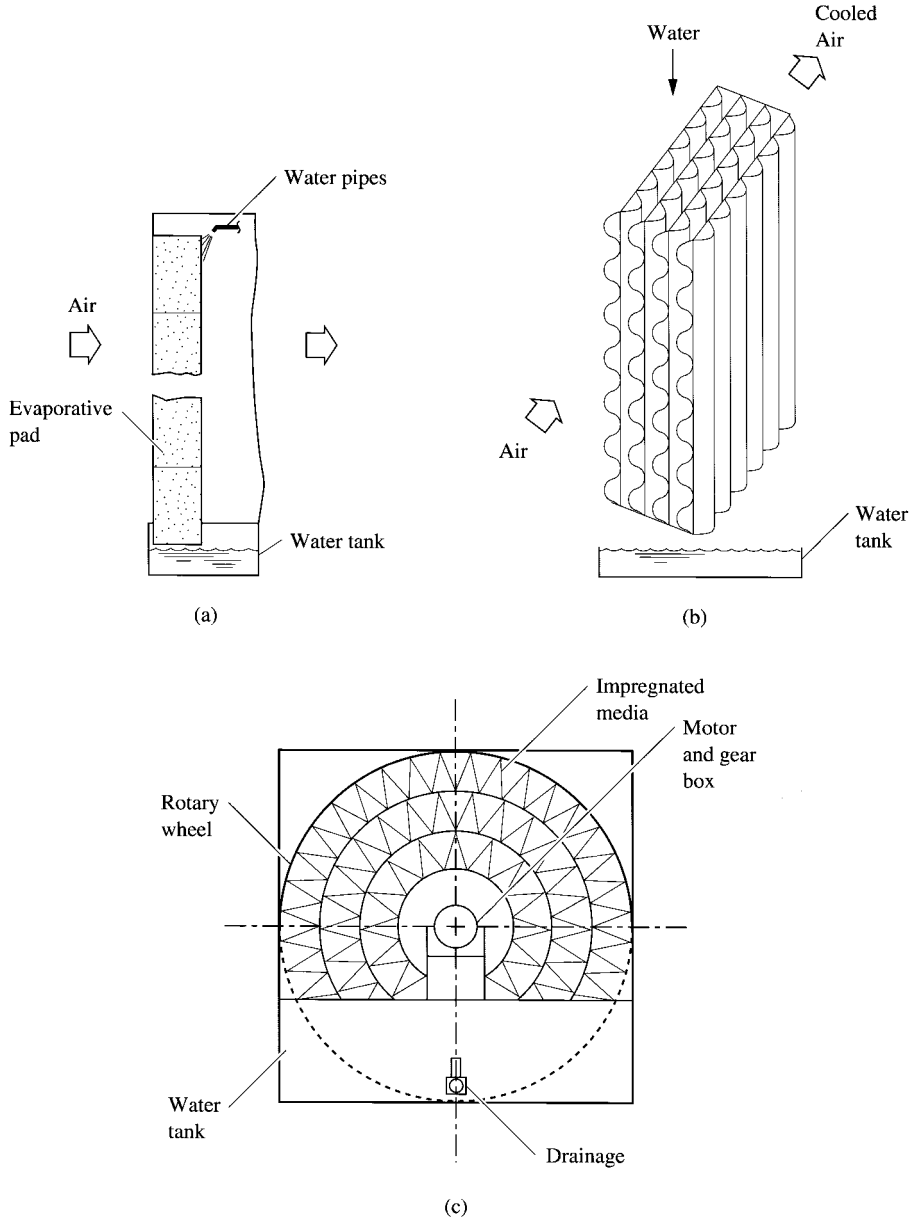


FIGURE 27.2 Wetted media for direct evaporative coolers: (*a*) evaporative pad; (*b*) rigid media; (*c*) rotary wheel.

The depth of the wheel along the airstream direction is from 6 to 10 in. (150 to 250 mm). The rotary wheel is often driven by a motor and gearbox and rotates slowly at a speed of 1 to 2 r/min. The bottom of the wheel is submerged in a water tank. Air flows through various channels of the medium in the direction along the depth of the rotary wheel.

TABLE 27.1 Operating Characteristics of Direct Evaporative Coolers with Various Wetted Media

Type of medium	Saturation efficiency ϵ_{sat}	Face velocity, fpm	Air-side pressure drop, in. WC	Water-air ratio $\dot{m}_w/\dot{m}_a$	Water usage, gal/h · 1000 cfm	Remarks
Air washer	0.80–0.90	400–800	0.2–0.5	0.1–0.4		
Evaporative pad	0.80	100–300	0.1		1.3	Pad thickness of 2 in.
Rigid media	0.75–0.95	200–400	0.05–0.1			Thickness of 8–12 in.
Rotary wheel		100–600	0.5			

Operating Characteristics

Table 27.1 lists the operating characteristics of direct evaporative coolers with various types of wetted media. The evaporative pad is the traditional type of wetted medium widely used in residential and small commercial buildings. It has a low initial cost and is easy to operate and maintain.

Rigid media such as impregnated cellulose need no support structure, do not emit debris, and have a service life as long as that of aspen pads. They withstand a comparatively higher face velocity, provide a lower air pressure drop, and have a slightly greater saturation efficiency than aspen pads.

The rotary wheel has a more complicated structure. However, it has no water-recirculating system. It is easier to connect in series with other refrigeration coolers or desiccant dryers as an add-on direct evaporative cooler in an AHU or PU for energy-efficient and cost-effective operation.

An air washer is a large-capacity, bulky, and expensive direct evaporative cooler. It is usually used for both humidification and evaporative cooling in industrial applications. The saturation efficiency values for direct evaporative coolers usually range from 0.75 to 0.95.

Among these direct evaporative coolers, only the cooled air discharged from the air washer is often at an oversaturation state. It carries numerous minute liquid water droplets. Oversaturation causes wetted surfaces and dampness along the evaporatively cooled air distribution passage. Wetted debris are the nutrients for molds and other microorganisms. To prevent serious IAQ problems, it is recommended that evaporative pad, rigid media, and rotary wheel be selected as the evaporative wetted media in comfort air conditioning. If an air washer must be used in process air conditioning, refer to Sec. 15.23 for necessary remedies.

27.3 INDIRECT EVAPORATIVE COOLING AND INDIRECT EVAPORATIVE COOLERS

Indirect Evaporative Cooling Process

In an indirect evaporative cooling process, the primary airstream to be cooled is separated from a wetted surface by a flat plate or a tube wall, as shown in Fig. 27.1*b*, and is called *cooled air*. The cooled air does not directly contact the evaporating liquid. A secondary airstream flows over the wetted surface so that the liquid water will evaporate and extract heat from the primary airstream through the flat plate or tube wall. This wet secondary airstream is known as *wet air*. The purpose of the secondary airstream is to cool the wetted surface, evaporatively approaching the wet-bulb temperature, and to absorb the evaporated water vapor.

In an indirect evaporative process, the cooled airstream's humidity ratio remains constant because the air to be cooled does not contact the evaporating liquid. This process is represented by horizontal line *1-2* on the psychrometric chart shown in Fig. 27.1*b*.

Indirect Evaporative Coolers

Figure 27.3 shows a typical indirect evaporative cooler made by an Australian manufacturer. The main components of this cooler are a plate heat exchanger, a water spray and recirculating system, an outdoor air intake with filters, a supply fan and exhaust fan connected by the same vertical shaft, and a fiberglass or stainless-steel casing to prevent corrosion.

The core part of this indirect evaporative cooler is the plate heat exchanger. The plates are made from dimpled, thin polyvinyl chloride plastic. These plates are spaced from 0.08 to 0.12 in. (2 to 3 mm) apart and form alternate horizontal and vertical passages (i.e., the air to be cooled flows horizontally, and the air that is sprayed flows vertically). Because the plates are only about 0.01 in. (0.25 mm) thick, the thermal resistance of each plastic plate is very small, although the thermal conductivity of the plastic is low.

Hot, dry outdoor air at point *o* enters the intake and filters and is extracted by the supply fan. It then enters the back of the exchanger and is forced through the horizontal passages, in which it releases its heat through the plastic plates to the adjacent wetted surfaces of the vertical passages. The cooled air at point *s* flows out the front to the conditioned space, as shown in Fig. 27.3*a*.

Water sprays over the vertical passages at the top of the plate heat exchanger, and forms both wetted surfaces and water droplets. Evaporation from these wetted surfaces and droplets absorbs heat released from the air flowing horizontally through the plastic plates. Excess water drops to a sump, which recirculates it to spraying nozzles by means of a pump. Makeup water is supplied from the city water supply to account for the evaporation and carryover. Water is periodically bled off to prevent the buildup of solid matter.

Return air from the conditioned space at point *r* is drawn through the vertical passages between the plastic plates. It absorbs the evaporated water vapor, and its humidity ratio increases. The higher the velocity of the wet air, the greater the wet surface heat-transfer coefficient h_{wet}, the larger the enthalpy difference $\Delta h_{s,w}$ between the saturated air film at the wetted surface and the wet airstream, and the higher the pressure drop of the wet airstream. Wet air is then forced through the exhaust fan and discharged to the outdoor atmospheric at point ex.

Other types of indirect evaporative coolers may use absorbent-lined vertical passages to drip water from the top through distributing troughs instead of using water sprays. Propeller fans may be used instead of centrifugal fans for wet air exhaust. Dampers may be used to extract outdoor air from outdoors or return air from the conditioned space as the wet air depends on which of them has a lower entering wet-bulb temperature.

Heat-Transfer Process

There are three fluid streams in a plate heat exchanger: cooled air, wet air, and water films along the vertical passages. Because the temperature of the saturated air film above the wetted surface is nearly equal to the wet-bulb temperature of the wet air flowing over the surface, the heat from the airstream to be cooled on the other side of the plastic plate is transferred to the wetted surface to evaporate liquid water. The heat transfer process in an indirect evaporative cooler takes place mainly between the cooled and wet airstreams.

On the cooled air side, the amount of water vapor that permeates the plastic plate is very small and can be ignored; therefore, the entering air is sensibly cooled from point *o* to point *s* at a constant humidity ratio, i.e., along a horizontal line *os* on the psychrometric chart shown in Fig. 27.3*d*. According to Dowdy and Karabash (1987), the heat-transfer coefficient h_{air} on the air side, in $\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ ($\text{W/m}^2 \cdot {}^\circ\text{C}$), is a function $h_{\text{air}} = f\,(\text{Re}_o{}^{0.8}\ \text{Pr}^{0.33})$ and can therefore be calculated as

$$h_{\text{air}} = 0.023\left(\frac{k_a}{D_h}\right)\text{Re}_D^{0.8}\text{Pr}^{0.3} \tag{27.3}$$

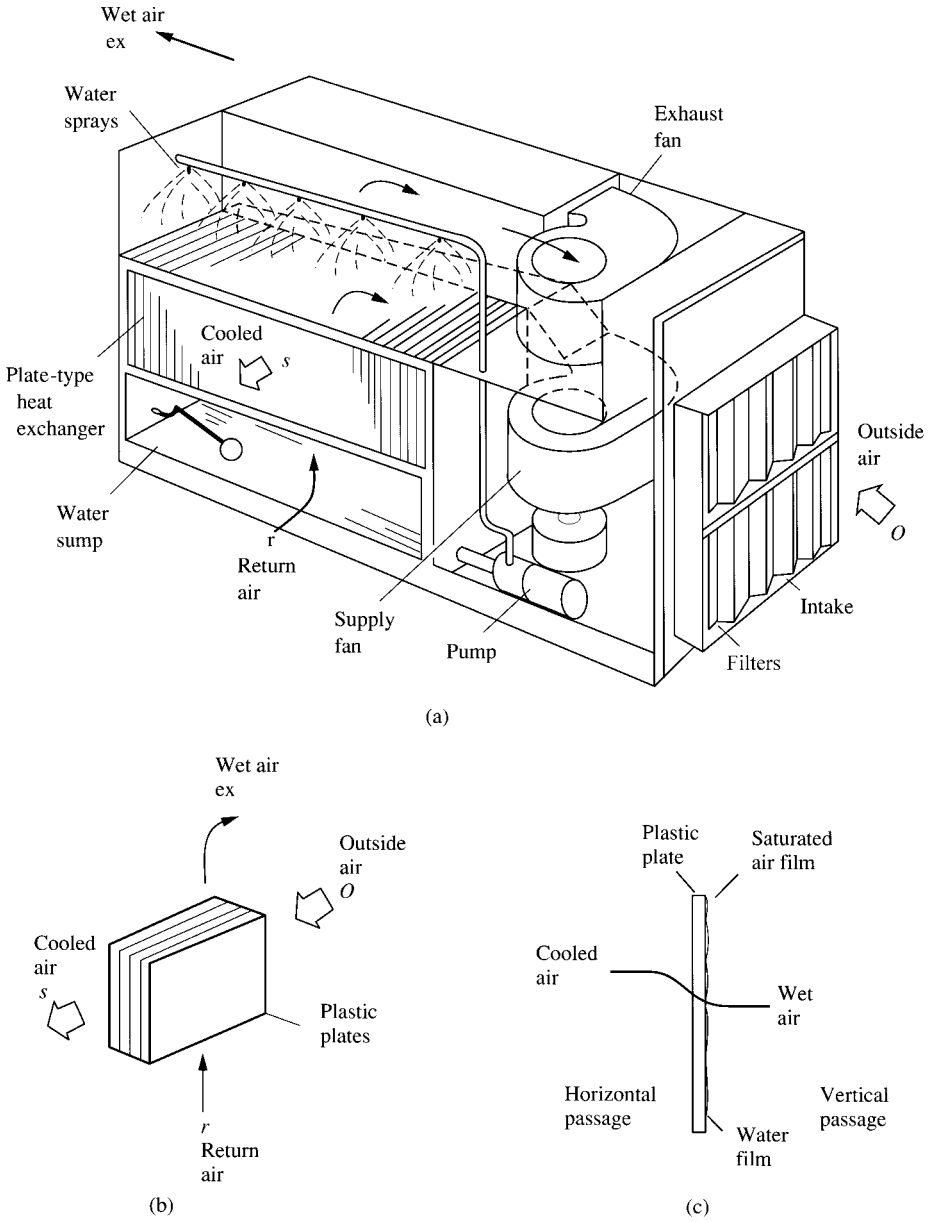
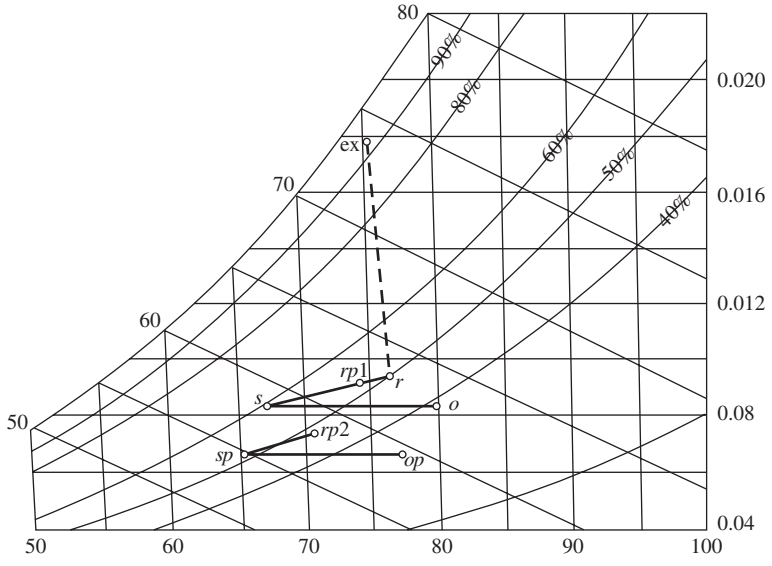


FIGURE 27.3 A typical indirect evaporative cooler: (*a*) schematic diagram; (*b*) airstream flowing through the passages; (*c*) heat transfer through plastic plates; (*d*) process on psychrometric chart.



(d)

FIGURE 27.3 *(Continued)*

where k_a = thermal conductivity of air, Btu/h · ft · °F (W/m² · °C)
D_h = hydraulic diameter of cooled air passage, ft (m)

In Eq. (27.3) the hydraulic diameter D_h is given as

$$D_h = \frac{4A_{ca}}{P_{ca}} \tag{27.4}$$

where A_{ca} = area of cooled air passage, ft² (m²)
P_{ca} = perimeter of cooled air passage, ft (m)

On the wet air side, water is sprayed onto the wet airstream; however, because of the heat transfer from the cooled air through the plastic plate to the wet air, the saturation process is no longer adiabatic. Return air from the conditioned space (which becomes the wet airstream) is humidified from point *r* to *ex*, as shown in Fig. 27.3*d*. According to Wu and Yellot (1987), the relative humidity of the air exhausted from the indirect evaporative cooler is about 95 percent, and the change in the dry-bulb temperature is rather small. Consequently, there is an increase in wet-bulb temperature and air enthalpy due to the increase in latent heat.

In the plate heat exchanger, the cooled air and the wet air are in a crossflow arrangement. The temperature of the saturated air film on the wet air side depends on the wet-bulb temperature of the local wet airstream, and the wet-bulb temperature of the wet air gradually increases during the humidifying process. The increase in the wet-bulb temperature of the wet air can be determined from its enthalpy increase Δh_{wet}, in Btu/lb (J/kg), and can be calculated as

$$\Delta h_{\text{wet}} = h_{ex} - h_r = \frac{\dot{V}_{\text{ca}}\rho_{\text{ca}}c_{pa}(T_o - T_s)}{\dot{V}_{\text{wet}}\rho_{\text{wet}}} \tag{27.5}$$

where h_{ex}, h_r = enthalpy of wet air at points *ex* and *r*, Btu/lb (J/kg)
T_o, T_s = temperature of cooled air at points *o* and *s*, °F (°C)
$\dot{V}_{ca}$,$\dot{V}_{wet}$ = volume flow rate of cooled air and wet air, cfm (m^3/s)
ρ_{ca}, ρ_{wet} = cooled air and wet air density, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of moist air, Btu/lb · °F (J/kg · °C)

The average temperature of the saturated film on the wet air side $T_{s,a}$, in °F (°C), is approximately equal to the average water temperature in the water sump $T_{w,s}$, in °F (°C). According to actual observation, $T_{w,s}$ is about 3°F (1.7°C) higher than the wet-bulb temperature of the return air for this indirect evaporative cooler. As with Eq. (10.21) in Sec. 10.8, the surface heat-transfer coefficient on the wet air side h_{wet} can be calculated as

$$h_{wet} = \left(\frac{m''}{c_{pa}}\right) h_{dry} \tag{27.6}$$

In Eq. (27.6), h_{dry} indicates the sensible heat-transfer coefficient from the wetted surface when it is dry; it can be calculated as in Eq. (27.3):

$$h_{dry} = 0.023\left(\frac{k_{wet}}{D_{h,w}}\right)\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{27.7}$$

where k_{wet} = thermal conductivity of wet air, Btu/h · ft · °F (W/m · °C)
$D_{h,w}$ = hydraulic diameter of wet air passage, ft (m)
L_{wet} = length of wet air passage, ft (m)

Effectiveness

As the heat flow in an indirect evaporative cooler is often in a crossflow pattern, the performance of an indirect evaporative cooler is mainly determined by its effectiveness. Usually, the cooled air has a smaller heat capacity rate than the wet air. Therefore, the indirect evaporative cooler effectiveness ε_{in} is defined as follows:

$$\varepsilon_{in} = \frac{T_{ca,e} - T_{ca,l}}{T_{ca,e} - T'_{wet,e}} \tag{27.8}$$

where $T_{ca,e}$, $T_{ca,l}$ = temperature of air to be cooled entering and leaving indirect evaporative cooler, °F (°C)
$T'_{wet,e}$ = wet-bulb temperature of wet air entering indirect evaporative cooler, °F (°C)

Peterson and Hunn (1992) and Peterson (1993), based on experiments in the performance of indirect evaporative coolers, recommended that the effectiveness of an indirect evaporative cooler in either outdoor air or mixing air applications be represented by the following relationships:

$$\varepsilon_{in} = \frac{1}{1 + C_{min}/C_{max}} \tag{27.9}$$

and from Eqs. (15.36) and (15.37)

$$C = \frac{C_{min}}{C_{max}} = \frac{C_{ca}}{C_{wet}} \tag{27.10}$$

$$C_{ca} = 60\,\dot{V}_{ca}\rho_{ca}c_{pa}$$

$$C_{wet} = 60\,\dot{V}_{wet}\rho_{wet}c_{sat} \tag{27.11}$$

where C_{ca}, C_{wet} = heat capacity rate of cooled air and wet air, Btu/h · °F (W/°C)
$\dot{V}_{ca}$, $\dot{V}_{wet}$ = volume flow rate of cooled air and wet air, cfm [m^3/(60 s)]
ρ_{ca}, ρ_{wet} = density of cooled air and wet air, lb/ft^3 (kg/m^3)
c_{pa} = specific heat of moist air, Btu/lb · °F (J/kg · °C)

In Peterson (1993), when cooled and wet airstreams are all outdoor air, the theoretically calculated effectiveness $\varepsilon_{in,c}$ and the measured ε_{in} based on tests are as follows:

Test	C_{min}/C_{max}	$\varepsilon_{in,c}$	ε_{in}
1	0.295	0.77	0.75
2	0.322	0.76	0.76
3	0.363	0.73	0.73
4	0.411	0.71	0.71
5	0.467	0.68	0.68
6	0.543	0.65	0.66
7	0.680	0.60	0.60
8	0.868	0.54	0.55

Wang (1996) conducted tests on the surface wettability effect on heat exchanger plates. The base test used 0.063-in.- (1.6-mm-) thick aluminum plates with various coatings 0.004 in. (0.1 mm) thick. The cooled air was set at a volume flow rate of 26.5 cfm (0.75 m^3/s), and the ratio of wet air to cooled air was set at 1.09. Also, the inlet temperature of spraying water was controlled at 77°F (25°C). For a typical test (ANC plates), when the water-air ratio by mass increased from 0.067 (10 mL/min) to 0.27 (40 mL/min), the mean value of effectiveness ε_{in} of six types of coated aluminum plates increased from 0.38 to 0.63 as follows:

Water spraying rate, lb/min (mL/min)	Water-air ratio	Average ε_{in}
0.022 (10)	0.011	0.38
0.044 (20)	0.022	0.57
0.088 (40)	0.044	0.63
0.11 (50)	0.055	0.63

As the water-air ratio exceeds 0.044, any further increase has an insignificant effect on effectiveness ε_{in}. In Eq. (27.11), c_{sat} indicates the saturation specific heat per degree of wet-bulb temperature of the wet air at constant pressure, in Btu/lb · °F (J/kg · °C); as discussed in Sec. 10.8,

$$m'' = c_{sat} = \left[\frac{dh_s}{dT'}\right] \tag{27.12}$$

where h_s = enthalpy difference along the saturation curve, Btu/lb (J/kg). In Peterson's (1993) tests, c_{sat} varied between 0.825 and 0.899 Btu/lb · °F (3.46 and 3.77 kJ/kg · °C).

The total cooling capacity of the indirect evaporative cooler Q_c, in Btu/h (W), can then be calculated as

$$Q_c = 60\dot{V}_{ca}\rho_{ca}c_{pa}\varepsilon_{in}(T_{ca,e} - T'_{wet,e}) \tag{27.13}$$

Operating Characteristics

Stand-alone, self-contained indirect evaporative coolers are made in sizes to handle volume flow rates of 1060, 2600, and 3200 cfm (0.53, 1.3, and 1.6 m^3/s). The size of the indirect evaporative cooler shown in Fig. 27.3 is 2600 cfm (1.3 m^3/s). Based on observation, its maximum power

consumption on hot summer days is 1.68 hp (1250 W), and the centrifugal fan's total pressure is from 1 to 1.3 in. WC (250 to 325 Pa).

In indirect evaporative coolers, either outside air or return air from the conditioned space can be used as the air to be cooled or the wet air. It depends on which can provide better indirect evaporative cooling results and indoor temperature and relative humidity. In Phoenix, Arizona, the cooling design wet-bulb temperature corresponding to 1 percent annual cumulative frequency of occurrence is 75°F (23.9°C). For a summer space temperature of 80°F (26.7°C) and a relative humidity of 50 percent, the wet-bulb temperature of return air is only 66.5°F (19.2°C). However, when the return air is used as the cooled air (primary airstream), a specified amount of outdoor ventilation air must be mixed with return air, and also there is always a space latent load; therefore, the space relative humidty of the space air w_r using the return-outdoor mixture is always significantly greater than that when only the outdoor air is used. Refer to the discussion in the next section for more details.

According to Peterson and Hunn (1992), a seasonal energy efficiency ratio (SEER) of 17.7 Btu/Wh (5.2 COP) is expected from an indirect evaporative cooler in Dallas, Texas. This is 70 percent higher than an air conditioner with refrigeration.

The operating characteristics of indirect evaporative coolers are affected by the flow rates and the pressure drops on the cooled air and wet air sides. For a specific cooler, the greater the volume flow, the greater the heat-transfer coefficients, the higher the air velocity flowing through the passages in the plate heat exchanger, and the higher the pressure drop.

The air velocity of the cooled air flowing through the passages is usually from 400 to 1000 fpm (2 to 5 m/s). It is important to limit the air velocity of the wet air in order to prevent carryover of water droplets. The indirect evaporative cooler effectiveness ε_{in} usually ranges from 0.60 at an air-side pressure drop of about 0.2 in. WC (50 Pa) to an effectiveness of 0.80 at a pressure drop of about 1 in. WC (250 Pa).

Usually, the cooled air-side pressure drop of indirect evaporative coolers ranges from 0.2 to 1.5 in. WC (50 to 375 Pa) depending on the air velocities in the heat exchanger and in the distributing duct. The wet air-side pressure drop varies from 0.5 to 1 in. WC (125 to 250 Pa). The volume flow ratio of cooled air to wet air changes from 0.85 to 1.67. This ratio is the key parameter that affects the heat capacity ratio C and, therefore, the indirect cooler's effectiveness, as shown in Eq. (27.9). The altitude of the unit's location also has a significant effect on its air density and, therefore, its performance.

Part-Load Operation and Control

For a constant-airflow unit, if the conditions of the outdoor air and the sensible heat ratio of the space conditioning line both remain constant, when there is a reduction in the space cooling load, the space conditioning line *sr*, as shown in Fig. 27.3*d*, tends to extend a shorter distance from point *s*. The space temperature T_{rp1} drops, and space relative humidity φ_{rp1} is slightly higher at part-load operation, as shown by point *rp*1 on the psychrometric chart. If the space cooling load remains the same and the condition of outdoor air changes from point *o* to point *op*, with lower outdoor dry-bulb and wet-bulb temperatures, then the supply temperature T_{sp}, space air temperature T_{rp2}, and space relative humidity φ_{rp2} will all be lower at part load.

For a small, stand-alone, self-contained indirect evaporative cooler, the fans automatically cycle on and off by means of a control system according to the space temperature at part-load operation. In large coolers, a multispeed fan motor can often be modulated at part-load operation when the space temperature drops below a predetermined limit. Serial combination of two indirect evaporative coolers has the same effect of increasing the water-air ratio as well as contact area and is usually not cost-effective.

Example 27.1. An indirect evaporative cooler is used in a retail store in Denver, Colorado. The 1 percent summer design wet-bulb temperature for Denver is 63°F (17.2°C), and the coincident dry-bulb temperature is 80°F (26.7°C). If the summer indoor design temperature for this retail store

is 77°F (25°C) and the supply temperature differential is 10°F (5.6 °C), calculate the effectiveness required for this indirect evaporative cooler.

Solution If outdoor air is used as the cooled air, from Eq. (27.8),

$$\varepsilon_{\text{in}} = \frac{T_{\text{ca},e} - T'_{\text{ca},l}}{T_{\text{ca},e} - T'_{\text{wet},e}} = \frac{80 - (77 - 10)}{80 - 63} = 0.76$$

A cooling effectiveness of 0.76 is required for this indirect evaporative cooler.

27.4 INDIRECT-DIRECT TWO-STAGE EVAPORATIVE COOLING SYSTEMS

Indirect-Direct Two-Stage Evaporative Cooler

When the cooled air leaves an indirect evaporative cooler during a hot summer, its dry-bulb temperature may be still above 70°F (21.1°C) with a relative humidity between 60 and 80 percent. It is beneficial to add a direct evaporative cooler so that the temperature of the cooled air can be reduced further with an increase in relative humidity. A higher relative humidity of cooled supply air is often acceptable when the supply air absorbs the space sensible load and maintains a desirable space relative humidity during hot summer. Compared with a single-stage indirect evaporative cooler, an indirect-direct two-stage evaporative cooler significantly lowers the space temperature it serves.

In an indirect-direct *two-stage evaporative cooling system* (TSECS), a direct evaporative cooler is always connected in series after an indirect evaporative cooler to form an indirect-direct evaporative cooler. If an indirect evaporative cooler is connected in series after a direct evaporative cooler, such a direct-indirect evaporative cooler is actually a single-stage direct evaporative cooler with a considerably higher discharge cooled air temperature than that of an indirect-direct cooler.

An indirect-direct two-stage cooler can provide cooled air with a dry-bulb temperature of 67°F (19.4°C) and a relative humidity of about 95 percent when the entering outdoor air is at a dry-bulb temperature of 93°F (33.9°C) and a wet-bulb of 70°F (21.1°C). The temperature of this cooled air may increase to an indoor temperature of 78° F (25.5°C) with an indoor relative humidity of 66 percent after absorbing the space cooling load during a hot summer. Indirect and direct evaporative cooling processes connected in series are illustrated in Figs. 27.1*c* and 27.4*a* along with a psychrometric chart.

For an indirect-direct two-stage cooler at part-load operation, one of the stages, either direct or indirect, can be turned off based on the indoor and outdoor conditions. In an indirect-direct two-stage cooler, plate heat exchangers are widely used in the indirect cooler. On the wet air side, either a water spray or a wetted surface treated with water-absorbent material is used. In the direct cooler, a rigid medium with impregnated cellulose is often used.

Compared with central systems using a chilled water cooling coil or a DX packaged system using a DX coil, an indirect-direct two-stage evaporative cooling system with or without a DX coil refrigeration has the following advantages:

- Evaporative cooling replaces all the refrigeration power required and maintains an indoor thermal environment within the ASHRAE summer comfort zone when the outdoor air condition is favorable.
- More outdoor ventilation air can be extracted so that IAQ will be improved.
- Better indoor relative humidity control is possible in winter heating when the indoor relative humidity is very low.

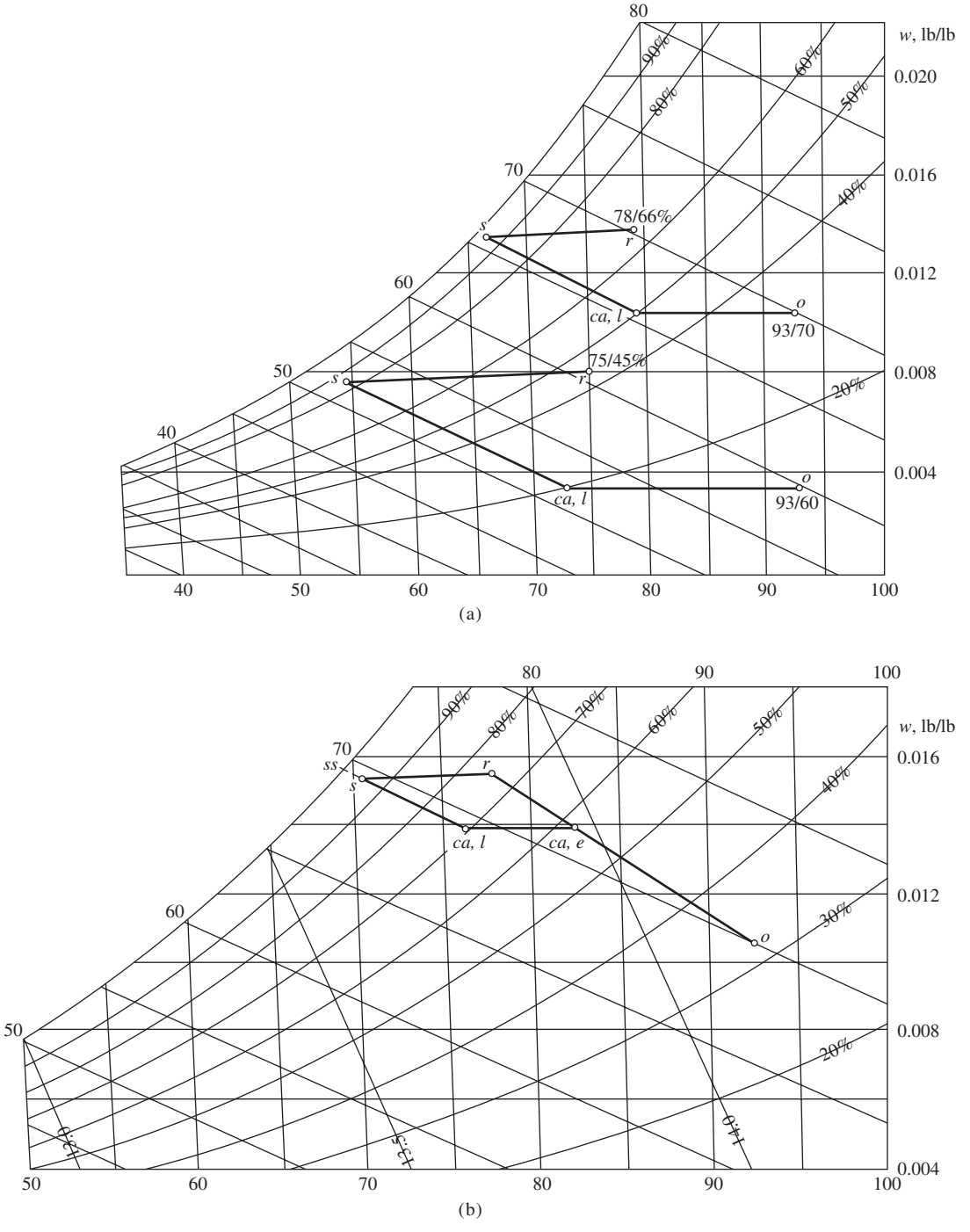


FIGURE 27.4 Operating characteristics of an indirect-direct two-stage evaporative cooling system: (*a*) using outdoor air as cooled air and wet air; (*b*) using 30 percent outdoor air and 70 percent recirculating air mixture as cooled air and return air as wet air.

The disadvantages include more maintenace for evaporative coolers because of water sprays and wetted media.

Operating Characteristics Using Outdoor Air as Cooled and Wet Air

According to the analysis in *ASHRAE Handbook 1995, HVAC Applications*, for an indirect-direct evaporative cooler that uses all outdoor air for its cooled and wet air, if the effectiveness of its indirect evporative cooler is taken as 60 percent, then the saturation efficiency of its direct evaporative cooler is 90 percent. Also, its cooled air discharged from the indirect-direct evaporative cooler can be controlled to maintain a space air condition within ASHRAE summer comfort zones after absorbing the space cooling load at a temperature of 78°F (25.6°C) or 75°F (23.9°C), with varying space relative humidity as the condition of outdoor air and the discharged cooled air change. If the space conditioning line has a sensible heat ratio of 0.90, the relationships between outdoor air state point *o*, discharge air points, and indoor space conditions point *r* are shown in Fig. 27.4*a*.

In Fig. 27.4*a* upper diagram, when the outdoor air enters the indirect-direct two-stage evaporative cooler at a 93°F (33.9°C) dry-bulb temperature and 70°F (21.1°C) wet-bulb temperature, it is evaporatively cooled indirectly to point *ca,l* with a dry-bulb temperature of 78°F (25.6°C) and a wet-bulb temperature of 66°F (18.9°C), and then evaporatively cooled directly to 66.5°F (19.2°C) with a relative humidity of 95 percent. Cooled air absorbs the system heat gain and the space cooling load, and is controlled to maintain a space temperature of 78°F (25.6°C) with a relative humidity of about 66 percent. This outdoor air condition (93°F dry/70°F wet or 33.9°C dry/21.1 °C wet) is the upper limit of the space air condition within the ASHRAE summer comfort zone that a properly designed indirect-direct two-stage evaporative cooling system can provide.

In Fig. 27.4*a* lower diagram, when the outdoor temperature enters the two-stage evaporative cooler at a dry-bulb value of 93°F (33.°C) and a 60°F (15.6°C) wet-bulb temperature, the cooled air is discharged from the two-stage evaporative cooler at 54°F (12.2°C) dry-bulb and 52°F (11.1°C) wet-bulb temperatures. The space condition can then be controlled and maintained at 75°F (23.9°C) with a relative humidity of 45 percent.

When a two-stage evaporative cooling system uses all outdoor air to provide an indoor environment within the ASHRAE comfort zones, the key parameter that affects the final indoor temperature and relative humidity is the entering outdoor wet-bulb temperature T'_o.

Operating Characteristics Using Return Air as Wet Air and Outdoor-Return Air Mixture as Cooled Air

Consider an indirect-direct two-stage evaporative cooling system using return air as the wet air. Suppose the following conditions hold:

- The outdoor air dry-bulb temperature is 93°F (33.9°C), and the outdoor wet-bulb temperature is 70°F (21.1°C).
- Thirty percent of outdoor air is mixed with the return air, and the mixture is used as the cooled air.
- The effectiveness of the indirect evaporative cooler is 0.60, and the saturation efficiency of the direct evaporative cooler is 90 percent.
- Space temperature is controlled and maintained at 78°F (25.6°C).

For a mixture of 30 percent outdoor air and 70 percent return air, the temperature of the mixture T_m, in °F (°C), can be approximately calculated as

$$T_m = 0.3(93) + 0.7(78) = 27.9 + 54.6 = 82.5°\text{F}\ (28.1°\text{C})$$

Assume that the wet-bulb temperature of the space air is 72°F (22.2°C). From Eq. (27.8), the cooled air temperature leaving the indirect cooler $T_{ca,l}$, in °F (°C), can be calculated as

$$T_{ca,l} = T_{ca,e} - \varepsilon_{in}(T_{ca,e} - T_{wet,e}) = 82.5 - 0.6(82.5 - 72) = 76.2°F\ (24.6°C)$$

From Fig. 27.4*b*, draw line *ca,l-ss* from point *ca,l* that parallels the thermodynamic wet-bulb temperature line. As the saturation efficiency is 0.9, point *s*, the condition of cooled air leaving the direct evaporative cooler, can then be determined; T_s is 70.5°F (21.4°C). Draw line *sr* from point *s* with a sensible heat ratio of 0.9. Point *r*, which represents the condition of the space air, intersects line *ro* at *r*. The space air temperature is 78°F (25.6°C), space relative humidity is 75 percent, and the space wet-bulb temperature is 72°F (22.2°C), which is approximately equal to the assumed value.

From the above analysis:

- Because of the addition of the space latent load, the space humidity ratio and space relative humidity at point *r* must be higher than the humidity ratio and the relative humidity of the outdoor-return air mixture point *ca,e* in Fig. 27.4*b*. The space humidity ratio and relative humidity are also higher when the outdoor-return air mixture is used instead of all outdoor air as shown in Fig. 27.4*a* upper diagram.
- All outdoor air supplied to the conditioned space significantly improves the IAQ more than when the outdoor-return air mixture is used.

Energy Efficiency Ratio and Energy Use Intensities

According to *ASHRAE Handbook 1999, HVAC Applications*, the energy efficiency ratio (EER) for an indirect-direct, two-stage evaporative cooling system, including the energy use of fan and pumps in indirect-direct two-stage coolers, ranges between 12.9 and 38.7. Cities like Austin, Texas, have the lowest EER; and Denver, Colorado, and Albuquerque, New Mexico, both have the highest EER. Compared with a conventional refrigeration system whose EER = 10, an indirect-direct two-stage evaporative system installed in the Los Angeles area only needs 35 percent of the energy use of a conventional refrigeration system.

However, a stand-alone evaporative cooling system needs ducts, filters, circulating pumps, and supply, recirculating, and exhaust fans. According to DOE/EIA Commercial Buildings Consumption and Expenditures 1995, the energy consumption of evaporative cooling systems in commercial buildings in 1995 in the United States was 38.0 kBtu/ft^2·yr (11.1 kWh/ft^2·yr, or 119 kWh/m^2·yr), as shown in Table 25.1, which is less than that of other air conditioning systems except for heat pump systems.

Case Study: A Two-Stage Evaporative Cooling System in Nevada's College

Scofield (1998) reported an indirect-direct two-stage evaporating cooling system with DX coil refrigeration which was used in a new three-story building at Western Nevada Community College, at an elevation of 5100 ft (1555 m), in Carson City, Nevada. This new classroom and laboratory complex has a floor area of about 70,000 ft^2 (6500 m^2).

HVAC&R System. An indirect-direct two-stage *evaporative cooling-VAV packaged system* (EC-VAVPS) with DX coil and air-cooled condenser was compared with a fan-powered VAV central system with water-side economizer. When only an indirect-direct two-stage evaporative cooler is adopted without any refrigeration as the final stage of cooling, occupants have complained of higher indoor relative humidity. The final decision was made to install an EC-VAVPS with heat recovery by heat-pipe heat exchangers.

The 55°F (12.8°C) all outdoor cooled air from the two-stage evaporative cooler provides cooling for 95 percent of the cooling hours. However, this cooled air temperature is again lowered to 50°F (10.0°C) by the DX coil so that the space relative humidity in summer can be maintained at 40 to 45 percent. This also reduces the supply volume flow as the air density is increased.

Winter Heating. During winter heating, the heat-pipe heat exchanger uses exhaust heat to raise the outdoor air temperature from 0°F (−17.8°C) up to 42°F (5.6°C). As the supply volume flow rate is turned down 50 percent in winter, a DDC controller positions the outdoor and return themselves to introduce 76 percent outdoor air and 24 percent recirculating air. A 10 percent difference

TABLE 27.2 System Characteristics of Indirect-Direct Two-Stage Evaporative Cooling Systems

System structure	TSECS with or without DX refrigeration
Zone thermal and sound controls	
Control zones	Single zones or multiple zones
Control method	VAV or CV
Control mode	DDC, PI control mode, or on-off
Sound control	NC 30 to 45
Indoor air quality	
Minimum ventilation control	All outdoor air, or 20% to 30% fixed amount of outdoor ventilation air
Filters	Medium efficiency filters (MER 9 to 12)
Air systems	
Types	Two-stage evaporative VAV packaged system (VAV reheat)
Supply fan, types	Centrifugal, unhoused or plug fans
Supply fan total pressure	3.0 to 4.0 in. WC
Combined supply fan and motor drive efficiency	0.5
Supply fan volume flow control	Inlet vanes, variable-speed drives
Return/relief fan	Axial fan, centrifugal, unhoused or plug fans
Return/relief fan total pressure	0.6 to 1.2 in. WC
Return fan volume flow control	
Supply-relief fan combination	Automatically balanced
Supply-return fan combination	Space pressurization control
Heat recovery	Heat pipe heat exchanger, 0.65 effectiveness
Operating modes	Cooling, heating, nighttime setback, purging, warm-up, optimum start and stop
Cooling systems	
Indirect evaporative cooler	Effectiveness 0.6
Direct evaporative cooler	4- to 12-in. rigid media, saturation efficiency 90%
Refrigeration systems	
Compressors	Scroll, or reciprocating
Energy use	EER 10 to 12
Capacity control	Multiple units, staging on/off
Evaporator	DX coil
Condensers	Air-cooled or evaporatively-cooled
Heating systems	
Type	Hot water reheat
Boiler	Gas and electric boilers
Maintenance	Sumps equipped with side-stream centrifugal slit separators; periodical flush of sumps

in supply and return/exhaust volume flow is used to maintain a positive space pressure. When the outdoor air temperature is raised above 10°F (−12.2°C), the heat-pipe heat exchanger can furnish 100 percent outdoor air.

During the dry winter season another alternative is to overheat the outdoor air by using a hot water heating coil. At the same time, use the 4-in.- (100-mm-) deep wetted media direct evaporative cooler to add necessary humidification and still maintain the required 50°F (10°C) supply air temperature. When the outdoor air temperature is between 30 and 40°F (−1.1 and 4.4°C), it is expected to maintain the indoor relative humidity at 30 percent.

Cooling Mode Performance. The operating parameters of this indirect-direct two-stage evaporative cooling VAV packaged system with DX coil refrigeration are as follows:

- Outdoor air enters the indirect evaporative cooler at 96°F (35.6°C) dry-bulb and 62°F (16.7°C) wet-bulb temperatures.
- It is evaporatively cooled indirectly with an effectiveness of 0.66 to a dry-bulb temperature of 73.5°F (23.0°C) and a wet-bulb temperature of 54.3°F (12.4°C).
- It is sensibly cooled in a DX coil by refrigeration to a dry-bulb temperature of 59.2°F (15.1°C) and a wet-bulb temperature of 48.5° F (9.2°C).
- It is evaporatively cooled in a direct evaporative cooler to a dry-bulb temperature of 50°F (10.0°C) and a wet-bulb temperature of 48.5°F (9.2°C) and is then supplied to the conditioned space with a saturation efficiency of 90 percent.
- The cooled supply air absorbs the fan and duct heat, as well as the space cooling load, and finally becomes the space air at a dry-bulb temperature of 75°F (23.9°C) and a wet-bulb temperature of 60°F (15.6°C) with a space relative humidity of 45 percent.

Energy and Initial Cost Saved. For a duty cycle of 7 a.m. to 7 p.m., 6 d/week, there is a 78 percent reduction in annual ton-hour (kWh) consumption when an indirect-direct two-stage evaporative cooling VAV packaged system with DX coil refrigeration is used compared with a conventional VAV central system. The installation cost for the two-stage evaporative cooling VAV packaged system is $857,183 and for the conventional VAV central system is $971,016.

System Characteristics

System characteristics of an indirect-direct *two-stage evaporative cooling system* (TSECS) are listed in Table 27.2.

27.5 ADD-ON EVAPORATIVE COOLERS

Add-on Indirect-Direct Evaporative Cooler to a DX Packaged System

In locations where outdoor air has a higher wet-bulb, temperature, evaporative cooling alone cannot provide the required cooling to maintain a desirable indoor environment. In many applications, an add-on evaporative cooler to a packaged system, as shown in Fig. 27.5*a* either with a component sequence: indirect cooler, DX coil, and direct cooler or with a sequence: indirect cooler, direct cooler, and DX coil, is often more economical than a packaged system alone.

Anderson (1986) has compared a packaged system with an add-on indirect-direct cooler and a packaged system using a DX coil only:

Outdoor conditions	100°F (37.8°C) dry, 70°F (21.1°C) wet
Supply air	57.5°F (14.2°C) dry, 56.5°F (13.6°C) wet
Space temperature	78°F (25.6°C)

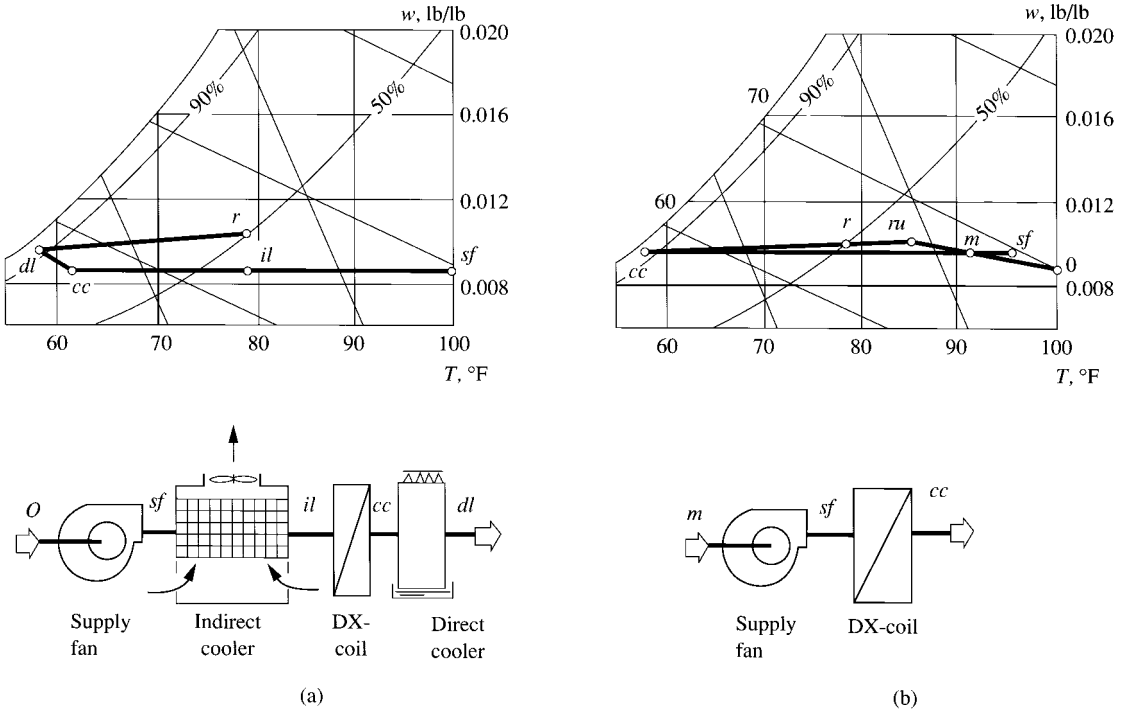


FIGURE 27.5 Evaporative cooler and DX coil versus DX coil only: (*a*) Evaporative cooler and DX coil; (*b*) DX coil only.

Space relative humidity	50 percent
System pressure drop	
DX coil only	2 in. WC (500 Pa)
Evaporative cooling and DX coil	2.75 in. WC (688 Pa)
Wet air	0.85 in WC (213 Pa)
Fan efficiency	0.6
Indirect cooler effectiveness	0.7
Direct cooler saturation efficiency	0.9
EER refrigeration	9 Btu/Wh (COP 2.64)
Outdoor ventilation air	15 percent
Ratio of installation cost of evaporative coolers and DX coil to DX coil only	2.25
Energy use (air and refrigeration side)	
Evaporative coolers and DX coil	1.10 kW/ton (COP 3.20)
DX coil only	1.79 kW/ton (COP 1.96)

In the evaporative cooler and DX coil combination, return air is used for the wet air, and outdoor air is used only for the cooled air. For cooling capacity control of such a combination, components are usually energized in the following sequence: indirect cooler first, then direct cooler, and finally the DX coil.

If the electric power rate is \$0.05/kWh, the number of payback hours for an indirect-direct evaporative cooler added onto a DX coil packaged system is 7948; if the rate is \$0.075/kWh, the number of payback hours is 5295; and if the rate is \$0.1/kWh, the number of payback hours is 3974 h.

Tower Coil and Rotary Wheel Combination

When a cooling tower is connected to a water cooling/heating coil as shown in Fig. 27.6*a*, the tower coil becomes an indirect evaporative cooler. During the cooling season, the condenser water from the tower is forced through the coil to cool the air flowing over it. Such a cooling coil is often used as a precooling coil, because there may be another cooling coil downstream. In the heating season, hot water from the condenser may flow through the coil and heat the air.

Using water from the cooling tower to cool the air by means of a precooling coil (to replace all or part of the refrigeration) is often called a water-economizing process. The combination of a cooling tower and the connected water cooling coil is called a water economizer and is shown in Fig. 27.6*a*.

If the tower coil is in series with a direct evaporative cooler using a rotary wheel, the combination is actually an indirect-direct cooler which can maintain space conditions in summer similar to those achieved with refrigeration for areas where the outdoor wet-bulb temperature is below 65°C (18.3°C), as discussed in Sec. 27.4. Water returns from the precooling coil, typically at a temperature of 78°F (25.6°C), enters the cooling tower, and is evaporatively cooled to about 70°F (21.1°C). Water is then drawn through the precooling coil, where it absorbs the heat from the air flowing over the coil. It is then pumped back to the tower at a temperature of about 78°F (25.6°C) to be evaporatively cooled again.

Outdoor air at a dry-bulb temperature of 100°F (37.8°C) and a wet-bulb temperature of 65°F (18.3°C) is drawn through the precooling coil by the supply fan and is sensibly cooled to 75°F (23.9°C). It then flows through a rotary wheel type of direct cooler and is evaporatively cooled, typically to 57.5°F (14.2°C) dry-bulb and 56°F (13.6°C) wet-bulb temperatures. After that, air is supplied to the conditioned space. Recirculating air may be used instead of outdoor air when it is more beneficial.

In an effective tower coil, the approach of the cooling tower should be around 5°F (2.8°C). In order to have such an approach, the cooling tower must be 60 percent larger than a tower with a 10°F (5.6°C) approach. The row depth and fin spacing of the precooling coil should be selected to sensibly cool the outdoor air from 100°F (37.8°C) to 75°F (23.9°C) at an entering water temperature of 70°F (21.1°C).

Field experience and tests have shown that a low face velocity through the wetted media of the rotary wheel results in a saturation efficiency above 0.90 and prevents carryover. In these tests, the air velocity was about 700 fpm (3.5 m/s) with a pressure drop of 0.25 in. WC (63 Pa). The rotary wheel revolved at a rate of approximately 1.5 r/min.

In a system with an add-on evaporative cooler, the horsepower and number of operating hours of the fan always far exceed those of the water pump. To save energy and reduce operating costs for a large unit, a two-speed (or even three-speed) fan is often economical. The efficiency of the cooler that consists of cooling tower, cooling coil, and rotary wheel depends largely on the conditions of the fills in the tower, the inner surface of the cooling coil, and the wetted media in the rotary wheel. If the wetted surfaces are clogged with dirt and scale, the efficiency will decrease proportionally to the resulting drop in evaporation and airflow. Periodic bleedoff and other necessary water treatments are essential for good performance.

Another type of spraying coil–rotary wheel combination is shown in Fig. 27.6*b*. When a spraying coil is connected to a cooling coil, water that has been evaporatively cooled in the former can be pumped to the cooling coil to absorb heat from the ambient air. Such a spraying coil-coil combination is actually an indirect cooler. If this indirect cooler is connected with a rotary wheel or other type of direct cooler, the resulting combination has a system performance similar to that of a tower-

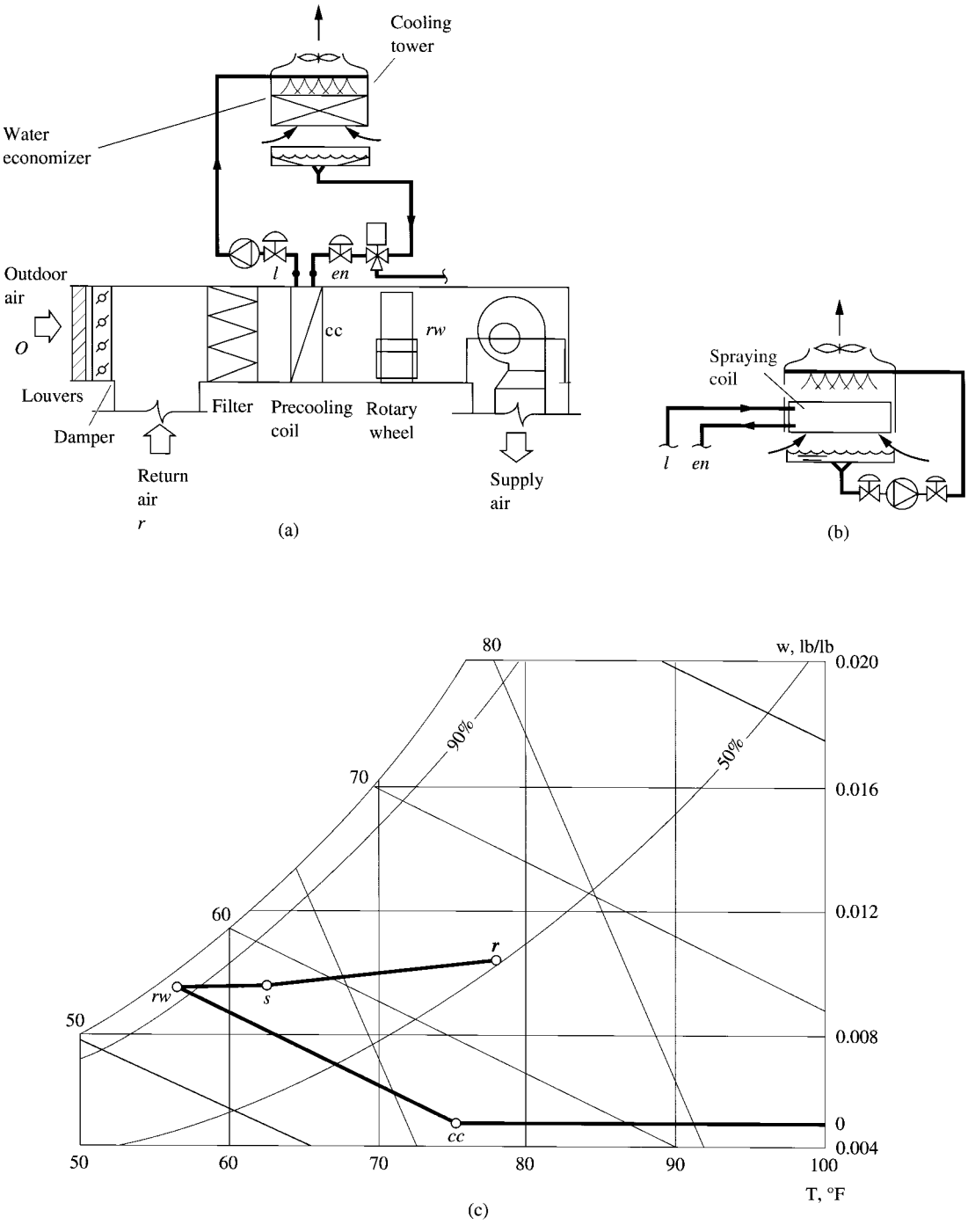


FIGURE 27.6 (*a*) Tower coil and rotary wheel combination; (*b*) spraying coil-coil and rotary wheel combination; (*c*) evaporative cooling process.

coil and rotary wheel combination. The heat transfer between two airstreams by means of a spraying coil-coil combination is always higher than that of a coil-to-coil arrangement, which is commonly located inside the outdoor air and exhaust airstreams or between makeup air and exhaust airstreams, and is often called a runaround system.

Tower, Plate-and-Frame Heat Exchanger, and Coil Combination

When the outdoor wet-bulb temperature drops below a certain value (about 40°F, or 4.4°C), it is possible to use the condenser water from the cooling tower to cool the air and thus replace all or part of the chilled water from the refrigeration plant, as shown in Fig. 27.7. To prevent the dirt and solid matter contained in the open-circuit tower condenser water from scaling the inner surface of the chilled water coil, a plate-and-frame heat exchanger is often used so that the tower condenser water does not enter the coil directly.

In Fig. 27.7, the condenser water from the tower at 45°F (7.2°C) enters the plate-and-frame heat exchanger. It cools all or part of the chilled water returning from the cooling coil to 50°F (10.0°C). Chilled water from the plate-and-frame heat exchanger is then mixed with the water from the centrifugal chiller and enters the chilled water cooling coil at 45°F (7.2°C).

To cool the chilled water at the heat exchanger when the outdoor wet-bulb temperature is 40°F (4.4°C), the approach of the cooling tower should be no greater than 5°F (2.8°C) and the temperature difference at the heat exchanger between the condenser water from the tower and the chilled

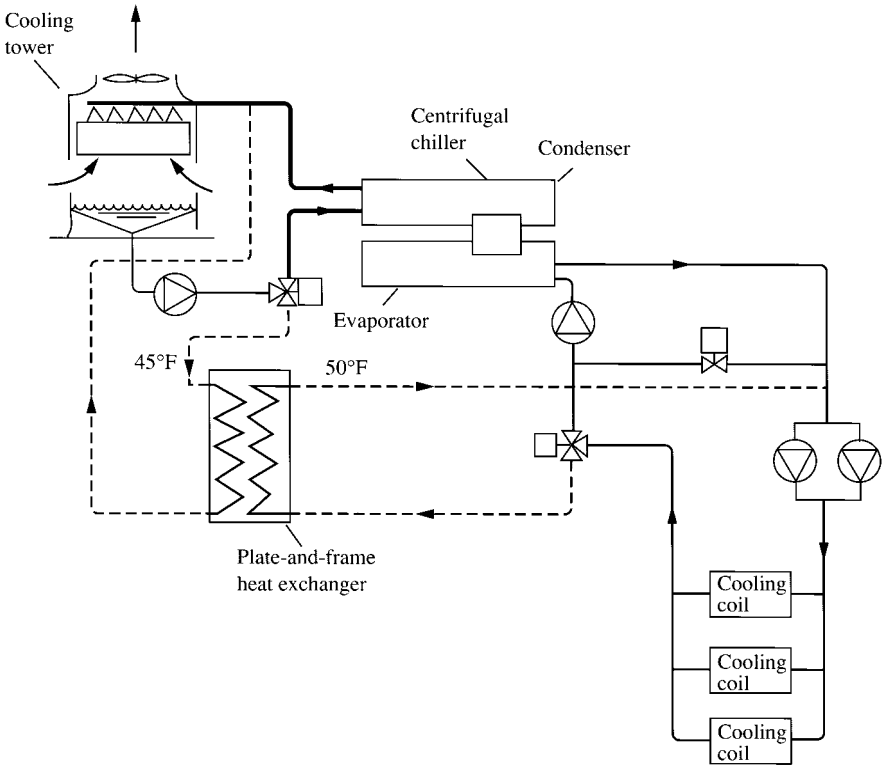


FIGURE 27.7 Tower, plate-and-frame heat exchanger, and coil combination.

water from the coil also should not exceed 5°F (2.8°C). Cost analysis is required to determine the optimum size of the cooling tower and heat exchanger and the flow rates of the condenser water and chilled water.

Plate-and-Frame Heat Exchanger

A plate-and-frame heat exchanger, as shown in Fig. 27.8, is a liquid-to-liquid heat exchanger. It consists of a number of corrugated metal plates that are usually made of stainless steel, titanium, or aluminum-brass alloy. Corrugated plates and gaskets form alternate passages that separate two different fluids. Gaskets are usually made of elastomers. Warm fluid flows downward on one side of the corrugated plate, and cold fluid flows upward on the other side in a counterflow arrangement. The corrugated plates are compressed together by a fixed-end frame and a movable-end frame with clamping bolts. Proper selection of the gasket material and proper operating conditions are important to prevent fluid leakage. The plates and movable-end frame are suspended from an upper carrying bar and lower guide bar. Fluid connections are located in the fixed-end frame.

The gap between two adjacent plates is rather small, usually ranging from 0.1 to 0.2 in. (2.5 to 5 mm). Because the plates are corrugated, a high degree of turbulence is produced, which results in a high heat-transfer coefficient. The following equation may be used for the calculation of the heat transfer:

$$h = 0.2536\left(\frac{k}{L_g}\right)\mathrm{Re}_L^{0.65}\mathrm{Pr}^{0.4} \tag{27.14}$$

where k = thermal conductivity of fluids, Btu/h·ft·°F (W/m·°C)
L_g = spacing of gap, ft (m)

The water velocity is usually between 60 and 200 fpm (0.3 and 1 m/s). The higher the fluid velocity, the higher the heat-transfer coefficient and the greater the pressure drop of the liquid flowing through the heat exchanger. For a typical plate-and-frame heat exchanger, the overall heat-transfer coefficient may be 740 Btu/h·ft²·°F (4200 W/m²·°C) through the corrugated plate between the

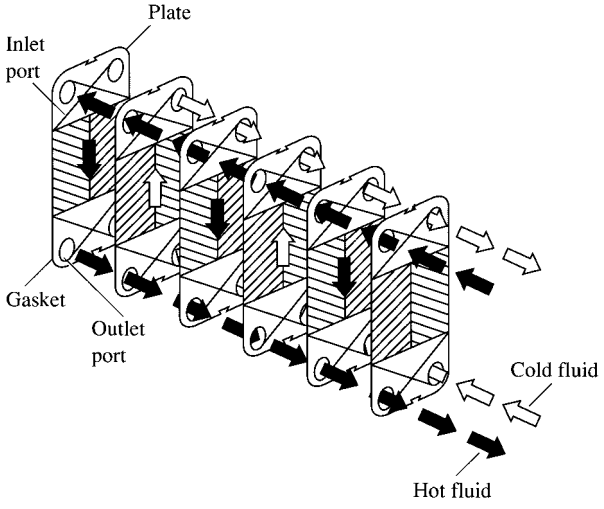


FIGURE 27.8 Plate-and-frame heat exchanger.

warm and cold liquids at a pressure drop of about 14.7 psi (101 kPa). The maximum working pressure for a high pressure model can be as high as 350 psig (2412 kPa).

The plate-and-frame heat exchanger has a high heat-transfer coefficient on the surfaces of its corrugated plates, a smaller temperature difference between the fluids on the two sides of each plate, and a compact size. It is easily dismantled for cleaning and routine maintenance.

27.7 DESIGN CONSIDERATIONS

Scope of Applications

In locations where the outdoor dry-bulb temperature $T_o \leq 95°F$ (35°C) and outdoor wet-bulb temperature $T'_o \leq 62°F$ (16.7°C), a stand-alone direct-indirect two-stage evaporative cooling system plus DX coil refrigeration can provide a comfortable indoor environment for occupants at a space temperature of 75°F (23.9°C) and a relative humidity of 45 to 50 percent, using evaporating cooling most of the time in the cooling season.

In locations where the outdoor dry-bulb temperature $T_o \leq 95°F$ (35°C) and outdoor wet-bulb temperature $T'_o \leq 65°F$ (18.3°C), a stand-alone indirect-direct two-stage evaporating cooling system can maintain a comfortable indoor environment for occupants at a space temperature of 75°F (23.9°C) and a relative humidity of 55 to 65 percent. In locations where the outdoor dry-bulb temperature $T_o \leq 95°F$ (35°C) and the outdoor wet-bulb temperature $T'_o \leq 70°F$ (21.1°C), a stand-alone indirect-direct two-stage evaporating system can maintain an indoor environment within the summer comfort zones for occupants at a space temperature of 78°F (25.6°C) and a relative humidity of 55 to 66 percent.

In locations where the outdoor wet-bulb temperature $T'_o \leq 72°F$ (22.2°C), an add-on evaporative cooler or an add-on evaporative cooler plus DX coil refrigeration save energy and is often economical. Life-cycle cost analysis is recommended, especially for small and medium-size systems or systems with a low number of operating hours during the cooling season.

In addition to the comfort cooling in residential and commercial buildings, evaporative cooling has been used in process cooling in industrial applications such as spot cooling, cooling of large motors, cooling of gas turbines and generators, textile mills, cooling of wood and paper products, laundries, animal barns and houses, and product storage cooling.

Evaporative cooling is best suited to applications where both summer cooling and winter humidification are required.

Beware of Dampness, Sump Maintenance, and Water Leakage

In direct evaporative coolers, cooled air contacts directly with liquid water. Select rigid wetted media instead of atomizing humidifiers in which water droplets may cause dampness or wetted surfaces at a certain distance downstream of the atomizing humidifiers. Sumps must be investigated and cleaned regularly. Water bleedoffs and necessary water treatments are essential to maintain a clean and efficient evaporative cooling system. The quantity of bleedoff should be no greater than the rate of evaporation. The water sump must be properly sealed so that water leaks are prevented. These considerations should be taken into account during system design.

Selection of Summer Outdoor Design Conditions

During the design of an evaporating cooling system, outdoor conditions should be selected to calculate the performance of the evaporative coolers; dry-bulb, wet-bulb,or dew point temperature; and whether this corresponds to 0.4, 1, or 2.5 percent annual accumulative frequency of occurrence and the mean coincident dry-bulb, wet-bulb, or dew point temperature.

Figure 27.9 shows the performance of an indirect-direct two-stage evaporating cooling VAV system if 1 percent dry-bulb and mean coincident wet-bulb (1 percent DB/MWB) temperatures, 1 percent wet-bulb and mean coincident dry-bulb (1 percent WB/MDB) temperatures, and 1 percent dew point and mean coincident dry-bulb (1 percent DP/MDB) temperatures of Denver, Colorado, are selected.

From *ASHRAE Handbook 1997, Fundamentals*, in Denver the outdoor design condition based on 1 percent wet-bulb temperature and the mean coincident dry-bulb temperature T_o' has a dry-bulb temperature of 80°F (26.7°C) and 63°F (17.2°C). In an indirect evaporative cooler, if its effectiveness is 0.60, the temperature of cooled air discharged from the cooler is

$$T'_{\text{ca},l} = 80 - 0.6(80 - 63) = 80 - 10.2 = 69.8°\text{F}(21.0°\text{C})$$

Draw a line from point ca,l' that parallels the thermodynamic wet-bulb temperature line. With the saturation efficiency of the direct evaporating cooler $\varepsilon_{\text{sat}} = 0.9$, the cooled air leaves this direct evaporating cooler with a dry-bulb temperature of 60.3°F (15.7°C) and a relative humidity of 95 percent.

If the space temperature is maintained at 75°F (23.9°C) and if the sensible heat ratio of the space conditioning line is 0.9, the space relative humidity can be determined from the psychrometric chart to be 60 percent. The supply temperature differential used to calculate the supply volume flow rate is $\Delta T_s = 75 - 60.3 = 14.7°\text{F}$ (8.2°C).

Similarly, the performance of the indirect-direct two-stage evaporating cooling system using 1 percent dry-bulb temperature and the mean coincident wet-bulb temperature, also 1 percent dew point temperature and the mean coincident dry-bulb temperature can be calculated as follows:

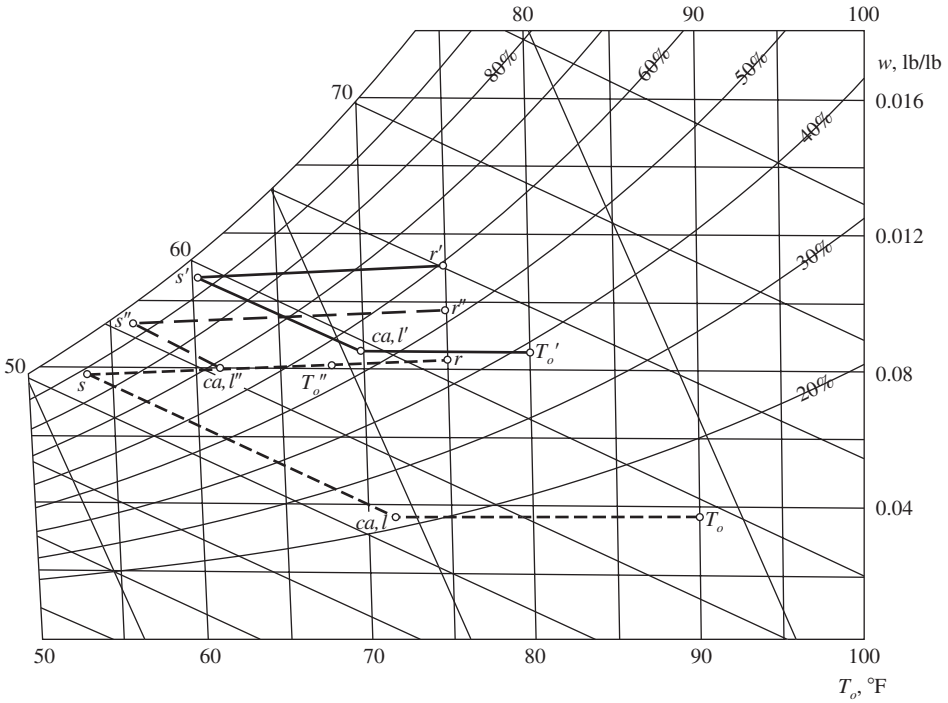


FIGURE 27.9 Comparison of 1 percent dry-bulb, 1 percent wet-bulb, and 1 percent dew point temperatures in Denver, Colorado.

Description	1 percent DB/MWB	1 percent WB/MDB	1 percent DP/MDB
Supply cooled air temperature T_s, °F	53.5	60.3	56.5
Space relative humidity, percent	44	60	52
Supply air differential ΔT_s, °F	21.5	14.7	18.7

From these analyses:

- If 1 percent DB/MWB is adopted as the outdoor air design condition, the design supply differential is 21.5°F (11.9°C). When outdoor air of 1 percent WB/MDB occurs, the volume flow rate must increase 21.5/14.7 = 1.46, or 46 percent, to maintain a space temperature of 75°F (23.9°C). Otherwise,the space temperature T_r may increase to 75 + 21.5 − 14.7 = 81.8°F (27.7°C).
- From Fig. 27.9, the temperature of supply cooled air is lower than others if 1 percent DB/MWB is adopted as the outdoor air design condition. The 1 percent DB/MWB does not include all the outdoor air at 1 percent WB/MDB and 1 percent DP/MDB.

The annual cumulative frequency of occurrence is no longer 1 percent and is greater than 1 percent, or 1.5 percent, or even more. If 1 percent WB/MDB is adopted as the outdoor air design condition, from Fig. 27.9, it includes more outdoor air at 1 percent DB/MWB and 1 percent DP/MDB than 1 percent DB/MWB, and the annual cumulative frequency of occurrence will be less than 1 percent DB/MWB.

The performance of an evaporative cooling system is more closely related to the outdoor wet-bulb temperature T'_o than to other outdoor parameters. The value of T'_o not only determines the temperature of cooled air leaving the direct cooler, but also has a decisive effect on the lowest possible wetted surface temperature in an indirect cooler. Outdoor air at a high dry-bulb temperature T_o and a low wet-bulb temperature T'_o can easily be cooled in a direct cooler. Outdoor air with a high wet-bulb temperature often needs refrigeration.

For a stand-alone indirect-direct two-stage evaporating cooling system with or without DX refrigeration that serves a more demanding project, a wet-bulb temperature corresponding to 0.4 percent annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature (0.4 percent WB/MDB) are recommended for the design outdoor air condition.

For a stand-alone two-stage evaporative cooling system or an add-on evaporative cooler, a wet-bulb temperature corresponding to 1 percent annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature (1 percent WB/MDB) are recommended for the design outdoor air condition.

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CHAPTER 28
AIR CONDITIONING SYSTEMS: SPACE CONDITIONING SYSTEMS

28.1 SPACE AIR CONDITIONING SYSTEMS 28.1
- Applications 28.1
- Advantages and Disadvantages 28.2
- Induction Systems 28.3

28.2 FAN-COIL SYSTEMS AND FAN-COIL UNITS 28.3
- System Description 28.3
- Operating Characteristics 28.3
- Fan-Coil Units 28.5
- Volume Flow Rate 28.7
- Cooling and Dehumidifying 28.8
- Heating Capacity 28.9
- Sound Power Level of Fan-Coil Units 28.9

28.3 FOUR-PIPE FAN-COIL SYSTEMS 28.9
- General Description 28.9
- Dedicated Ventilation System 28.10
- Space Recirculation Systems 28.11
- Temperature of Chilled Water Supplied to Coils 28.11
- Exhaust Air to Balance Outdoor Ventilation Air Intake 28.12
- Part-Load Operation 28.13
- Zone Temperature Control and Sequence of Operations 28.13
- System Characteristics 28.14
- Calculation of Operational Parameters 28.14
- Applications 28.20

28.4 TWO-PIPE FAN-COIL SYSTEMS 28.20
- Two-Pipe Systems 28.20
- Nonchangeover Two-Pipe Systems 28.20
- Changeover Two-Pipe Systems 28.23
- System Characteristics 28.24
- Applications 28.24

28.5 WATER-SOURCE HEAT PUMP SYSTEMS 28.24
- System Description 28.24
- Operating Characteristics 28.24
- Loop Temperatures 28.25
- Water-Source Heat Pumps 28.26
- Energy Performance and Energy Use Intensity of WSHPs 28.27
- Closed-Circuit Evaporative Water Cooler 28.27
- Water Heater 28.29
- Storage Tanks 28.29
- Air Systems and Maintenance 28.29
- Controls 28.30
- System Characteristics 28.31
- Case Studies: Water-Source Heat Pump Systems 28.31
- Design Considerations 28.32

28.6 PANEL HEATING AND COOLING 28.33

REFERENCES 28.33

28.1 SPACE AIR CONDITIONING SYSTEMS

Applications

As discussed in Sec. 1.5, space air conditioning systems, or simply space conditioning systems are one of the four major air conditioning systems (individual, packaged, space, and central) that comprised about 8 percent of total floor area of commercial buildings in 1992 in the United States. According to the survey of the Census Bureau, the shipments of fan-coil units comprised 215,000 units, and the shipment of water-source heat pumps comprised 99,000 units in 1994 in the United States. Space conditioning systems can be subdivided into the following air conditioning systems:

- Four-pipe fan-coil air conditioning systems, or simply four-pipe fan-coil systems
- Two-pipe fan-coil air conditioning systems, or simply two-pipe fan-coil systems
- Water-source heat pump systems

Advantages and Disadvantages

Space conditioning systems have the following advantages compared to central and packaged systems:

- Both heating and cooling devices in the form of terminals, such as fan coils or water-source heat pumps, are installed directly above or within the conditioned space, or very near to it. There is no return duct in a space conditioning system except the large core water-source heat pumps. Only a delivery outlet or a short recirculating duct supplies conditioned air to the perimeter zone.
- As discussed in Sec. 20.9, a separately dedicated outdoor ventilation system is always required to provide ventilation air for occupants as well as the necessary exhaust air. At the same time, a space recirculating system is used to condition the recirculating air for fan-coil and water-source heat pump systems. The required amount of ventilation is always guaranteed, and a dedicated outdoor air ventilation system facilitates the adoption of demand-controlled ventilation. The air economizer cycle for free cooling is only limited to the volume flow of the outdoor ventilation air.

  In space conditioning systems except the large core water-source heat pumps, the outdoor ventilation air duct is probably the only main duct with comparatively less headroom required than in the supply duct that crosses under the beams in the ceiling plenum.
- Both four-pipe fan coils and console water-source heat pumps provide individual zone control with each fan coil or each console water-source heat pump serving an individual zone. There is often an additional high-low speed or high-medium-low speed control on the volume flow of the supply fan. The four-pipe fan coil or the water-source heat pump can be changed from cooling mode to heating mode and vice versa automatically or manually, and it adjusts its capacity to meet the variation of the zone load during part-load operation.
- There is less cross-contamination between rooms and control zones.
- According to Zaidi and Howell (1993), the energy use of a four-pipe fan-coil system was only 73 percent compared with a VAV reheat system, and the energy use of a water-source heat pump was only 63 percent that of a VAV system. Anantapantula and Sauer (1994) found that the energy use of a four-pipe fan-coil system compared to a VAV reheat system was even lower. Lower energy use in fan-coil and water-source heat pump systems is due to the following:

  First, there is far less energy required to transport the conditioned air and the recirculating air; also, no energy is needed to transport the return air.

  Second, a fan-coil system can transfer excess heat from the interior zone of a building to the perimeter zone through a heat recovery system, as discussed in Sec. 13.3. A water-source heat pump operated in heating mode can absorb the heat rejected by another water-source heat pump that is operated in cooling mode and recovers it for heating in the perimeter zone.

Space conditioning systems have the following disadvantages:

- Only low-efficiency filters are used in fan coils and water-source heat pumps. Low efficiency air filters are unable to remove particulates of size < 3 μm and are ineffective at proving an acceptable IAQ for occupants as well as protecting the coils and components in the air distribution system from dust and contaminants.
- Fan-coil systems reduce the chilled water flow rate to decrease their cooling capacity during part load. Therefore, this results in a significantly higher zone relative humidity at part-load operation. The zone relative humidity may increase to 70 percent at part load, especially when the fan coil is oversized.
- Because fan coils and water source heat pumps are often mounted directly above or near to the conditioned space, they may create a noise problem, as discussed in Sec. 26.2, which is always

annoying. A receiver may receive a sound pressure level between 35 and 40 dBA in a conditioned space produced by a typical fan coil of 400 cfm (189 L/s).

- Fan coils and water-source heat pumps are scattered terminals. Both have moving parts, filters, and condensate pans and require site maintenance directly above, near, or within the conditioned space, which causes difficulties, inconvenience, and possible leaks of water and condensate. In addition, because only low-efficiency or coarse filters are used in the fan coils and water-source heat pumps, more cleaning work is required for the coils, ducts, and air distribution components.
- The volume flow rate of outdoor ventilation air supplied to the conditioned space is not sufficient to pressurize the floor immediately above or below the fire floor, to prevent smoke contamination in case of a building fire.

Induction Systems

In an induction system, recirculating air extracted through the coil is due to a negative pressure formed adjacent to the high-velocity supply nozzle. An induction system sees a considerably greater energy use than fan-coil systems and is not recommended in new and retrofit projects.

28.2 FAN-COIL SYSTEMS AND FAN-COIL UNITS

System Description

A four-pipe or a two-pipe fan-coil system includes boilers and chillers in the central plant, water system supplying chilled or hot water to the fan coils, a space recirculating system using fan coils to condition the space recirculating air, and a dedicated outdoor ventilation air system using an outdoor air AHU to condition the outdoor air, as shown in Fig. 28.1. Both the space recirculating and dedicated outdoor air systems have ducts, diffusers, inlets, controls, and accessories.

Outdoor air is often cooled and dehumidified, heated, or sometimes even humidified in a separate outdoor air or makeup AHU. The outdoor air is then transported to the fan coils where outdoor air is mixed with the recirculated air and conditioned. If the conditioned outdoor air is supplied directly to the conditioned space via supply ducts in the dedicated ventilation system, then it is mixed with the conditioned recirculating air from the fan coil of the space recirculating system in the space.

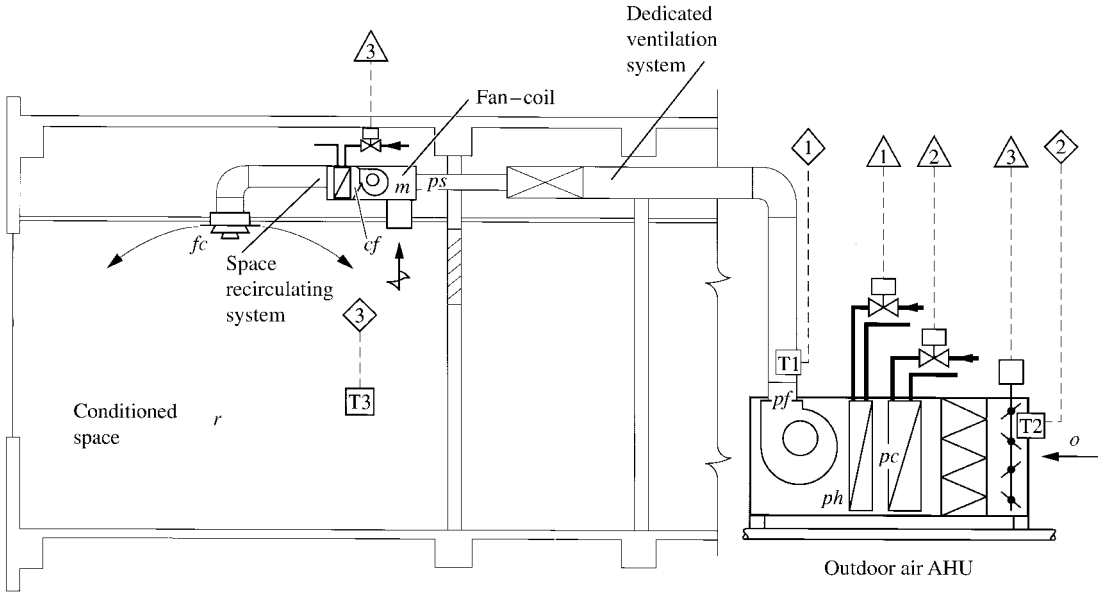
Outdoor (primary) air mixing with recirculating air in the outdoor air AHU is not economical because recirculating air must be transported back to the outdoor air AHU and conditioned there instead of in the fan coil in or near the conditioned space. Normally, there are no return fans or return ducts in four-pipe or two-pipe fan-coil systems.

Chilled water or hot water is supplied to the fan coil through a two-pipe or four-pipe water system. In locations with a moderate winter, or where electric rates are favorable in winter, an electric heating coil in each fan-coil unit is sometimes used for winter heating. This greatly simplifies the operation of fan-coil systems.

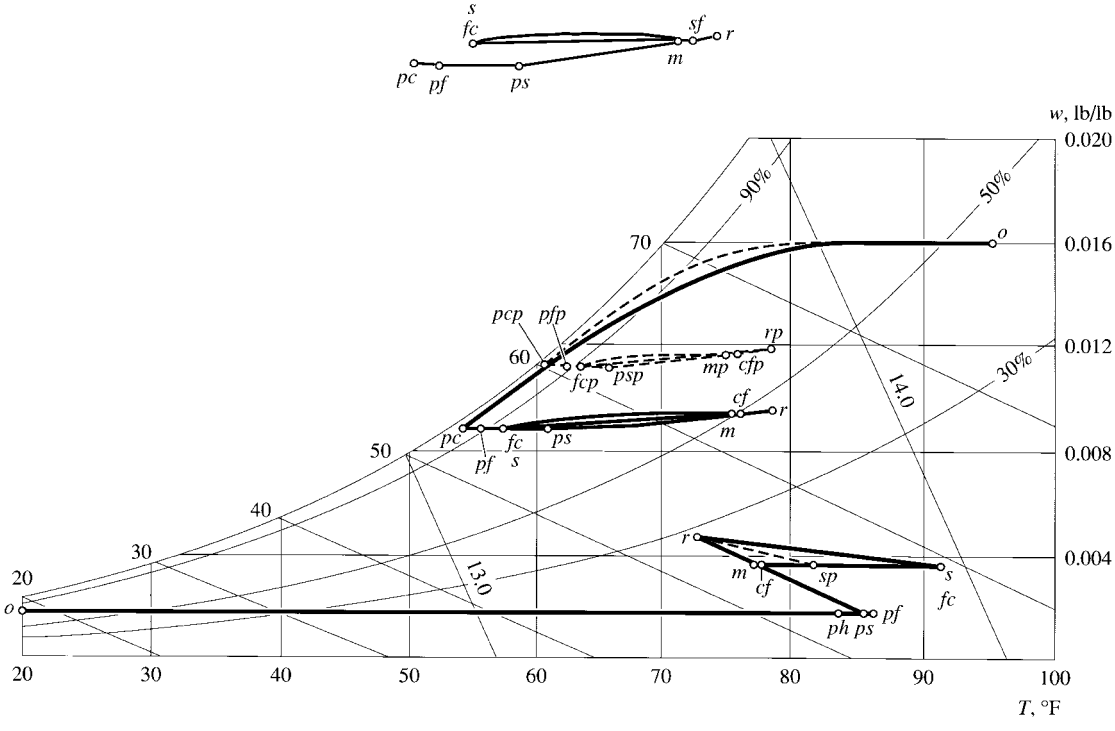
Four-pipe and two-pipe fan-coil systems and water-source heat pump systems are the air conditioning systems that use dedicated ventilation and space recirculating air systems.

Operating Characteristics

During summer cooling mode operation, outdoor air at point *o* in Fig. 28.1 is cooled and dehumidified, and filtered in the outdoor air AHU. The outdoor air leaves the cooling coil at point *pc*, flows through the supply fan outlet at point *pf* in the dedicated ventilation system, and is transported by the supply duct to the fan coil in the space recirculating systems or directly to the conditioned space



(a)



(b)

FIGURE 28.1 A four-pipe fan-coil system with outdoor air supplied to the mixing plenum of fan-coil unit: (*a*) schematic diagram; (*b*) air conditioning cycle.

at point *ps*. At *ps*, the dedicated ventilation system is combined with the space recirculating system and forms the air system of the four-pipe or two-pipe fan-coil system. The outdoor air can be mixed with the recirculating air in three ways:

- Outdoor air at point *ps* is supplied to the mixing plenum of the fan coil. It is then mixed with the recirculating air, which is extracted by the fan in the fan-coil unit through the filter and forms the mixture *m*. Air pressure at the mixing plenum is slightly less than the space pressure, so that space air can be extracted into the plenum. The mixture then flows through the fan and coil in the fan-coil unit, cools and dehumidifies at the coil, and leaves the coil at point *fc*. Air that has been conditioned in the fan coil is supplied to the conditioned space to offset the space load. State point *fc* may or may not be on line *pc-ps*, but will be near point *pf*.
- Outdoor air at point *ps* is supplied to the conditioned space directly. In this case, space air is filtered, and cooled and dehumidified (or heated), in the fan coil. The conditioned space air discharged from the fan-coil unit at point *fc* is mixed with the outdoor air in the space. It is assumed that the mixture at point *m* offsets the space load. In such an arrangement, the recirculating airstream will recirculate along its own circuit.
- Outdoor air at point *ps* is supplied just before the supply outlet of the fan-coil unit. It mixes with the conditioned air from the fan coil at point *fc*. The psychrometric cycle of this arrangement is approximately the same as that of outdoor air supplied directly to the conditioned space.

Outdoor air supplied to the mixing plenum or before the supply outlet of the fan-coil unit provides a balanced distribution of outdoor air in the conditioned space. On the other hand, outdoor air supplied directly to the conditioned space has a shorter outdoor air supply duct, and the conditioned air is better distributed when one of the fan-coil units is turned off during capacity control at part-load operation, as multiple fan-coil units are installed in a large room.

The temperature increase from fan power heat gain in the fan-coil unit is about 0.5°F (0.3°C) for a permanent-slit capacitor fan motor and about 0.8°F (0.5°C) for a shaded-pole fan motor. The supply duct in the ceiling plenum after the fan-coil unit is usually very short, so such a duct heat gain or loss is negligible.

During winter heating mode operation, in extremely cold weather, outdoor air may be preheated to point *ph* by the heating coil in the outdoor air AHU. After absorbing the supply fan power to point *pf* and releasing the outdoor air supply duct heat loss to point *ps* in the dedicated ventilation system, the outdoor air is mixed with recirculating air extracted by the fan coil at point *r*, and forms a mixture *m*, as shown by line *ps-m-r* in Fig. 28.1*b*. The mixture is heated in the fan coil to point *cf* by the fan power heat gain and then to point *s* by the heat released from the coil in the space recirculating system, and the mixture is supplied to the conditioned space.

Outdoor air can also be mixed with heated air from the fan-coil unit when outdoor air is supplied directly to the conditioned space. In locations with a mild winter, only outdoor air is heated in the outdoor air AHU.

Fan-Coil Units

A fan-coil unit, or a fan coil, is a terminal unit installed directly inside the conditioned space or in the ceiling plenum just above the conditioned space. A fan-coil unit includes a small motor driven centrifugal fan or two small centrifugal fans connected in parallel, a finned coil, a filter, an outer casing, and controls. Sometimes, a cooling coil and a heating coil may be connected in series along the airflow, as shown in Fig. 28.2.

A fan-coil unit can be a horizontal unit installed inside the ceiling plenum (Fig. 28.2*a* and *b*) or a vertical unit mounted on the floor under the windowsill (Fig. 28.2*c*) or a stack unit installed vertically along the two sides of the window. Vertical and stack units are usually used to offset the cold draft on the inner surface of window glass or on the external wall during cold weather. Cold draft often flows downward along the glass because it is heavier than the surrounding air. Warm air

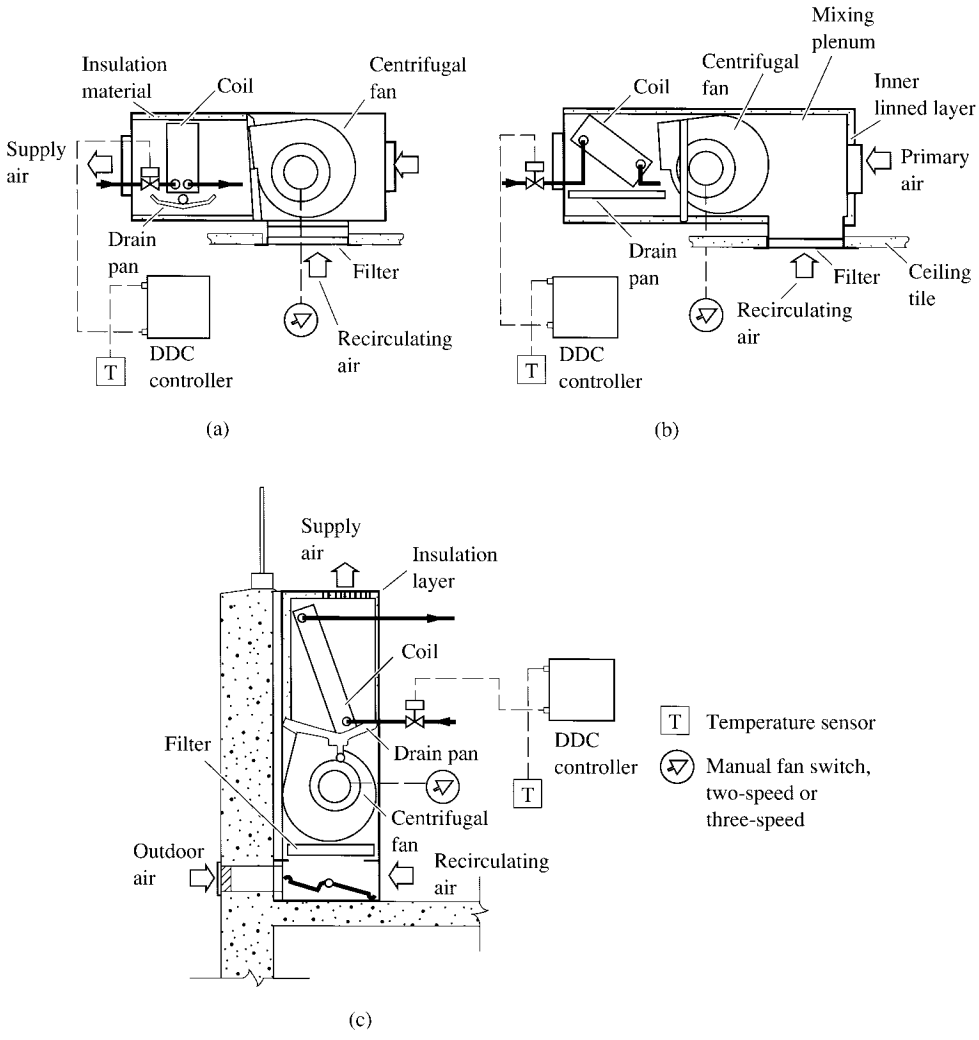


FIGURE 28.2 Fan-coil units: (*a*) horizontal unit; (*b*) horizontal unit with inner lined plenum; (*c*) vertical unit.

discharged from a fan coil during winter heating raises the inner surface temperature of window glass. Horizontal and vertical fan-coil units are shown in Fig. 28.2. Fan-coil units are available in standard sizes 02, 03, 04, 06, 08, 10, 12, 16, and 20. Size 02 means a nominal flow rate of 200 cfm (0.1 m^3/s), 04 means 400 cfm (0.2 m^3/s), and so on.

Fan. Double-width, double-inlet, forward-curved centrifugal fans are usually used because of their compact size and lower noise level. The fan wheels are usually made of aluminum or galvanized steel with a diameter less than 10 in. (250 mm) in most cases. Fan housings are die-formed with integral scrolls and inlets.

Fan Motor. Permanent-split capacitor (PSC) motors and shaded-pole (SP) motors are used. As in Sec. 26.4, PSC motors have an efficiency of 55 to 70 percent which is considerably higher than that of SP motors, which have 30 to 40 percent efficiency.

Two-speed high-low switches or three-speed high-medium-low switches are used to vary the fan speed manually or automatically (by a DDC controller). Fan motors are generally protected by a thermal overload protector. Periodic oiling of the bearing (twice per year) is required.

Coils. Coils are usually made from copper tubes and aluminum fins. Cooling/heating coils usually have two, three, or four rows of fins, depending on the coil's cooling capacity and the sensible heat ratio of the cooling and dehumidifying process SHR_c. Two-row coils or three-row coils are widely used. Four-row coils have a greater dehumidifying capacity.

Usually, there is only one coil for both heating and cooling. A separate electric heating coil is sometimes used with two-stage step control in locations where the heating season is short or energy rates are low in winter. Manual air vents are installed to prevent the formation of air pockets inside the water circuit. A galvanized-steel pan with an insulating liner is often used to drain the condensate during dehumidification and to prevent outer surface condensation on the fan-coil unit.

To reduce the air-side pressure loss, the face velocity of the air flowing through the coil is usually from 200 fpm (1 m/s) to 300 fpm (1.5 m/s).

Filters. Usually low-efficiency, low-pressure-drop permanent filters are used. They are easy to clean and replace periodically. Sometimes disposable, low-efficiency fiberglass filters are used.

Casing. The external cabinet is usually made of 18-gauge (1.3-mm) galvanized-steel sheet with a corrosion-resistant surface coating. The cabinet is lined with insulation to prevent outer surface condensation.

Although there are fans, water coils, and filters in both fan-coil units and air-handling units, a fan-coil unit is distinguished from an AHU by the following characteristics:

	Fan coil	AHU
Classification of equipment	A terminal unit	Basic equipment in air system
Location	Under window sill or in ceiling plenum	Fan room
Volume flow, cfm (m^3/s)	≤ 2000 (1)	1200–50,000 (0.6–25)
Fan total pressure, in. WC (Pa)	≤ 0.6 (150)	≤ 6 (1500)
Sound power level, dB	Lower	Higher
Filter efficiency	Low	Medium or high
Coil row depth	2, 3, or 4 rows	2, 3, 4, 6, or 8 rows
Fin spacing	Fixed	Custom-made
Selection of unit	Based on cooling capacity	Based on volume flow rate

Volume Flow Rate

The volume flow rate of a fan-coil unit is affected by the following factors:

- The external pressure drop because of the filters, the supply duct, and the diffusers in the space recirculating system
- The position of the fan switch, whether is high-low or high-medium-low
- The elevation of the fan coil above sea level

The nominal flow rate is the flow rate of a fan coil whose external pressure drop is at a specific value when the fan switch is in the high position and the fan coil is at sea level. The higher the external pressure drop, the lower the volume flow rate. If the external pressure drop increases from 0.06 to 0.3 in. WC (15 to 75 Pa), the volume flow rate of the fan coil may decrease to 55 percent of its nominal value.

For a typical fan coil, the volume flow rate of fan switch position medium is about 80 percent of the high value, and low is only 70 percent of the high value.

Cooling and Dehumidifying

Because the face velocity of a specific fan-coil unit at nominal volume flow rate is nearly the same for various sizes, and because the outer surface area of a water cooling coil is divided into dry and wet parts during cooling and dehumidifying, the coil capacity and the sensible heat ratio of the cooling and dehumidifying process SHR_c depend on the following factors:

- Dry- and wet-bulb temperatures of entering air
- Entering water temperature
- Water temperature rise in the coil
- The surface area and number of rows, including both pipe surface and fin area, of the coil

For a 04 fan-coil unit with an entering air temperature of 80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb, and an entering water temperature of 45°F (7.2°C), if the water temperature rise is 10°F (5.6°C), then its cooling and dehumidifying capacity Q_{cc} varies from 11 to 14 MBtu/h (3.22 to 4.1 kW), and SHR_c varies from 0.65 to 0.80.

If the volume flow rate of a fan-coil unit deviates from the nominal value because of a greater external pressure drop or a higher altitude, its total cooling capacity should be corrected as follows:

$$Q_{c,c} = C_p C_a Q_{c,r} \tag{28.1}$$

where $Q_{c,c}$ = corrected total cooling capacity, MBtu/h (kW)
$Q_{c,r}$ = manufacturer's catalog listed cooling capacity at a specific nominal volume flow rate, MBtu/h (kW)
C_p, C_a = total cooling capacity correction factor for excessive external pressure drop and high altitude, respectively

The corrected sensible cooling capacity of the fan coil $Q_{cs,c}$, in MBtu/h (kW), is

$$Q_{cs,c} = C_{ps} C_{as} q_{cs,r} \tag{28.2}$$

where $Q_{cs,r}$ = manufacturer's catalog-listed sensible cooling capacity at nominal volume flow rate, MBtu/h (kW)
C_{ps}, C_{as} = sensible cooling capacity correction factor for excessive external pressure and high altitude, respectively

Correction factors C_p, C_{ps}, C_a, and C_{as} can be found in the manufacturer's catalog. If these data are not available, the following values can then be used for C_p and C_{ps}:

External pressure drop, in. WC (Pa)	C_p	C_{ps}
0.06 (15)	1	1
0.1 (25)	0.96	0.96
0.2 (50)	0.84	0.83
0.25 (63)	0.76	0.73

If the sensible heat factor of the cooling and dehumidification process $0.5 \leq \text{SHR}_c \leq 0.95$, for each 1000 ft (328 m) higher than sea level up to an altitude of 10,000 ft (3280 m);

$$C_a = 1 - \frac{0.01(\text{altitude, ft})}{1000}$$
$$C_{as} = 1 - \frac{0.03(\text{altitude, ft})}{1000} \tag{28.3}$$

Heating Capacity

The heating capacity of the selected fan coil can be found in the manufacturer's catalog. Usually, for the same fan coil, a greater heating capacity can be provided at winter design conditions because the temperature difference between the hot water and heated air is higher than that between chilled water and cooled air.

Sound Power Level of Fan-Coil Units

Because the fan-coil unit is usually located inside the ceiling plenum or directly under the window sill, the room effect and the short supply duct are often not sufficient to attenuate fan noise in the fan-coil unit. Therefore, sound power level is often an important factor to consider during the selection of a fan-coil unit.

For a typical fan-coil unit of size between 02 and 12, the sound power level L_p rating, measured in a reverberant room according to ARI Standard 443-70 for an octave band with a middle frequency of 1000 Hz, varies from 45.5 to 52 dB. For a fan-coil unit of size 04, because $L_p = 47.5$ dB for an octave band with a middle frequency of 1000 Hz, if the room effect is 7.5 dB, the space NC level is about 40. The greater the size of the fan-coil unit, the higher the NC level. Refer to the manufacturer's catalog for details.

28.3 FOUR-PIPE FAN-COIL SYSTEMS

General Description

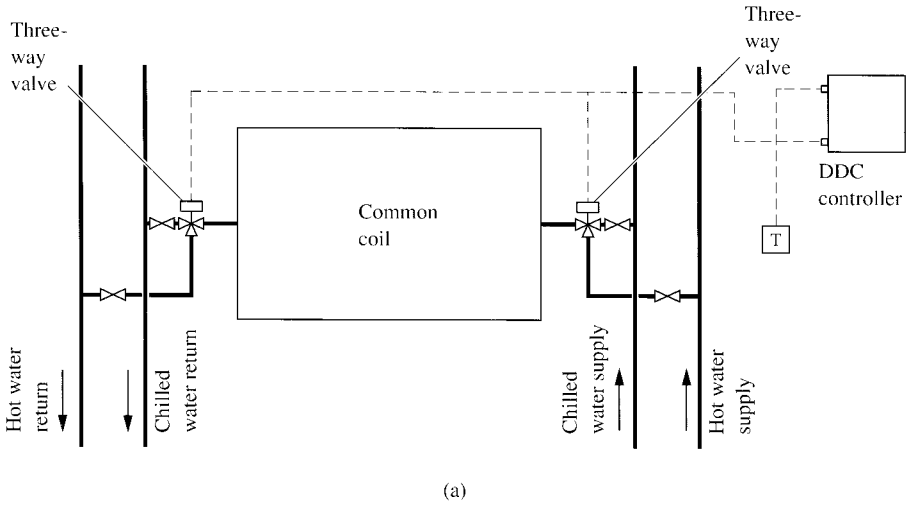
The water systems used for the fan coils can be classified into two categories: two-pipe systems and four-pipe systems. The three-pipe system was discontinued because of the energy loss in the common return pipe.

A four-pipe fan-coil system is equipped with two supply mains: a chilled water supply and a hot water supply. There are also two return mains: a chilled water return and a hot water return, as shown in Fig. 28.3. The finned coil may be a common coil, as shown in Fig. 28.3*a*, or two separate coils, a cooling coil of two or three rows, and a heating coil of one row, as shown in Fig. 28.3*b*. In a common coil, the DDC controller admits the chilled water and hot water in sequence to two three-way valves and modulates their water flow at part-load operation to maintain a preset space temperature. The chilled water stream never mixes with the hot water in these three-way valves.

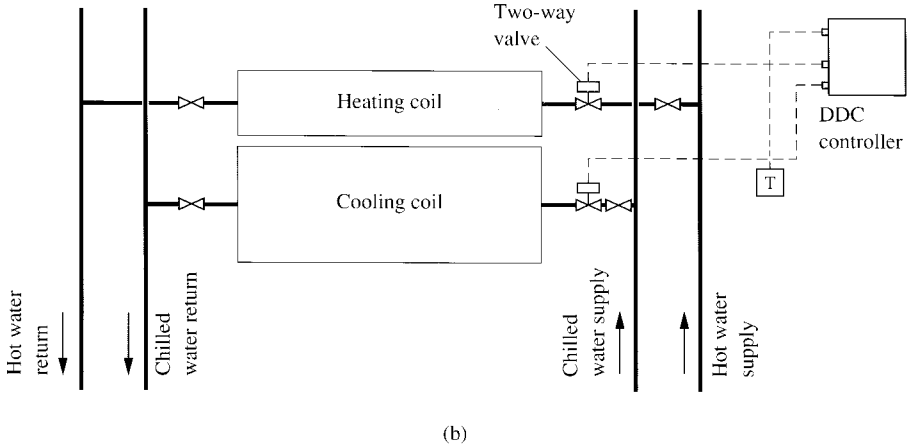
When two separate coils are used, a DDC controller admits chilled water to the cooling coil and hot water to the heating coil in sequence and modulates their water flow at part load. Hot water, steam, or electric energy can be used as the heat source.

A drainage pipe for condensate in the fan-coil unit is needed as the air is often cooled and dehumidified in the fan coil, especially where conditioned space may have hot and humid infiltrated outdoor air.

A four-pipe system is more flexible, easy to operate because there are no troublesome



(a)



(b)

FIGURE 28.3 Four-pipe fan-coil unit system: (*a*) common coil; (*b*) separate coil.

changeover problems, and lower in operating cost than a two-pipe system. On the other hand, it has a higher initial cost; and if reverse return pipe is used, many pipes must be squeezed into the ceiling plenum. Four-pipe fan-coil systems are most widely used.

Dedicated Ventilation System

The dedicated ventilaton system in a four-pipe fan-coil system must provide minimum outdoor air $\dot{V}_p$, in cfm (L/s), specified for occupants according to ASHRAE Standard 62-1999, served by each fan coil. The estimated cooling capacity Q_{cpn}, in Btu/h (W), provided by the minimum outdoor air to offset the space cooling load can be calculated as

$$Q_{cpn} = 60\dot{V}_p\rho_{ps}(h_r - h_{ps}) \tag{28.4}$$

where ρ_{ps} = air density of outdoor air, lb/ft^3 (kg/m^3)
h_r, h_{ps} = enthalpy of space air and outdoor air, Btu/lb (J/kg)

The estimated sensible cooling capacity of the minimum outdoor air $Q_{s,pn}$, in Btu/h (W), that offsets the space sensible cooling load can be calculated as

$$Q_{spn} = 60\dot{V}_p\rho_{ps}c_{pa}(T_r - T_s) \tag{28.5}$$

where T_r, T_s = temperature of space air and outdoor air, °F (°C). The total cooling capacity Q_{cp} and the total sensible cooling capacity Q_{sp} of the cooling coil in the outdoor air AHU, both in Btu/h, are the sum of the cooling capacities required to condition the outdoor air supplied to each of the fan coils served by this outdoor air AHU, and they can be calculated as

$$Q_{cp} = \sum_1^n Q_{cpn} \qquad Q_{sp} = \sum_1^n Q_{spn} \tag{28.6}$$

where n = number of fan coils served by the outdoor air AHU.

Space Recirculation Systems

Each fan-coil unit forms a space recirculation system. Its system total pressure varies between 0.06 and 0.6 in. WC (15 and 150 Pa).

Estimated Cooling Capacity. The size of a fan-coil unit is selected to meet the required cooling capacity and sensible cooling capacity to offset the space cooling load Q_{rc} and sensible cooling load Q_{rs}, both in Btu/h (W). As the conditioned outdoor air supplied to a specific fan coil or to the conditioned space served by the fan coil offsets part of the space load, the estimated total cooling capacity $Q_{c,\text{fc}}$ and sensible cooling capacity $Q_{s,\text{fc}}$ of a fan coil, both in Btu/h (W), for a specific control zone or conditioned area it serves can be calculated as

$$\begin{aligned} Q_{c,\text{fc}} &= Q_{rc} - Q_{c,p} = 60\dot{V}_p\rho_{ps}(h_r - h_{ps}) \\ Q_{s,\text{fc}} &= Q_{rs} - Q_{sp} = 60\dot{V}_p\rho_{ps}c_{pa}(T_r - T_{ps}) \end{aligned} \tag{28.7}$$

where h_r, h_{ps} = enthapy of zone air and conditioned outdoor supply air, Btu/lb (J/kg)
T_r, T_{ps} = temperature of zone air and conditioned outdoor supply air, °F (°C)
$\dot{V}_p$ = volume flow rate of outdoor (primary) air supplied to each fan coil, cfm [m^3/(60 s)]

Sensible Heat Ratios. When a four-pipe fan-coil system is operated under steady equilibrium, the sensible heat ratio of the space conditioning line SHR_s is usually approximately equal to the sensible heat ratio of the cooling and dehumidifying process of the fan coil $\text{SHR}_{c,f}$ because the cooling and dehumidifying process in the fan coil *cf-fc* and the space conditioning line *fc-r* form a cycle.

During summer design conditions, the temperature of chilled water supplied to the fan coils and to the outdoor air AHU is usually the same, $T_{w,p} = T_{w,f}$, both in °F (°C). Also, the humidity ratios of the conditioned outdoor air at point *ps* and the air leaving the fan-coil point *fc* are approximately equal to each other, $w_{ps} \approx w_{\text{fc}}$, both in lb/lb (kg/kg). For fan-coil units serving the interior zone, more often sensible ratios $\text{SHR}_s \approx \text{SHR}_{c,f} \approx 0.8$. For fan-coil units serving a perimeter zone, $\text{SHR}_s \approx \text{SHR}_{c,f} \approx 0.95$, as shown in Fig. 28.1*b*.

Temperature of Chilled Water Supplied to Coils

In a four-pipe fan-coil system, the space or zone temperature is a specified design criterion within the ASHRAE comfort zones. In summer, it is usually between 75 and 78°F (23.9 and 25.6°C). The humidity ratios of the supply air from the outdoor air AHU w_{ps} and the supply air from the fan coil w_{fc}, as well as the humidity ratio of the space air w_r depend on the temperature of the chilled water

supplied to the fan coils $T_{w,f}$ and to the outdoor air AHU $T_{w,p}$. Also $T_{w,f}$ and $T_{w,p}$ have a direct influence on the cooling capacity of the fan coils and the outdoor air AHU.

If $T_{w,p} < T_{w,f}$, then $w_{ps} < w_{\text{fc}}$, and $w_{ps} < w_r$. A considerable amount of both sensible and latent load is undertaken by the outdoor air during cooling mode operation. The required cooling capacities of the fan-coil units are considerably less than those of the space cooling load.

If $T_{w,p} = T_{w,f}$, $w_{ps} \approx w_{\text{fc}}$, and $w_{ps} < w_r$, there is a certain amount of sensible load and a comparatively less amount of latent load undertaken by the outdoor air supplied from the outdoor air AHU during cooling mode operation. The required sensible cooling capacity of the fan coils is smaller than the space sensible cooling load.

In the early days, the chilled water temperature supplied to the fan coils $T_{w,f}$ was deliberately equal to or higher than the dew point of the entering air to the fan coil $T_{f,\text{en}}''$, so that there was no condensate in the fan coil. The coil was easier to maintain and clean. Only dehumidified outdoor air was used to offset the space latent load. However, many four-pipe fan-coil systems are now designed so that $T_{w,p} = T_{w,f}$, as shown in Fig. 28.1*b*, because of the following:

- In locations where outdoor climate is hot and humid, the infiltrated air most probably forms condensate in the fan coil
- It is less expensive as well as more convenient to control and operate when $T_{w,p} = T_{w,f}$.

Exhaust Air to Balance Outdoor Ventilation Air Intake

First, as discussed in Sec. 20.3 and expressed in Eq. (20.8), at a steady state, the mass flow rate of entering air must be equal to the mass flow rate of leaving air, and

$$\dot{m}_o + \dot{m}_{\text{rec}} + \dot{m}_{\text{inf}} = \dot{m}_{\text{rt}} + \dot{m}_{\text{ex}} \tag{28.8}$$

where $\dot{m}_o$, $\dot{m}_{\text{rec}}$ = mass flow rate of outdoor air intake and recirculating air, lb/min (kg/min), and $\dot{m}_s = \dot{m}_o + \dot{m}_{\text{rec}}$. If the designed entering air is exceeds the leaving air, the space will be pressurized to a higher space pressure, a portion of space air is then exfiltrated, and a new balance is formed with an additional term $-\dot{m}_{\text{inf}}$.

Second, for a typical office building, the required outdoor ventilation air intake $\dot{V}_{o,\text{sys}}$ is 0.14 cfm/ft^2 (0.70 L/s·m^2) of floor area. The exhaust in the rest rooms is about 0.03 cfm/ft (0.15 L/s·m^2), which results in an excessive outdoor ventilation air of $0.14 - 0.03 = 0.11$ cfm/ft^2 (0.56 L/s·m^2) of floor area. If there is no exhaust system to exhaust this excessive amount of outdoor air to outdoors, then all the excessive outdoor air will be squeezed out through the leakage area on the exterior walls at a pressurized space-outdoor pressure difference Δp_{ro}.

For an average leaky office building that has an exterior wall area of 10,000 ft^2 (929 m^2) and a floor area of 20,000 ft^2 (1858 m^2), as discussed in Sec. 23.2, its effective leakage area is 2.08 ft^2 (0.193 m^2). From Eq. (23.4), the space and outdoor pressure differences to exfiltrate all the excessive outdoor ventilation air are as follows:

	$A_{e,l}$, ft^2 (m^2)	Δp_{ro}, in. WC (Pa)
Average leaky	2.08 (0.193)	0.07 (17)
Leaky	4.16 (0.387)	0.017 (4.3)

Therefore, the following is recommended when a four-pipe fan coil system is used:

- For leaky buildings, outdoor ventilation air exhaust is not required.
- For average leaky buildings, the space pressurization due to excessive outdoor ventilation air should be calculated and analyzed. If the pressurized space-outdoor pressure difference $\Delta p_{ro} \leq 0.03$ in. WC (7.5 Pa), outdoor ventilation exhaust due to excessive outdoor ventilation air is not recommended.

Part-Load Operation

At cooling mode part-load operation as the zone sensible cooling load decreases, the zone temperature decreases accordingly. In a four-pipe fan-coil system, when the zone temperature sensor senses this temperature decrease, a DDC controller closes the two-way valve in a fan coil, reducing the chilled water flow rate entering the coil. As the chilled water flow reduces, the temperature of chilled water leaving the fan coil T_{wfl} rises accordingly, which raises both the temperature and humidity ratio of the conditioned (cooled and dehumidified) air leaving the fan coil T_{fcp} and w_{fcp} as well as the humidity ratio of the zone air w_{rp}, as shown in Fig. 28.1*b*. As soon as the reduction in the sensible cooling effect of the conditioned air due to the increase of T_{fcp} in $Q_{s,fcp} = 60\dot{V}_{fc}\rho_s c_{pa}$ $(T_r - T_{fcp})$ is just equal to the reduced sensible zone cooling load, a new balance is formed and the zone temperature is maintained at the indoor design temperture T_r.

At cooling mode part-load operation, the chilled water temperature entering the fan coil and the cooling coil of the outdoor air AHU T_{wfe} is better reset 5 to 7°F (2.8 to 3.9°C) higher, according to either the outdoor air temperature or the system load. Chilled water temperature reset can save compression energy and reduces the sensible cooling effect of the conditioned air. On the contrary, it also increases the zone relative humidty. Analyses should be done to select an appropriate value.

During cooling mode part-load operation, reducing the chilled water flow rate entering a cooling coil to match the reduction in zone or space load always causes a higher zone relative humidty. But reducing the supply air volume flow rate to match the reduction in zone or space load in VAV systems causes the zone relative humidity to remain approximately the same.

Zone Temperature Control and Sequence of Operations

The zone temperature control of a four-pipe fan-coil system includes a fan-coil unit FC-1, using a separate cooling coil and a heating coil that serve a control zone *rx*1 in the perimeter zone, and the corresponding outdoor air AHU which supplies the conditioned outdoor ventilation air to the fan coil FC-1. The operating mode can be divided into cooling mode, deadband mode, and heating mode.

1. When the time-of-day clock signals the four-pipe fan-coil system in the off position, and the outdoor air AHU and fan-coil unit serving control zone *rx*1 in the perimeter zone is shut off, the two-way valves for both cooling and heating coils in the fan coil FC-1 should be closed. Also the outdoor air damper and the supply fan in the outdoor air AHU (OAAHU) are shut off.

2. When the time-of-day clock signals the four-pipe fan-coil system to the on position, and the perimeter zone temperature sensor senses a zone temperature $T_{rx} > 75$°F (23.9°C), both the outdoor air AHU and fan coil FC-1 call for cooling. A DDC system controller then opens the two-way valves, resets the discharge air temperature T_{dis}, and starts the supply fan motor of the OAAHU with the outdoor damper closed. After a time delay, such as 30 s, the outdoor damper of the OAAHU is fully opened.

3. A DDC terminal controller next opens the two-way valve of the cooling coil of the fan coil, starts the fan motor and the small fan, and modulates the valve opening to maintain a zone temperature of 75°F (23.9°C). The two-way valve of the heating coil remains closed.

4. During cooling mode part-load operation, the zone sensible cooling load reduces. When the drop in zone temperature T_{rx1p} is sensed by the temperature sensor of control zone *rx*1, a DDC terminal controller reduces and modulates the opening of the two-way valve of the cooling coil in the fan coil, to maintain a preset zone temperature of 75°F (23.9°C).

Because of the reduction in the chilled water flow in the cooling coil, the result is an increase in the temperature of chilled water leaving the cooling coil as well as in the temperature and humidity ratio of the cooled conditioned air leaving the coil, which results in a higher zone relative humidity φ_{rp}, as shown in Fig. 28.1*b*.

Usually, the sensible heat ratio of the space conditioning line (SHR_{sp}) at part load is smaller than that at summer design conditions. This further increases the zone relative humidity.

5. When the space cooling load is reduced to 50 percent of the design load, a DDC system controller resets the chilled water temperature in linear proportion from 45 to 50°F (7.2 to 10.0°C) according to a preset range of outdoor temperature T_o. Also, a DDC terminal controller reduces the fan speed in the fan coil from high to low.

6. When the zone temperature drops within the range 72.0°F $< T_{rx1p} \leq$ 75°F (22.2°C $< T_{rx1p} \leq$ 23.9°C), the fan coil FC-1 is operated in deadband mode. In deadband mode, the DDC terminal controller actuates the following:

- Both two-way valves for cooling and heating coils in the fan-coil are closed.
- The fan is still operating at low speed to provide air movement and filtration.

7. When the zone temperature $T_{rx1} \leq$ 72.0°F (22.2°C), the fan coil FC-1 is operating in heating mode. In heating mode operation:

- The modulation of the hot water flow by the DDC terminal controller should be reverse-acting; i.e., the higher the sensed zone temperature T_{rx1}, the smaller the output; and the lower the sensed T_{rx1}, the greater the output.
- The DDC controller modulates the two-way valve opening of the heating coil to maintain a zone temperature T_{rx1} of 72°F (22.2°C).
- The fan in the fan coil is operating at low speed.

8. In heating mode, the heating process in the fan coil is a horizontal line *ms* with a constant humidity ratio (as shown in Fig. 28.1*b*). The discharge temperature of the OAAHU $T_{o,\text{dis}}$ is often reset between 80 and 100°F (26.7 and 37.8°C) according to outdoor temperature T_o through a DDC system controller. The lower the outdoor temperature T_o, the higher $T_{o,\text{dis}}$. When the outdoor temperature T_o drops to a preset value, the fan in the fan coil raises its speed to high speed.

During heating mode part-load operation, if the outdoor humidity ratio w_o is higher than the winter design conditions, the humidity ratio of the mixture of outdoor and recirculating air w_m is raised accordingly. The result is a higher zone relative humidity which is more comfortable to occupants in winter

Systems Characteristics

System characteristics of four-pipe fan-coil (4PFC) systems are listed in Table 28.1.

Calculation of Operational Parameters

Consider a large office in an office building using a four-pipe fan-coil system. The outdoor ventilation air from the outdoor-air AHU is supplied to the mixing plenum of the fan-coil unit. At summer and winter design loads, the operating parameters for this four-pipe fan-coil system in this office room are as follows:

	Summer	Winter
Indoor space temperature, °F	78	72
Space relative humidity, percent (wet-bulb temperature)	45 (63.5°F)	
Space cooling or heating load, Btu/h	29,000	9500
Space sensible cooling load, Btu/h	26,200	
Space latent load, Btu/h		2700
Part-load, operation, cooling load, Btu/h	15,900	
Part-load operation, sensible cooling load, Btu/h	13,100	
Outdoor temperature, °F	90	20
Outdoor relative humidity, percent	50	80
Outdoor air, cfm	200	200

TABLE 28.1 System Characteristics of Space Conditioning Systems

	4PFC	2PFC	WSHP
Zone thermal and sound controls			
Control zone	Multizone	Multizone	Multizone
Control methods	DDC or electric	DDC or electric	DDC or electric
Control modes	PI or on/off	PI or on/off	PI or on/off
Heating-cooling changeover	Automatic	Nonchangeover; changeover, auto matic or manual	Automatic
Sound control	35–45 dBA	35–45 dBA	40–45 dBA
Indoor air quality			
Minimum ventilation air control	Guarantee minimum outdoor ventilation air supply		
Filters: FC or WSHPs	Low efficiency	Low efficiency	Low efficiency
Outdoor-air AHU	Medium to high efficiency		Medium to high efficiency
Air systems			
Space recirculating system	Fan coil	Fan coil	Water-source heat pump
Type of supply fan	Forward-curved centrifugal	Forward-curved centrifugal	Forward-curved centrifugal
Combined fan-motor-drive efficiency	25%	25%	Console, 25% Core, 30 to 40%
Volume flow control	High-low or high-medium-low fan speed	High-low or high-medium-low fan speed	Console, multi speed Core, constant speed
Space recirculating system total pressure, in. WC	0.06–0.6	0.06–0.6	Console, 0.06–0.6 Core, $\leq$ 3.0
Dedicated outdoor ventilation system	Constant-volume or DCV	Constant-volume or DCV	Constant-volume or DCV
Type of supply fan	Centrifugal	Centrifugal	Centrifugal
Combined fan-motor drive efficiency	45%	45%	45%
Volume flow control	Constant/variable-speed	Constant/variable-speed	Constant/variable-speed
Outdoor ventilation system total pressure, in. WC	3–4	3–4	3–4
Refrigeration systems			
Type	Centrifugal or screw	Centrifugal or screw	Reciprocating, rotary, or scroll
Refrigerants	HCFC-123, HCFC-22, or HFC-134a	HCFC-123, HCFC-22, or HFC-134a	HCFC-22, HFC-407C, or HFC-410A
Evaporator	Chiller	Chiller	DX coil
Condenser	Water-cooled	Water-cooled	Water-cooled
Cooling tower	Open-circuit	Open-circuit	Open-circuit, or closed-circuit
Refrigerant flow control	Orifice, float, or expansion valve	Orifice, float, or expansion valve	Capillary tube
Minimum performance, compressor, kW/ton	0.5–0.7	0.5–0.7	10–15 EER
Heating system			
Type	Hot water or electric heating	Hot water or electric heating	WSHP
Gas-fired boiler efficiency	AFUE 80 to 93%	AFUE 80 to 93%	AFUE 80 to 93%
Maintenance	More site maintenance	More site maintenance	More site maintenance
Fault detection and diagnostics	Chiller	Chiller	Core WSHPs

1. Assume that the temperature of outdoor air supplied from the OAAHU T_{ps} is 61°F (16.1°C) and the relative humidity is 81 percent. From the psychrometric chart, $h_{ps} = 24.8$ Btu/lb (57.7 kJ/kg). At summer design conditions, the enthalpy of space air is 28.8 Btu/lb (67.0 kJ/kg), and outdoor air density $\rho_{ps} = 0.075$ lb/ft^3 (1.2 kg/m^3); from Eq. (28.7) the estimated fan-coil cooling capacity is

$$Q_{c,\text{fc}} = Q_{rc} - 60\dot{V}_p\rho_{ps}(h_r - h_{ps})$$
$$= 29{,}000 - 60(200)(0.075)(28.8 - 24.8) = 25{,}400 \text{ Btu/h (7442 W)}$$

The estimated fan-coil sensible cooling capacity is

$$Q_{s,\text{fc}} = Q_{rs} - 60\dot{V}_p\rho_{ps}c_{pa}(T_r - T_{ps})$$
$$= 26{,}200 - 60(200)(0.075)(0.243)(78 - 61) = 22{,}482 \text{ Btu/h (6587 W)}$$

If the chilled water supplied to the fan coils and the outdoor air AHU are both at 45°F (7.2°C) during summer design conditions, for an entering air of 78°F (25.6°C) dry bulb and 63.5°F (17.5°C) wet bulb temperatures and a chilled water temperature rise of 10°F (5.6°C), one manufacturer's catalog gives the following capacities for a size 03 fan-coil unit:

Total cooling capacity	7.2 MBtu/h (2.1 kW)
Sensible cooling capacity	6.3 MBtu/h (1.8 kW)

Because the external pressure drop in the supply duct after the fan coil is 0.06 in. WC (15 Pa) and the fan coils are installed at sea level, no volume flow corrections are needed.

Four 03 fan-coil units are selected, so that the total cooling capacity is

$$Q_{c,\text{fc}} = 4 \times 7200 = 28{,}800 \text{ Btu/h (8438 W)}$$

which is greater than the estimated cooling capacity of 25,400 Btu/h (7442 W) at summer design load. The sensible cooling capacity of fan coils is

$$Q_{s,\text{fc}} = 4 \times 6300 = 25{,}200 \text{ Btu/h (7384 W)}$$

which is also greater than the estimated sensible cooling capacity of 22,482 Btu/h at summer design load.

2. Draw line *r-fc* from the conditioned zone point *r* with known $\text{SHR}_s = 26{,}200/29{,}000 = 0.9$, and a $\rho_{\text{fc}} = 1/13.7 = 0.73$ lb/ft^3. The temperature of supply air leaving the fan-coil unit T_{fc}, in °F (°C), can be calculated as

$$T_{\text{fc}} = T_r - \frac{Q_{rs}}{60\dot{V}_{\text{fc}}\rho_{\text{fc}}c_{pa}}$$
$$= 78 - 26{,}200/60 \times 4 \times 300 \times 0.073 \times 0.243 = 57.5°\text{F (14.2°C)} \qquad (28.9)$$

where $\dot{V}_{\text{fc}}$ = corrected volume flow rate of fan coil, cfm (L/s)
ρ_{fc} = air density at fan outlet in fan coil, lb/ft^3 (kg/m^3)

Point *fc* can then be determined. From the psychrometric chart, $h_{fc} = 23.3$ Btu/lb (54.2 kJ/kg).

3. Let ΔT_{wp} be the water temperature rise in the cooling coil of the outdoor air AHU. It can be assumed that the temperature of air leaving the cooling coil of the outdoor air AHU T_{pc}, in°F (°C), is

$$T_{pc} = T_{w,p} + \Delta T_{wp} \qquad (28.10)$$

For $\Delta T_{wp} = 10$°F, $T_{pc} = 45 + 10 = 55$°F (12.8°C).

Generally, the cooling coil may have six or eight rows. The relative humidity of air leaving the cooling coil $\varphi_{pc} = 98$ percent, and the humidity ratio w_{pc} can thus be determined. For $T_{pc} = 55°F$ (12.8°C) and $\varphi_{pc} = 98$ percent, $w_{pc} = 0.0092$ lb/lb (0.0092 kg/kg) and $h_{pc} = 23.2$ lb/lb (54.0 kJ/kg).

For simplicity, the fan power heat gain for the OAAHU can be assumed as 2°F, and the duct heat gain can be estimated at 4°F, so the supply temperature of the OAAHU T_{ps}, in °F, is

$$T_{ps} = T_{pc} + 2 + 4 = 61°F\ (16.1°C) \tag{28.11}$$

Because $w_{ps} = w_{pc}$, state point *ps* can also be determined.

4. For a fan coil whose outdoor air is mixed with recirculating air in the mixing plenum of the fan coil, draw line *r-ps*. From Fig. 28.1b,

$$\frac{m\text{-}r}{ps\text{-}r} = \frac{\dot{V}_p}{\dot{V}_{\text{fc}}} = \frac{200}{4 \times 300} = 0.17 \tag{28.12}$$

Point *m* can thus be determined. In this case, $T_m = 75.0°F$ (23.9°C), and $w_m = 0.0092$ lb/lb (0.0092 kg/kg). Because $T_{cf} = T_m + 0.5 = 75 + 0.5 = 75.5°F$ (24.2°C), and $w_{cf} = w_m = 0.0092$ lb/lb (0.0092 kg/kg), point *cf* can also be determined. From the psychrometric chart, $h_{cf} = 28.2$ Btu/lb (65.6 kJ/kg).

5. If $\rho_{cf} = 1/13.7 = 0.073$ lb/ft^3 (1.168 kg/m^3), the cooling coil load $Q_{c,\text{fc}}$ and sensible cooling coil load $Q_{s,\text{fc}}$ of the fan-coil unit, both in Btu/h, can be calculated as

$$\begin{aligned} Q_{c,\text{fc}} &= 60\dot{V}_{\text{fc}}\rho_{cf}(h_{cf} - h_{\text{fc}}) \\ &= 60 \times 4 \times 300 \times 0.073(28.2 - 23.3) = 25{,}754\ \text{Btu/h}\ (7546\ \text{W}) \end{aligned} \tag{28.13}$$

$$\begin{aligned} Q_{s,\text{fc}} &= 60\dot{V}_{\text{fc}}\rho_{cf}c_{pa}(T_{cf} - T_{\text{fc}}) \\ &= 60 \times 4 \times 300 \times 0.073 \times 0.243(75.5 - 57.5) = 22{,}990\ \text{Btu/h}\ (6736\ \text{W}) \end{aligned} \tag{28.14}$$

where h_{cf}, h_{fc} = enthalpy of air at fan outlet and supply outlet, as shown in Fig. 28.1*a*, Btu/lb (J/kg)
T_{cf}, T_{fc} = temperature of air at fan outlet and supply outlet, °F (°C)

Check the calculated cooling coil load and sensible load against the selected fan coil's cooling and sensible capacity. The calculated coil load $Q_{c,\text{fc}}$ of 25,754 Btu/h (7546 W) is very near to the estimated capacity of 25,400 Btu/h (7442 W), and both are less than the selected capacity $Q_{c,\text{fc}}$ of 28,800 Btu/h (8438 W). Similarly, the calculated sensible coil load $Q_{s,\text{fc}}$ is 22,990 Btu/h (6736 W). It is slightly greater than the estimated 22,482 Btu/h (6587 W) because of the fan power heat gain. Both are less than the selected capacity $Q_{s,\text{fc}}$ of 24,000 Btu/h (7032 W).

6. Calculate the cooling coil load in the outdoor air AHU Q_{cp}, in Btu/h (W). From the psychrometric chart, $T_o = 90°F$, $\varphi_o = 50$ percent, and $h_o = 38.5$ Btu/lb, so

$$\begin{aligned} Q_{cp} &= 60\dot{V}_p\rho_{ps}(h_o - h_{ps}) \\ &= 60 \times 200 \times 0.075(38.5 - 23.2) = 13{,}770\ \text{Btu/h}\ (4035\ \text{W}) \end{aligned} \tag{28.15}$$

The cooling coil load due to the outdoor air in the OAAHU Q_{cp} is about one-half of the fan coil's cooling coil load. This includes the cooling and dehumidifying load of outdoor air, the system heat gain of the dedicated outdoor ventilation air system, and a portion of the zone cooling load.

7. During cooling mode part-load operation, the zone temperature is still maintained at 78°F (25.6°C); that is, $T_{rp} = T_r$. The fan in the fan coil can be switched from high speed to low speed while chilled water is reset to a higher temperature. If the zone sensible load ratio is reduced to 50 percent of the full-load value and the volume flow rate of the fan at low speed is only 70 percent of the high speed, then the temperature of air leaving the fan coil at part load $T_{\text{fc},p}$, in °F (°C), can be

calculated as

$$T_{\text{fc},p} = T_{rp} - \frac{Q_{rsp}}{60 \times 0.7 \dot{V}_{\text{fc}} \rho_{cf} c_{pa}} \tag{28.16}$$

$$= 78 - \frac{0.5(26{,}200)}{60 \times 4 \times 300 \times 0.7 \times 0.073 \times 0.243} = 63.3°\text{F}(17.4°\text{C})$$

where Q_{rsp} = zone sensible cooling load at part load, Btu/h (W)
T_{rp} = zone air temperature at part load, °F (°C)

At part-load operation, the relative humidity of supply air leaving the fan coil $\varphi_{\text{fc},p}$ may vary between 80 and 90 percent. If $\varphi_{\text{fc},p} = 88$ percent, point *fcp* can be plotted on the psychrometric chart as shown in Fig. 28.1*b*. From the chart, $h_{fcp} = 27.2$ Btu/lb (63.3 kJ/kg).

8. During cooling mode part-load operation, the sensible heat ratio of space conditioning line $\text{SHR}_s = 13{,}100/15{,}900 = 0.82$. Draw line *fcp-rp* from *fcp* with a sensible heat ratio of the space conditioning line at part load $\text{SHR}_{sp} = 0.82$. Line *fcp-rp* intersects the zone temperature line T_{rp} at *rp*. Therefore, point *rp* can be determined. From the psychrometric chart, $\varphi_{rp} = 57.5$ percent.

9. When the space cooling load is reduced to 50 percent of design load, if the temperature of chilled water supplied to fan coils $T_{w,f}$ and that to the outdoor air AHU $T_{w,p}$ are reset to 50°F (10°C) at part-load operation, and the chilled water temperature increase in the cooling coil of the outdoor-air AHU is still 10°F, then the condition of supply air from the outdoor air AHU at part-load point *psp* can be determined as in full-load operation, $T_{psp} = 60 + 2 + 4 = 66°\text{F}$ (18.9°C), and $w_{psp} = 0.0110$ lb/lb (0.0110 kg/kg).

Draw line *psp-rp*. The condition of the mixture at part load, point *mp*, can be determined from the following relationship:

$$\frac{rp\text{-}mp}{rp\text{-}ps} = \frac{\dot{V}_p}{\dot{V}_{\text{fc}}} = \frac{200}{4 \times 300 \times 0.7} = 0.24 \tag{28.17}$$

From the psychrometric chart, $T_{mp} = 75.3°\text{F}$ (24.1°C), and $w_{mp} = 0.0116$ lb/lb (0.0116 kg/kg). Because $T_{cfp} = T_{mp} + 0.5°\text{F} = 75.3 + 0.5 = 75.8°\text{F}$, and $w_{cfp} = w_{mp} = 0.0116$ lb/lb, from the psychrometric chart, $h_{cfp} = 31.0$ Btu/lb (72.1 kJ/kg).

10. Fan-coil cooling coil load at part load Q_{cfcp} and sensible coil load at part load Q_{sfcp}, both in Btu/h (W), can be calculated as

$$Q_{cfcp} = 60 \dot{V}_{fc} \rho_{cf} (h_{cfp} - h_{fcp}) \tag{28.18}$$

$$= 60(4)(300)(0.7)(0.073)(31.0 - 27.2) = 13{,}980 \text{ Btu/h (4096 W)}$$

$$Q_{sfcp} = 60 \dot{V}_{fc} \rho_{cf} c_{pa} (T_{cfp} - T_{fcp}) \tag{28.19}$$

$$= 60(4)(300)(0.7)(0.073)(0.243)(75.8 - 63.3) = 11{,}176 \text{ Btu/h (3275 W)}$$

11. At winter heating mode design load operation, if only two fan-coil units are operated and $h_{fg,o} = 1061$ Btu/lb (2468 kJ/kg), the humidity ratio difference between zone air and the fan-coil supply air is

$$w_r - w_s = \frac{Q_{rl}}{60 \dot{V}_{fc} \rho_{hf} h_{fg,\text{o}}} \tag{28.20}$$

$$= \frac{2700}{60 \times 2 \times 300 \times 0.073 \times 1061} = 0.00097 \text{ lb/lb (0.00097 kg/kg)}$$

where w_s = humidity ratio of fan-coil supply air, lb/lb (kg/kg). As shown in Fig. 28.1*b*, $w_s = w_m = w_{fc}$, and from the psychrometric chart, the humidity ratio of outdoor air $w_o = 0.0017$ lb/lb. Therefore,

$$\frac{w_r - w_s}{w_r - w_o} = \frac{\dot{V}_p}{\dot{V}_{fc}} = \frac{200}{600} = 0.33 \tag{28.21}$$

So

$$w_r = w_o + \frac{w_r - w_s}{0.33} = 0.0017 + \frac{0.00097}{0.33}$$
$$= 0.0046 \text{ lb/lb } (0.0046 \text{ kg/kg})$$

and $w_s = w_m = w_r - 0.00097 = 0.0046 - 0.00097 = 0.0036$ lb/lb (0.0036 kg/kg).

12. Because at heating mode operation $T_r = 72°F$ (22.2°C), from the psychrometric chart, the relative humidity at winter design condition $\varphi_r = 28$ percent.

13. If air leaves the preheating coil of the outdoor air AHU at $T_{ph} = 83°F$ and $T_{pf} = T_{ph}$ + fan temperature rise = 83 + 2 = 85°F, because $w_o = w_{pf} = 0.0017$ lb/lb, from the psychrometric chart, air density at the fan supply outlet in the outdoor-air AHU $\rho_{pf} = 1/13.75 = 0.073$ lb/ft^3. The preheating coil load Q_{cph}, in Btu/h (W), can be calculated as

$$Q_{cph} = 60\dot{V}_p\rho_{pf}c_{pa}(T_{ph} - T_o) \tag{28.22}$$
$$= 60 \times 200 \times 0.073 \times 0.243(83 - 20) = 13{,}411 \text{ Btu/h } (3929 \text{ W})$$

14. If the heat loss of outdoor air supply duct is 0.5°F, $T_{ps} = 85 - 0.5 = 84.5°F$, and $w_{ps} = w_{pf} = 0.0017$ lb/lb, then point *ps* can be determined. Draw line *ps-r*. Because $w_m = 0.0036$ lb/lb (0.0036 kg/kg), point *m* can be plotted on the psychrometric chart. From the chart, $T_m = 76.2°F$ (24.5°C).

15. The temperature of the fan-coil supply air at winter design load T_s, in °F (°C), can be calculated as

$$T_s = T_r + \frac{Q_{rh}}{60\dot{V}_{fc}\rho_{cf}c_{pa}} \tag{28.23}$$
$$= 72 + \frac{9500}{60 \times 2 \times 300 \times 0.073 \times 0.243} = 72 + 14.9 = 86.9°F\,(30.5°C)$$

As $w_s = w_m = 0.0036$ lb/lb (0.0036 kg/kg), point *s* can be determined, and lines *r-s* and *m-s* can be drawn. The temperature difference $T_s - T_r = 86.9 - 72 = 14.9°F$ (8.3°C), which is smaller than the supply air temperature difference for warm air during heating mode 15°F (8.3°C) specified by ASHRAE handbooks to prevent an excessive buoyancy effect, as discussed in Sec. 18.6.

16. The air temperature at the fan outlet in the fan coil $T_{cf} = T_m + 0.5 = 76.2 + 0.5 = 76.7°F$. The heating coil load in the fan-coil unit Q_{ch}, in Btu/h (W), can be calculated as

$$Q_{ch} = 60\dot{V}_{fc}\rho_{cf}c_{pa}(T_s - T_{cf}) \tag{28.24}$$
$$= 60 \times 2 \times 300 \times 0.073 \times 0.243(86.9 - 76.7) = 7152 \text{ Btu/h } (2095 \text{ W})$$

17. At winter heating mode part-load operation, the warm supply air from the fan coil, point *sp*, moves along the horizontal line *m-sp* and is supplied at a lower temperature T_{sp} in order to maintain the required zone temperature T_{rp}.

Applications

Four-pipe fan-coil systems with a dedicated ventilation and space recirculating air system are widely used in hotels and motels where the exhaust air system for bathrooms can easily balance part of the outdoor ventilation air supply. They are also used in hospitals, schools, and offices.

28.4 TWO-PIPE FAN-COIL SYSTEMS

Two-Pipe Systems

A two-pipe system equipped with a supply pipe and a return pipe is shown in Fig. 28.4. In such a system, chilled water is supplied to the coil to cool and dehumidify the air during cooling mode operation. In heating mode operation, chilled water is changed over to hot water and then supplied to the coil to heat the air.

A two-way valve is usually installed before the coil inlet because it costs less, is easier to install, and saves pump power at part-load operation when the water flow rate is reduced. A DDC controller is often used to modulate the water flow at part load.

Nonchangeover Two-Pipe Systems

When a two-pipe fan-coil system is used to serve a perimeter zone in a building, changeover from chilled water to hot water or vice versa is a troublesome process and may take several hours. Therefore, in locations where winter weather is moderate, a nonchangeover two-pipe fan-coil system may be used.

In a nonchangeover two-pipe system, chilled water is supplied to the fan coil throughout the year when the space is occupied. Warm outdoor air is supplied to the fan coil in winter to offset the space heating load in the perimeter zone, and in spring and fall when space heating is required. In such an arrangement, various zones at different orientations in the building that need cooling and heating simultaneously during spring and fall can be served, as shown in Fig. 28.5*a*.

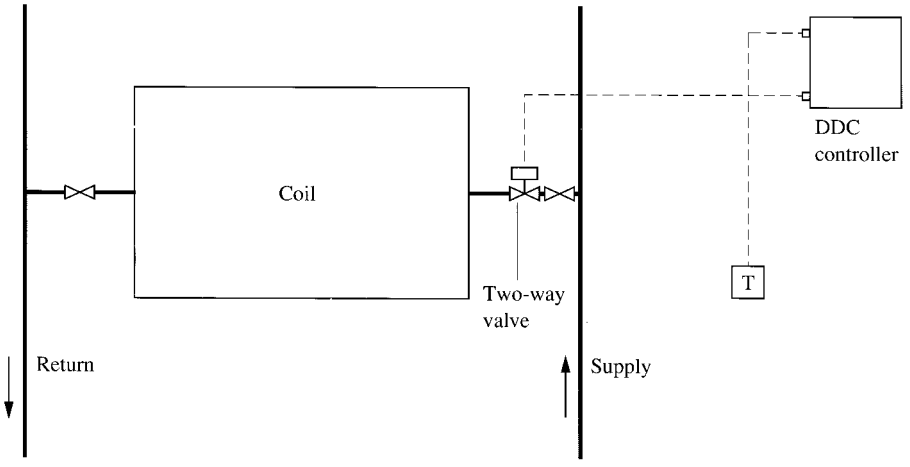


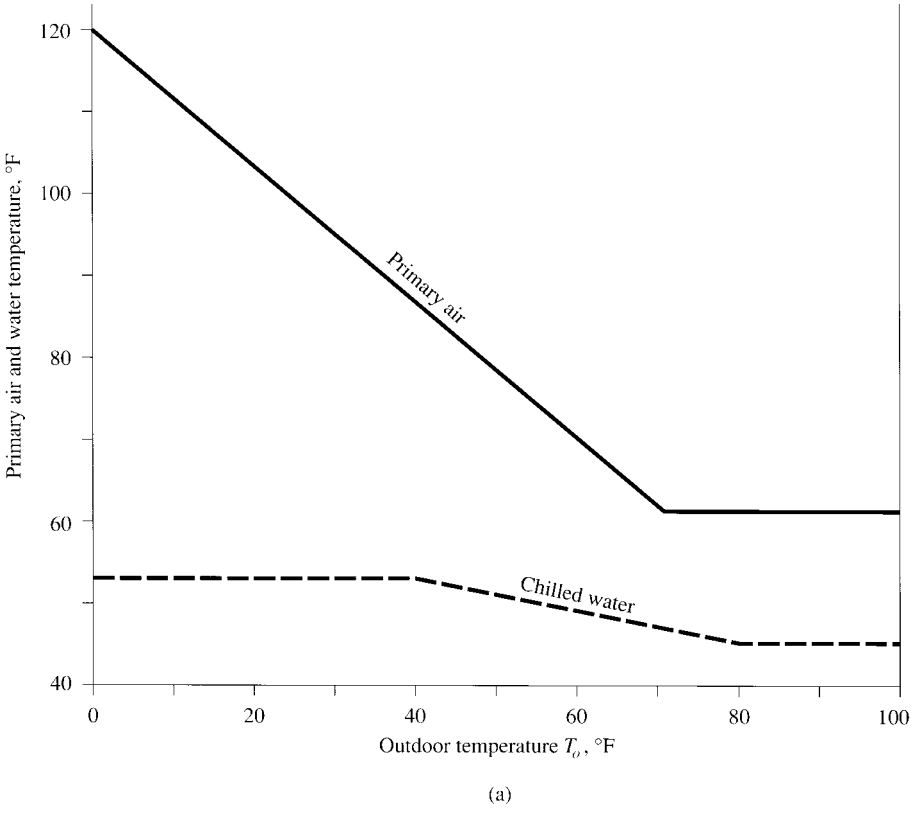
FIGURE 28.4 Supply and return mains in a two-pipe fan-coil system.

In a typical nonchangeover two-pipe fan-coil system, as shown in Fig. 28.5*a*, chilled water T_{el} enters the coil at a temperature of about 45°F (7.2°C) at summer design conditions, whereas the temperature of outdoor air supply T_{ps} may be maintained at 62°F (16.7°C). At cooling mode part-load operation, T_{el} supply for both fan-coil units and the outdoor air AHU can be reset gradually, up to 52°F (11.1°C), when the space sensible cooling load drops to 30 percent of the design value. When outdoor temperature T_o drops below 70°F (21.1°C), the outdoor air supply temperature T_{ps} begin to rise. The lower T_o, the higher T_{ps}.

If the space heating load of any room or control zone in the perimeter zone is offset entirely by the heated outdoor air in a nonchangeover two-pipe fan-coil system, the required volume flow rate of heated outdoor air $\dot{V}_p$, in cfm [$m^3/(60\ s)$], can be calculated as

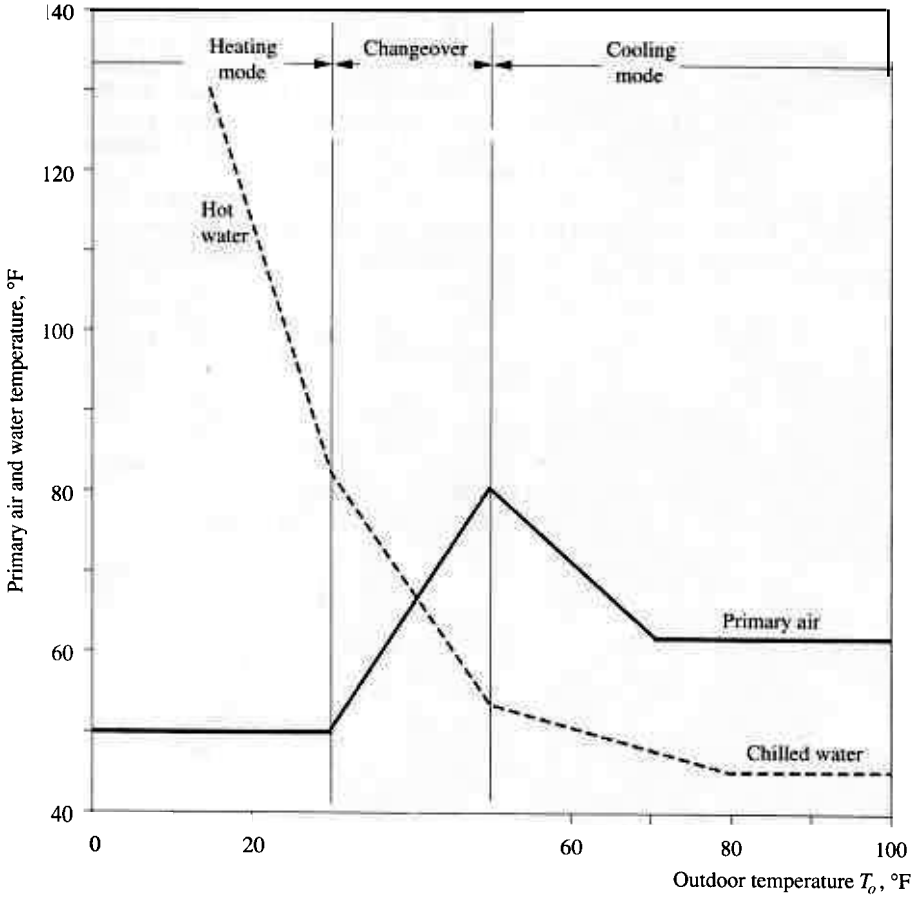
$$\dot{V}_p = \frac{Q_{rh}}{60\rho_{ps}c_{pa}\,(T_{ps} - T_r)} = \frac{(A_{ex}U_m + 60\dot{V}_{inf}\rho_o c_{pa})(T_r - T_o)}{60\rho_{ps}c_{pa}(T_{ps} - T_r)} \tag{28.25}$$

where Q_{rh} = room heating load, Btu/h (W)
A_{ex} = total area of building shell in that room, ft^2(m^2)



(a)

FIGURE 28.5 Operating parameters of air and water in a two-pipe fan-coil system: (*a*) nonchangeover; (*b*) changeover.



(b)

FIGURE 28.5 *(Continued)*

U_m = weighted average of overall heat-transfer coefficient of building shell in that room, Btu/h·ft²·°F (W/m²·°C)
$\dot{V}_{inf}$ = volume flow rate of infiltrated air, cfm [m³/(60 s)]
ρ_o, ρ_{ps} = density of outdoor air and outdoor air supply, lb/ft³ (kg/m³)
T_r = zone air temperature, °F (°C)
T_{ps} = supply temperature of conditioned outdoor air, °F (°C)

The air transmission ratio is defined as the ratio of outdoor air volume flow rate supplied to a room in the perimeter zone $\dot{V}_p$, in cfm [m³/(60 s)], to the transmission loss per degree of outdoor-indoor temperature difference ΔT_{tran}, Btu/h·°F (W/°C). According to Eq. (28.25), it can be calculated as

$$\frac{\dot{V}_p}{\Delta T_{tran}} = \frac{\dot{V}_p}{A_{ex}U_m + 60\dot{V}_{inf}\rho_o c_{pa} \dfrac{T_r - T_o}{60\rho_{ps}c_{pa}(T_{ps} - T_r)}} \tag{28.26}$$

For all the rooms in the perimeter zone supplied by the same outdoor ventilation air system in a two-pipe fan-coil system, the temperature difference between room air and outdoor air $T_r - T_o$ is a constant at any time during operation. If the temperature drop due to the heat loss of the outdoor air supply duct is ignored, for the same outdoor ventilation air system, the supply air temperature difference $T_{ps} - T_r$ is again a constant. Because air density ρ_{ps} and specific heat of supply air c_{pa} can also be considered constant, and if the zone heating load in the perimeter zone is offset entirely by the conditioned outdoor air, the air transmission ratio $\dot{V}_p/\Delta T_{\text{tran}}$ for all the control zones in the perimeter zone in the same outdoor ventilation air system for a two-pipe nonchangeover fan-coil system must be a constant.

The current trend is to have an adequate amount of outdoor air supply for a better indoor air quality. Because the conditioned outdoor air supply must be higher than the outdoor ventilation air requirement, this often means that a nonchangeover two-pipe fan-coil system has sufficient conditioned outdoor air for winter heating. In massively constructed buildings, the air transmission ratio can be smaller, such as 0.7 $\dot{V}_p/\Delta T_{\text{tran}}$ because of the heat storage effect of the building shell.

One of the primary drawbacks of a nonchangeover two-pipe fan-coil system is the waste of energy during simultaneous heating and cooling in some control zones during winter heating. In a cold winter in the northern hemisphere, for a control zone facing south, warm outdoor air heating, solar radiation, and some internal loads often combine to result in a higher zone temperature than that for a control zone facing north in the same outdoor ventilation air system. Chilled water is then admitted to the fan coil to cool the zone air simultaneously. Simultaneous heating and cooling for the same control zone must be avoided. The waste of energy of the conditioned outdoor ventilation air system in a nonchangeover two-pipe fan-coil system is similar to the waste of energy in an air skin system, discussed in Sec. 21.3.

Changeover Two-Pipe Systems

During the cooling mode operation of a changeover two-pipe fan-coil system, the space cooling load is offset by the combined cooling effect of the cooled outdoor ventilation air and chilled water supplied to the fan coil, as shown in Fig. 28.5*b*. As discussed in Sec. 7.6, ASHRAE/IESNA Standard 90.1-1999 specifies that two-pipe fan-coil changeover systems are acceptable when the design dead band width between changeover from one mode to the other is at least 15°F (8.3°C) outdoor air temperature. It also will allow operation in one mode at least four hours before changeover to the other mode. At the changeover point, reset controls allow heating and cooling supply temperatures to be no more than 30°F (16.7°C) apart.

During the fall season, before the changeover from cooling mode operation to heating mode operation, when outdoor temperature T_o is higher than changeover temperature T_{co}, the supply temperature of outdoor ventilation air increases according to T_o when T_o falls below 70°F (21.1°C). The heating effect of the conditioned outdoor ventilation air is able to offset the individual zone heating load in the perimeter zone. Chilled water is used to provide cooling and dehumidifying when needed. Meanwhile, the chilled water supplied temperature T_{el} should be reset between 45 and 52°F (7.2 and 11.1°C) at part-load operation before changeover.

During changeover, the chilled water may be reset to 52°F (11.1°C), and it is changed over to hot water at a temperature higher than 80°F (26.7°C). Meanwhile, the warm air supply is changed over to cold air supply, typically at a temperature of 50°F (10.0°C). The changeover process was discussed in Sec. 7.6, and the outdoor changeover temperature T_{co} can be calculated from Eq. (7.5). Changeover may require several hours. To prevent two to three changeovers within the same day, operation in one mode lasts at least four hours before changeover to the other mode, and a tolerance of about $\pm$ 2°F (1.1°C) is often used. There are many factors that influence the changeover temperature T_{co} in actual practice. Therfore, calculated T_{co} should be modified according to actual operating experience.

When $T_o < T_{co}$, a changeover two-pipe fan-coil system is in heating mode operation. The hot water temperature is reset according to outdoor temperature T_o. Cold outdoor air is supplied, typically at 50°F (10.0°C). Any zone cooling load of an individual control zone in the perimeter zone can be offset by the combined effect of the cold outdoor supply air and the transmission and

infiltration loss. Zone heating loads are offset by the hot water supplied to the fan coils. During heating mode operation, the refrigeration system that serves the perimeter zone is usually shut down. However, the changeover two-pipe fan-coil system still operates with a cold source of outdoor air and a heat source of hot water simultaneously.

In spring, heating mode operation changes over to cooling mode operation at outdoor temperature T_{co} only when the combined effect of outdoor ventilation air cooling and the transmission and infiltration loss does not offset the cooling load of every individual control zone in the perimeter zone.

In a two-pipe fan-coil system, except for the changeover from heating mode to cooling mode operation or vice versa, operating characteristics of the fan-coil unit and the space recirculating system, exhaust air balance, and the zone temperature control are similar to those in a four-pipe fan-coil system.

System Characteristics

System characteristics of a two-pipe fan-coil system (2PFC) are listed in Table 28.1.

Applications

Because of the waste of energy in nonchangeover two-pipe fan-coil systems and the difficulties in changeover operations, the applications of two-pipe fan-coil systems are comparatively less.

28.5 WATER-SOURCE HEAT PUMP SYSTEMS

System Description

In a water-source heat pump system, some water-source heat pumps located in the shady side of the perimeter zone of a building may extract heat from a water loop to heat the supply air, and other water-source heat pumps in the core part of the building may reject heat to the water loop to cool the supply air. Excess heat is therefore transferred from the core of the building to the perimeter zone of the building. A water-source heat pump system conserves more energy than many other air conditioning systems only when simultaneous heating and cooling occur in a building and therefore excessive heat from the core part is recovered and transferred to the perimeter zone of the building.

A typical closed-circuit water-source heat pump system is illustrated in Fig. 28.6. It consists of water-source heat pumps, an evaporative water cooler, a boiler or water heater, two water circulating pumps, an expansion tank, piping, necessary accessories, and controls. One of the water circulating pumps is the lead pump, the other is a standby. A storage tank is optional.

Operating Characteristics

During hot summer, when the outdoor wet-bulb temperature is 78°F (25.6°C), all the water-source heat pumps are operting in cooling mode. The condenser water leaves the closed-circuit evaporative water cooler at 95°F (35.0°C) and enters the water loop. After all heat is rejected from water-source heat pumps, the condenser water returns to the evaporative water cooler at a temperature of 105°F (40.6°C) and cools to 95°F (35.0°C) again.

During moderate weather, water-source heat pumps serving the shady side of the building are in heating mode, whereas those serving the sunny side of the building are in cooling mode. The rise or fall in condenser water temperature depends on the ratio of the number of cooling units to the total number of units, or, more exactly, the ratio of heat rejection to heat extraction. If heat rejected from the cooling units is greater than heat extracted by the heating units, the average temperature of the

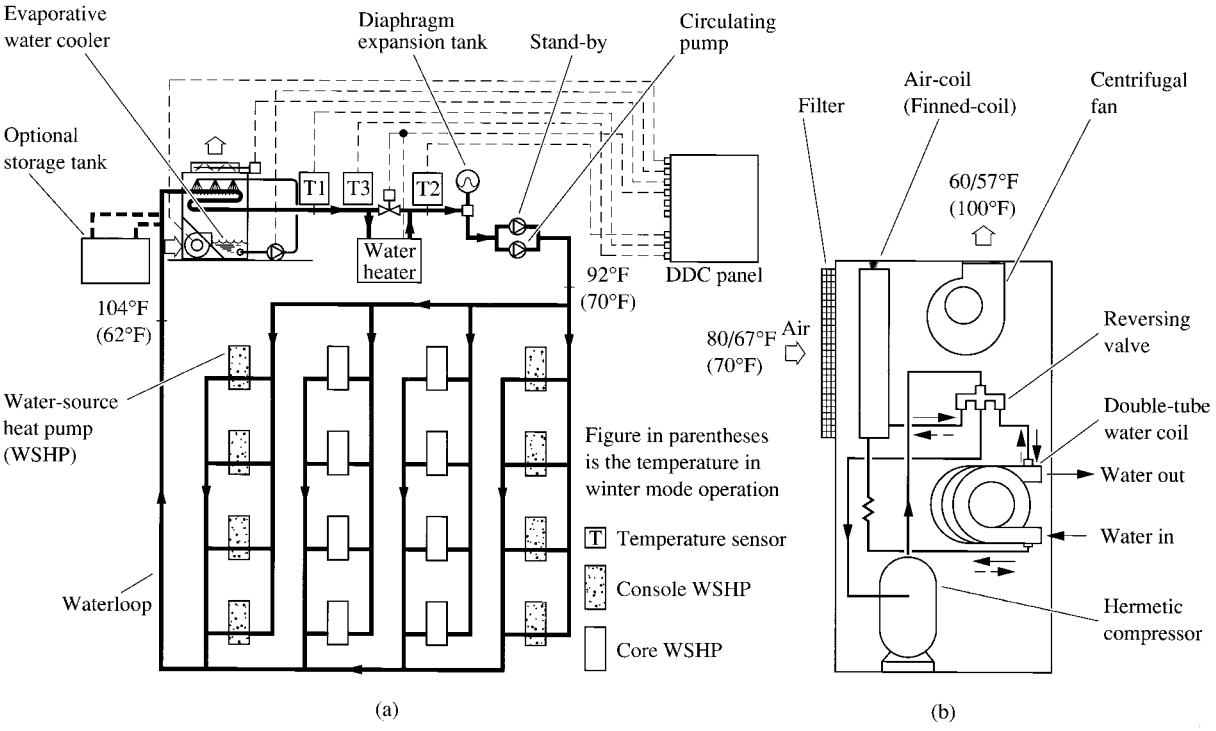


FIGURE 28.6 A typical closed-circuit water-source heat pump system: (*a*) schematic diagram; (*b*) water-source heat pump.

water loop tends to rise, and vice versa. When the condenser water temperature entering the evaporative water cooler $T_{e,ev}$ rises to 90°F (32.2°C), the cooler starts to operate and brings the condenser water temperature down to a lower value (say, 87°F, or 30.6°C).

During cold weather, some water-source heat pumps serving the core of the building still operate in cooling mode. If the heat extracted by the units operating in heating mode is greater than the heat rejected, the average temperature of the water loop drops. When the condenser water temperature $T_{e,ev}$ falls below 60°F (10.6°C), the boiler (or water heater) is energized. Heat is added to the water loop to raise $T_{e,ev}$ to 63°F (17.2°C) or higher.

A water-source heat pump system recovers the heat rejected to the water loop by the core units or by the units on the sunny side and supplies it to the units serving the perimeter zone in which heating is required to offset heat losses.

ASHRAE/IESNA Standard 90.1-1999 specifies that in a water-source heat pump system, water-source heat pumps are connected to a common water loop with central devices of heat rejection (such as a cooling tower) and heat addition (such as a boiler). Such a system shall have controls to provide heat pump water supply temperature with a dead band of at least 20°F (11.1°C) between initiation of heat rejection and heat addition by the cooling tower and boiler (for example, 90 − 60 = 30°F or 16.7°C).

Loop Temperatures

Water-source heat pumps (WSHPs) are rated according to Air-Conditioning and Refrigeration Institute (ARI) Standard 320 at the following entering condenser water temperatures: for cooling mode

performance 85°F (29°C) and for heating mode performance 70°F (21°C). Pietch (1991) listed the performance of WSHPs of a WSHP system at various loop temperatures:

Loop temperature, °F (°C)	Rating mode	Relative cooling capacity	EER/EER$_{c,ra}$	Heating capacity/ARI$_{rated\ c\ cap}$	COP/COP$_{h,ra}$
90 (32.2)		0.98	0.94	1.50	1.08
85 (29.4)	Cooling	1.00	1.00	1.45	1.06
80 (26.7)		1.02	1.06	1.40	1.04
75 (23.9)		1.04	1.12	1.35	1.02
70 (21.1)	Heating	1.06	1.18	1.30	1.00
65 (18.3)		1.08	1.24	1.25	0.98
60 (15.6)		1.10	1.30	1.20	0.96

Here EER$_{c,ra}$ indicates the energy efficiency ratio of the WSHP at the cooling mode rated conditions, ARI$_{rated\ c\ cap}$ indicates the ARI-rated cooling capacity, and COP$_{h,ra}$ indicates the COP of the WSHP at heating mode rated conditions.

WSHPs at cooling mode operation require lower condenser water temperature (CWT) to reduce energy use, and WSHPs at heating mode need higher CWT for a higher COP. Optimizing loop temperatures depend on the condition of zone loads, the number of WSHPs in cooling and heating mode operations, and the outdoor weather. Because a water loop may have WSHPs in both cooling and heating mode operations, the water loop temperature operating limit is usually maintained between 60 and 90°F (15.6 and 32.2°C).

Gottfried et al. (1997) suggested when the CWT has been lowered to a rather lower value in a late afternoon, this allows the CWT to rise to a level that maintains WSHP efficiencies and at the same time keep evaporative cooler (cooling tower) energy use to a minimum during the last working hours of the day.

Water-Source Heat Pumps

Three types of WSHPs are widely used in WSHP systems:

- Console type for perimeter zones. Most have a cooling capacity of 1 ton (12,000 Btu/h, or 3.5 kW) or less.
- Vertical WSHPs often installed in mechanical rooms. They are used to serve the core (interior zone) of the buildings and are connected with ducts. Vertical units often have cooling capacity from 10 up to 24 tons.
- Ceiling-mounted horizontal WSHPs, often for special rooms. Ceiling units have a cooling capacity, usually between 0.5 and 5 tons.

Most WSHPs have a heating capacity about 10 percent higher than their cooling capacity.

A water-source heat pump usually consists of an air coil that is a finned coil to condition the air; a double-tube water coil to reject or extract heat from the water loop; a forward-curved centrifugal fan, which is often located downstream of the air coil; single or twin hermetic compressors; a short capillary tube; a reversing valve; an outer casing; controls; and accessories. A two-speed fan motor is often used for better capacity control. A typical water-source heat pump is shown in Fig. 28.6*b*. In a vertical water-source heat pump, the centrifugal fan is usually located at the top outlet, and the hermetic compressor is often mounted in the ottom of the unit. In a horizontal unit, the centrifugal fan is usually located at the end of the unit.

During cooling mode operation, the air coil acts as an evaporator and the water coil as a condenser. Air in a typical water-source heat pump is cooled and dehumidified at the air coil from an entering dry-bulb temperature of 80°F (26.7°C) and wet-bulb temperature of 67°F (19.4°C) to a

leaving condition of 60°F (15.6°C) dry-bulb and 57°F (13.9°C) wet-bulb temperatures. The suction temperature is about 43°F (6.1°C). On the other hand, in heating mode operation, the air coil acts as a condenser and the water coil as an evaporator. During cold weather air is heated at the air coil from 70°F (21.1°C) to an off-coil temperature of 100°F (37.8°C). The suction tmperature in winter is about 45°F (7.2°C). Sometimes, electric resistance heating, in steps, is added between the air coil and the centrifugal fan instead of using a centralized water heater.

Energy Performance and Energy Use Intensity of WSHPs

ASHRAE Standard 90.1-1999 mandatorily specifies the minimum efficiency requirements for water-source heat pumps as follows:

Operation	Capacity, Btu/h	Rating conditions	Minimum efficiency	Efficiency as of 10/29/2001
Cooling	$Q_{rc} < 65{,}000$	85°F entering water	9.3 EER	
		86°F entering water		11.2 EER
	$17{,}000 \le Q_{rc} < 65{,}000$	85°F entering water	9.3 EER	
		86°F entering water		12.0 EER
	$65{,}000 \le Q_{rc} < 135{,}000$	85°F entering water	10.5 EER	
		86°F entering water		12.0 EER
Heating	$Q_h < 135{,}000$ (heating capacity)	70°F entering water	3.8 COP	
		68°F entering water		4.2 COP

Gottfried et al. (1997) reported a new high-efficiency WSHP with an EER of 14.9 (4.33 COP with a cooling capacity from 21,000 ro 57,000 Btu/h, or 6.2 to 16.7 kW) and a high-efficiency motor with a minimum efficiency of 90 percent.

According to DOE/EIA Commercial Buildings Consumption and Expenditures 1995, the energy use intensity (EUI) for heat pumps (including mainly water-source heat pumps and air-source heat pumps) in commercial buildings in 1995 in the United States was 28.1 kBtu/ft$^2 \cdot$yr (8.2 kWh/ft$^2 \cdot$yr or 88 kWh/m$^2 \cdot$yr) which is the smallest of all the air conditioning systems.

Closed-Circuit Evaporative Water Cooler

A closed-circuit evaporative water cooler (CCEWC), sometimes called a closed-circuit cooling tower, resembles a cooling tower. However, condenser water in a cooling tower is an open-circuit system, whereas in an evaporative water cooler the condenser water is a closed-circuit system (see Fig. 28.6*a*). Condenser water flows through a coil onto which recirculating water is sprayed. Heat is rejected from the condenser water through the tube wall and is absorbed by the vaporized liquid at the outer surface of the coil. A closed-circuit evaporative water cooler is usually located outdoors, most probably on the rooftop.

Both centrifugal fans and propeller fans can be used in a closed-circuit evaporative water cooler. The capacity of a centrifugal fan is more easily controlled with a damper and makes less noise; therefore, it is more frequently used.

The capacity of a closed-circuit evaporative water cooler can be modulated by the following methods:

- Convective cooling — opening of the top damper only
- Natural-draft evaporative cooling — spraying of recirculating water and opening of the top damper
- Forced-draft evaporative cooling — operating the fan and the water spraying
- Forced-draft evaporative cooling with damper modulation

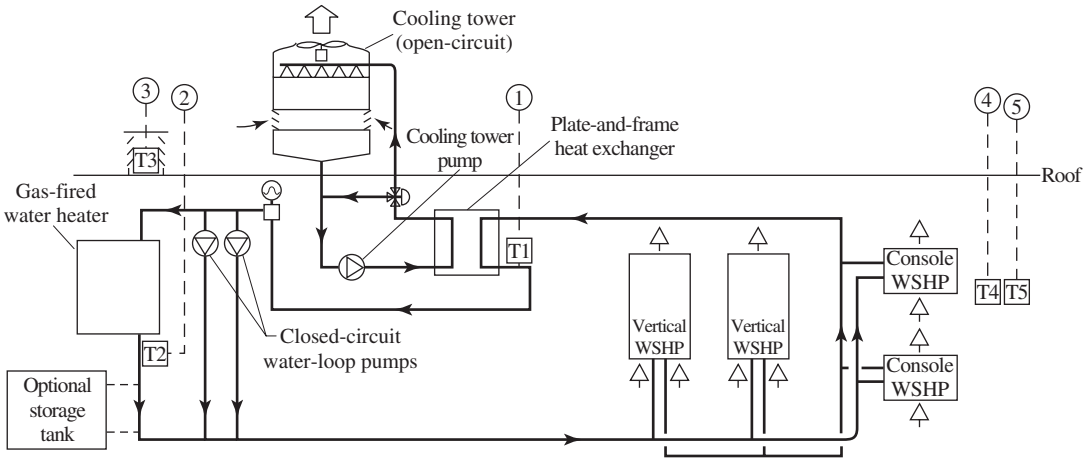


FIGURE 28.7 A water-source heat pump system using an open-circuit cooling tower.

The motive for a closed-loop water system is to prevent fouling of the water coils in water-source heat pumps. Closed-circuit evaporative water coolers were widely used in the early years of the application of WSHP systems. However, CCEWCs have the following disadvantages:

1. According to Cooper (1994), the maintenance of spray nozzles, pumps, and heat exchangers of a CCEWC required twice amount of service of an open-circuit cooling tower.
2. According to Kush (1990), field experience in an installation in Stamford, Connecticut, showed that the heat and pumping energy losses from the outdoor evaporative water cooler and piping work were about one-half of the energy input to the electric boiler in January 1988.
3. For locations where the outdoor temperature drops below 32°F (0°C), freeze protection of the outdoor portion of the water loop must be considered.

The current trend is as follows:

- Divide the water loop of a WSHP system into two portions: closed-circuit and open-circuit. A plate-and-frame heat exchanger is used to connect these two part and transfers heat energy from the closed circuit to open circuit, as shown in Fig. 28.7.
- Use a cooling tower (open-circuit) instead of a closed-circuit evaporative water cooler, and two sets of water circulating pumps, one for the closed-circuit and the other for the open-circuit cooling tower and accessories. Set only the cooling towers and the necessary pipework and accessories in the open circuit outdoors.
- Properly insulate the outdoor pipework and equipment. Provide freeze protection for the outdoor open-circuit water loop, including the addition of a certain percentage of ethylene glycol solution to the water loop. Ethylene glycol is expensive. Inhibitors must be added at the same time to resist corrosion. Ethylene glycol solution reduces the heat-transfer coefficient as well as the heating and cooling capacities of the equipment.

ASHRAE/IESNA Standard 90.1-1999 specifies that for climates with HDD65 greater than 1800 hours annually, if a closed-circuit tower is used, then either an automatic valve shall be installed to bypass all but a minimal volume flow of water around the tower for freezing protection or low-leakage positive closure air dampers shall be provided. If an open-circuit tower is used directly in the

heat pump water loop, then an automatic valve shall be installed to bypass all heat pump water flow around the tower. If an open-circuit tower is used in conjunction with a separate heat exchanger to isolate the tower from the heat pump water loop, then the heat loss shall be controlled by shutting down the circulating pump on the outdoor cooling tower loop during cold winter when the tower is not in use.

Water Heater

A water heater or a hot water boiler is used to provide heat energy to the water loop to maintain a preset operating temperature. Electric boiler is most widely used. It is easiest to install and maintain. In locations where electric rates are high, electric boilers require higher operating costs than gas- or oil-fired boilers.

For better capacity control, electric load in an electric boiler is often divided into stages, as discussed in Sec. 8.4. The heating capacity is controlled in steps by sensing the water temperature entering the electric boiler.

Gas- or oil-fired heaters are combustion devices containing combustion chambers. Water at a temperature of 60°F (21.1°C) from the water loop cannot directly contact one side of the combustion chamber and a flame on the opposite side. It will cause condensation and deteriorate the water heater. Water contacting one side of the combustion chamber must be recirculated at a temperature of 140°F (60.0°C) or above. Loop water can then be heated by mixing.

Storage Tanks

Whether a storage tank should be used in a WSHP system depends on the local electric rate structures and a careful analysis of initial costs and operating cost. The principle of thermal storage is covered in Chap. 31. Because of the electric deregulation as well as the application of real-time pricing (RTP) rate structures, the provision of a storage tank in a WSHP system is often cost-effective in many applications. Heat rejected to the water loop at a temperature between 60°F (15.6°C) and 90°F (32.2°C) can be stored in the storage tank to offset nighttime heat losses if the water loop operates at night with a setback space temperature.

Many utilities offer lower electric rates at off-peak times. A cooling tower or evaporative cooler can be used to cool the water to a lower temperature. It can then be stored in the tank for use during on-peak hours in summer to reduce costs. An electric boiler also may be energized at night to raise the water temperature to 180°F (82.2°C) or higher. High-temperature stored water can be used in winter peak hours to reduce electricity costs.

Air Systems and Maintenance

A WSHP system must have a dedicated outdoor ventilation system to provide ventilation air for occupants. Outdoor ventilation air can either be supply to the return plenum mixing with the recirculating air or supply to the conditioned space directly. The centrifugal supply fan in a WSHP itself forms a space recirculating system. Console WSHPs have no ducts. The operating characteristics of the air systems in a WSHP system are similar to those in a four-pipe fan-coil system.

According to Cooper (1994), maintenance in a WSHP system consists of mainly cleaning and changing filters and replacing belts and seal bearings. Sometimes a WSHP is removed and the condenser is acid-cleaned. There is also maintenance required for cooling towers, pumps, and water heaters. For the 3 million ft^2 (278,800 m^2) of commercial office buildings that have been maintained, records showed that on WSHPs wih a cooling capacity of 1 ton (3.5 kW) and less, the replacement percentage is less than 0.5 percent per year. For WSHPs of 2 to 5 tons (7 to 17.6 kW), the rate is slightly higher, for WSHPs of 6 to 10 tons (21.1 to 35.0 kW), slightly higher still.

Controls

Water-Loop Temperature Control. In a WSHP system with an open-circuit cooling tower and a plate-and-frame heat-exchanger, the water temperature leaving the plate-and-frame heat exchanger is controlled between 60 and 90°F (15.6 and 32.2°C) when the WSHP system is in operation. For a typical WSHP system, a temperature sensor T1 may be located at the exit of the plate-and-frame heat exchanger, as shown in Fig. 28.7. When the microprocessor-based DDC controller receives a signal from T1, it actuates the following:

At 57°F (13.9°C)	Alert by alarm. WSHP, cooling tower, water heater, and pumps all shut down. Start the water-loop water pumps.
At 60 to 68°F (15.6 to 20.0°C)	Energize the water heater.
At 84°F (28.9°C)	Start the cooling tower water circulating pumps.
At 86°F (30.0°C)	Open the damper of the cooling tower.
At 88°F (31.1°C)	Turn on the first-stage tower fans.
At 90°F (32.2°C)	Turn on the second-stage tower fans.
At 105°F	Alert by alarm. System will shut down.

An outdoor air sensor T3 resets the water heater energizing temperature T2 in such a manner that when the outdoor temperature T_o drops from 60 to 0°F (15.6 to −17.8°C), T2 increases linearly from 60 to 68°F (15.6 to 20.0°C).

To maintain space air at 75°F (23.9°C), 50 percent relative humidity, and a dew point of 55°F (12.8°C) in summer, it is critical to maintain a minimum water-loop temperature $T_{wl} = 57$°F (13.9°C). If $T_{wl} < 57$°F (13.9°C), condensate may form on the outer surface of uninsulated pipes in the conditioned space and cause damage.

The actuating temperatures for turning off the second-stage and first-stage tower fans, shutting off the cooling tower water pump, and closing the damper in the cooling tower as the water temperature decreases should be successively 2°F (1.1°C) lower to prevent short cycling [e.g., turn off the second-stage tower fans at 90°F (32.2°C), turn off the first-stage tower fans at 88°F (31.1°C)].

Capacity Control of Water Heater. If the water heater is a gas-fired hot water boiler or a hot water heat exchanger, a temperature sensor T2 senses the water temperature leaving the water heater. The heating capacity can be controlled by modulating the flow rate of gas or hot water by means of a DDC controller. Because most hot water boilers contain a limited amount of water, if the temperature sensor is located at the exit side of the boiler, the temperature response to modulation is fast. Therefore, a broadband DDC controller should be used to prevent short cycling.

If an electric boiler is used, step control in appropriate stages can be performed through a sensor located at the inlet to sense the temperature of entering water. Step control often causes temperature fluctuation at the leaving side.

Safety Control. In addition to the high and low operating limit control of the water-loop temperature, a flow switch or pressure differential sensor should be installed across the water-loop pump. In case of pump failure, the entire system, including the water-source heat pumps, cooling tower, and circulating pump, or water heater shuts down. After a delay of 10 to 15 s, the standby pump is energized. At the same time, an alarm sounds with an indicating light. When water flow is resumed, the cooling tower and circulating pumps, or the water heater and water-source heat pumps, are restarted, with a delay between each stage to limit any sudden increase in starting current.

Water-Source Heat Pump Control. When the pushbutton is engaged, the electric circuit is energized and the outdoor air damper is opened. When the zone temperature rises above the cooling mode set point, the temperature sensor T4 (Fig. 28.7) calls for cooling. The DDC controller then

starts the fan. For small console WSHPs used in the perimeter zone, a multispeed fan is usually used. Fan is at either the high or low position, which is set manually. After a 30- to 50-s delay, if the high-pressure and low-pressure safety control circuits are closed, the compressor is started. For large vertical or horizontal WSHPs, the fan is belt-driven. Fan motor starts the fan after safety control circuits are closed. If there are multiple compressors, the leading compressor is started. WSHP is now in cooling mode operation. When the zone temperature drops below the cooling mode set point and is sensed by T4, the DDC controller then stops the compressor, or one of the compressors is deenergized. The fan should be operated continuously when the space is occupied, to provide outdoor ventilation air to the space.

When the zone temperature drops below the heating mode set point and is sensed by the temperature sensor T5, the DDC controller starts the compressor. WSHP is operating at heating mode. When the zone temperature rises above the heating mode set point, the DDC controller stops the compressor. There is always a deadband of 1 to 3°F (0.6 to 1.7°C) if the cooling mode is automatically changed to heating mode or vice versa.

In either heating or cooling mode operation, if the discharge pressure rises above the high-pressure limit, or the suction pressure drops below the low-pressure limit, the high- or low-pressure control opens the electric circuit, lights an alarm lamp, and stops the compressor.

ASHRAE/IESNA Standard 90.1-1999 specifies that each water-source heat pump shall have a two-position valve for a water system having a total pump system power exceeding 10 hp (7.5 kW).

System Characteristics

System characteristics of water-source heat pump systems are listed in Table 28.1.

Case Studies: Water-Source Heat Pump Systems

Gottfried et al. (1997) introduced a water-source heat pump system with an open-circuit cooling tower in Ridgehaven Building, San Diego (RBSD), California, as discussed in Sec. 25.3. RBSD is an office building of 73,000 ft^2 (6784 m^2). In 1995, the new WSHP system installed in each of the two wings of the Ridgehaven Building included the following primary components:

- 72 high efficiency WSHPs having an EER of 14.9 (4.33 COP)
- 100-ton (open-circuit) cooling tower
- 400,000 Btu/h (117 kW) gas-fired hot water boiler
- Condenser water isolation valve for each WSHP to cut off condenser water flow when the WSHP is shut off

Ridgehaven Building provides 20 cfm (10 L/s) outdoor ventilation air for each occupant. Using specified high-efficiency filters for outdoor-air AHU, the RBSD target energy use is 9 $kWh/ft^2 \cdot yr$ (97 $kWh/m^2 \cdot yr$). The first month's operating data showed that the annual energy use for a WSHP system with an open-circuit cooling tower prior to commissioning is approximately 11 $kWh/ft^2 \cdot yr$ (118 $kWh/m^2 \cdot yr$).

For an office building in Stamford, Connecticut, Kush (1990) analyzed and reported on the actual performance of a closed-circuit water-source heat pump system that served a total area of 72,000 ft^2 (6691 m^2) of office space on three floors. This system has 140 one-ton perim (console) WSHPs, twelve 10-ton core WSHPs, a closed-circuit evaporaive cooler, and a 300-kW electric boiler. Stamford has a 97.5 percent winter design temperature of 9°F (−12.8°C) and 5617 heating degree-days. In summer, 2.5 percent design dry-bulb temperature is 84°F (28.9°C) with a mean coincident wet-bulb temperature of 71°F (21.7°C).

According to the results measured between November 1987 and October 1988, the breakdown of annual HVAC&R energy use was as follows:

Electric boiler	32 percent
Evaporative water cooler	4 percent
Water-loop pumps	11 percent
Perimeter WSHPs	18 percent
Core WHPs	
Fan	8 percent
Cooling	15 percent
Heating	2 percent
Others	9 percent

The energy use for heating in core WSHPs is mainly caused by excessive outdoor ventilation air.

The monthly HVAC&R energy use of this building between November 1987 and October 1988 is shown in Fig. 28.8. The highest HVAC&R energy use, 110,050 kWh, occurred in January 1988. Of this, 66 percent was consumed by the electric boiler. The highest in summer months was that for August 1988—65,715 kWh. About one-half of August's energy was used by core WSHPs. Year-round, the lowest energy use, 32,330 kWh, occurred in May 1988, when the core WSHPs consumed 40 percent of the energy input.

Design Considerations

- Cooper (1994) recommended the following rules for occupant satisfaction:

  Provide core WSHPs from mechanical rooms.

  Provide console WSHP with individual control for control zones in perimeter zone.

  Provide adequate outdoor ventilation air.

  Give the tenant means to use the HVAC system in off hours.

  Decentralize to the points free of major shutdowns.

  Adjust tenant space conveniently.

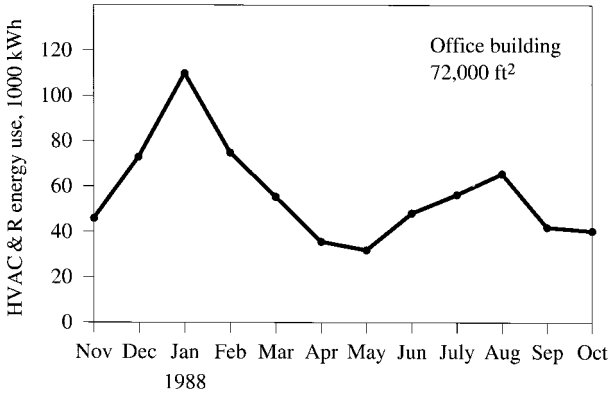


FIGURE 28.8 HVAC&R monthly energy use of a water-source heat pump system.

- If ceiling units are used, good design of ducting, return plenum, vibration hangers, and piping flex is mandatory and requires careful consideration for noise attenuation. When console WSHPs are used, they should not oversized. An oversized WSHP creates more noise and is not energy-efficient.
- A water flow rate of 2.5 to 3.5 gpm (0.16 to 0.22 L/s), typically 3 gpm (0.19 L/s) per cooling ton, is appropriate for water-source heat pump systems.
- Outdoor ventilation air rates should be adequate and should not be greater than that specified in ASHRAE Standard 62-1999, as listed in Table 5.9. Outdoor air damper should be closed when the conditioned space is not occupied.
- Fan power in water-source heat pumps has a certain influence on energy use. Both supply volume flow rate and total pressure loss should be carefully calculated in order to select the proper unit.
- Variable-speed water-loop pumps and cooling tower fans are often cost-effective in large WSHP systems.

28.6 PANEL HEATING AND COOLING

Panel heating and cooling involves the control of the mean radiant temperature of the conditioned space, which is closely related to the thermal comfort of the occupants. Floor-panel heating systems have been used by many residences in Europe since the 1980s, as discussed in Sec. 8.8. However, panel heating and cooling is expensive and has a very limited application in the United States. If the conditioned space has a large glaze area, metal ceiling panel should be used. A dedicated ventilation system should be used to provide the ASHRAE Standard 62-1999 specified amount of outdoor ventilation air. To prevent any possibility of condensation on the panels, the air system must maintain a space dew point temperature and humidity levels lower than the design conditions all the time when cooling panel is used.

Panel heating and cooling requires greater field experience in design and operation. Refer to *ASHRAE Handbook 1996*, *HVAC Systems and Equipment*, chapter 6 for detailed information.

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CHAPTER 29

AIR CONDITIONING SYSTEMS: PACKAGED SYSTEMS AND DESICCANT-BASED SYSTEMS

29.1 PACKAGED SYSTEMS 29.2
- Comparison between Packaged and Central Systems 29.2
- Applications 29.3
- Types of Packaged Systems 29.4

29.2 SINGLE-ZONE CONSTANT-VOLUME PACKAGED SYSTEMS 29.4
- System Description 29.4
- Air and Water Economizers 29.4
- Supply Volume Flow Rate, and Cooling and Heating Coil Loads 29.4
- Controls 29.5
- Energy Use Intensities 29.5
- System Characteristics 29.5

29.3 SINGLE-ZONE VAV PACKAGED SYSTEMS 29.7
- System Description 29.7
- System Calculations 29.7
- Controls 29.7
- System Characteristics 29.8

29.4 COMPLIANCE WITH STANDARD 90.1–1999 THROUGH SIMPLIFIED APPROACH OPTION FOR SMALL AND MEDIUM HVAC&R SYSTEMS 29.8

29.5 VAV COOLING PACKAGED SYSTEMS 29.9
- System Description 29.9
- Supply Volume Flow Rate and Coil Load 29.10
- Pressure Characteristics and Duct Static Pressure Control 29.10
- System Characteristics 29.12

29.6 VAV REHEAT PACKAGED SYSTEMS 29.12
- System Description 29.12
- Supply Volume Flow Rate and Coil Load 29.12
- Night Setback and Morning Warm-up 29.14
- Evenly Distributed Airflow at DX Coils 29.14
- Discharge Air Temperature Control for Packaged Systems 29.15
- Fan Modulation 29.16
- Air-Cooled, Water-Cooled, and Evaporatively-Cooled Condensers 29.17
- Controls and Sound Problems for Rooftop Packaged Units 29.17
- Case Study: VAV Reheat Packaged System for Precision Manufacturing 29.17
- System Characteristics 29.18

29.7 PERIMETER-HEATING VAV PACKAGED SYSTEMS 29.18
- System Characteristics 29.18

29.8 FAN-POWERED VAV PACKAGED SYSTEMS 29.18
- System Description 29.18
- Supply Volume Flow Rate and Coil Load 29.19
- Controls 29.20
- Case Study: A Fan-Powered VAV Packaged System with Rooftop Packaged Unit 29.20
- System Characteristics 29.22

29.9 DESICCANT-BASED AIR CONDITIONING SYSTEMS 29.22
- Desiccant-Based Air Conditioning 29.22
- Desiccant Dehumidification and Sensible Cooling 29.22
- Desiccant-Based Air Conditioning Systems 29.24
- Desiccants 29.24
- Rotary Desiccant Dehumidifiers 29.27

29.10 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR A SUPERMARKET 29.27
- Loads in Supermarkets 29.27
- System Description 29.28
- Space Conditioning Line 29.28
- Operating Parameters in Rotary Desiccant Dehumidifier 29.29
- Heat-Pipe Heat Exchanger 29.29
- Mixing of Process Air and Recirculating Air 29.30
- Indirect Evoporative Cooler or Refrigeration 29.30

Gas Heater 29.30
Operating Parameters of the Desiccant-Based Air Conditioning Cycle 29.30
Part-Load Operation and Controls 29.31
29.11 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR RETAIL STORES 29.31
System Description 29.31
Operating Characteristics 29.31
Performance 29.32
29.12 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR OPERATING ROOMS 29.32
Indoor Environment of Operating Rooms 29.32
System Description 29.33
29.13 APPLICATIONS OF DESICCANT-BASED AIR CONDITIONING SYSTEMS 29.34
Comparison between Conventional Vapor Compression Refrigeration System and Desiccant-Based Air Conditioning System 29.34
Conditions to Apply Desiccant-Based Air Conditioning Systems 29.34
System Characteristics 29.35
REFERENCES 29.35

29.1 PACKAGED SYSTEMS

Comparison between Packaged and Central Systems

The differences in construction and operational characteristics between packaged and central systems include these:

- A packaged system (PS) always uses a DX coil to cool the supply air directly in the PU, whereas a central system (CS) adopts chilled water as the cooling medium. Refrigerants cool the chilled water in the chiller, and the chilled water cools the supply air in the AHU. If both packaged and central systems are using water-cooled condensers, a central system will need a 3 to 7°F (1.7 to 3.9°C) lower evaporating temperature than a packaged system.
- In a packaged system, the size of the air system, refrigeration system, and air conditioning system that serves a specific area, a typical floor, an area of several floors, or the entire building is the same. In a central system, the size of the conditioned area of an air and refrigeration system is usually different.
- In a packaged system gas-fired furnace, electric heaters are often used to heat the air, and DX coils, often air-cooled, or sometimes evaporative condensers are used. Many packaged units are rooftop units and installed outdoors. Scroll and reciprocating compressors are usually used in packaged systems. In a central system, there is often a central plant where boilers and chillers are installed indoors to supply hot and chilled water to the AHUs and to cool and heat the supply air there. Centrifugal, screw, and reciprocating chillers and water-cooled condensers are used in central systems. AHUs are usually installed indoors in the fan rooms.
- In a packaged system, the controls of the heating and cooling system are often a part of the discharge air temperature control in the packaged unit. In a central system, there are separate water system controls, heating system controls, and refrigeration system controls in the central plant.
- In general, low- , medium- , and high-efficiency filters are used in packaged systems; and usually, medium- and high-efficiency filters are used in central systems.
- Packaged units used in packaged systems are factory-fabricated and -assembled, whereas some components in an AHU may be custom-built in the field.
- The modulation of cooling capacity in a packaged system with scroll and reciprocating compressors is achieved by cycling of the cylinders or step controls. The control actions of the modulation

of capacities in air, water, and refrigeration systems of a central system are often stepless, continuous modulation controls.

Because of these differences, packaged systems and central systems are compared as follows: First, consider the IAQ of the conditioned space that the air conditioning systems serve:

- Both the PS and the CS can provide the minimum ventilation air to dilute the air contaminants.
- Both the PS and CS can provide the humidity control and prevent wet surfaces and mold growth in the ducts and conditioned space.
- Because of the lower fan total pressure provided by the small and medium-size packaged units, low- and medium-efficiency filters are often used in small and medium-size PUs instead of medium- and high-efficiency filters in AHUs

Second, the energy efficiency of the centrifugal and screw chillers is higher than that of reciprocating and scroll compressors in packaged units, and the combined fan, motor, and drive efficiency is higher in AHUs in CS than in PUs in PS. However, a higher fan total pressure in AHUs, greater system heat gains, and an additional cooling medium require a greater evaporating-condensing pressure lift than a direct-expansion (DX) system if PS and CS both are using either water-cooled or evaporatively cooled condensers. All these cause the energy consumption of packaged systems to be generally less than that of the central systems.

According to the DOE/EIA 1995 Commercial Buildings Consumption and Expenditures, the annual energy use intensities of the PS and CS in 1995 in the United States are as shown below:

	EUI, kBtu/ft^2·yr (kWh/m^2·yr)
Packaged systems	
Water source heat pumps (HPs), air source HPs, and other HPs	28.1 (8.2)
VAV systems	37.1 (10.9)
Single-zone, constant-volume	44.0 (12.9)
Central systems	44.7 (13.1)

Third, packaged systems need more maintenance than central systems.

Fourth, rooftop packaged systems may be located directly above the occupied area and create potential noise problems. Proper design and attenuation are important.

Fifth, chillers and chilled water systems in a central system have greater reliability than more scattered packaged units in packaged systems. The flexibility of expansion and change of layout are similar for both packaged and central systems.

Sixth, most of the packaged systems and the central systems need return air to mix with the outdoor ventilation air, and install mechanical exhaust systems if required. Central systems are more easily coordinated with smoke-control systems in multistory buildings than packaged systems.

Finally a packaged system has lower initial cost and often requires less space; factory-built units are easier and faster to install than central systems.

Applications

Most current medium-size and large packaged units are equipped with microprocessor-based DDC control systems, higher-efficiency scroll compressors, medium- and sometimes high-efficiency filters, with many energy-efficient features, such as VAV boxes, air economizer, evaporative condenser, fault detection and diagnostics. All this makes the packaged system the most widely used system in small and medium-size air conditioning (HVAC&R) systems in both commercial buildings and households

in the United States, as discussed in Sec. 1.5. In commercial buildings in 1995, the packaged system had 48 percent of the floor area among the air conditioned buildings, and in households, 55 percent.

The packaged unit is the primary equipment in a packaged system, as discussed in Sec. 16.5. In a packaged system, air is often directly heated in the gas furnace or electric heater and is cooled by DX coils. These are the primary differences between an AHU and a PU, and between a central and a packaged system. On the other hand, because of the use of the microprocessor-based DDC system in the packaged system, today the difference in indoor environmental control between a custom-built central system and a packaged system is less than ever before.

Types of Packaged Systems

Packaged systems can be subdivided according to their configuration and operating characteristics into the following air conditioning systems:

- Single-zone constant-volume packaged system (SZCVPS)
- Single-zone VAV packaged system (SZVAVPS)
- VAV cooling packaged system (VAVCPS)
- VAV reheat packaged system (VAVRPS)
- Perimeter-heating VAV packaged system (PHVAVPS)
- Fan-powered VAV packaged system (FPVAVPS)

29.2 SINGLE-ZONE CONSTANT-VOLUME PACKAGED SYSTEMS

System Description

A single-zone, constant-volume packaged system is an air conditioning system that uses a packaged unit to supply and return a constant-volume flow rate of conditioned air to and from a single-zone conditioned space. It maintains a predetermined zone parameter at design and part-load conditions by controlling the capacity of the gas furnace or refrigeration compressors. Single-zone packaged systems are widely used in residences, indoor stadiums, arenas, and many industrial applications.

A typical single-zone constant-volume packaged system is simlar to that shown in Fig. 20.16*a*. It consists of mainly an upflow gas furnace and a DX refrigeration system with the following:

- A supply fan, most often of forward centrifugal type. Centrifugal fans with airfoil blades are usually used in large packaged units.
- A gas furnace with induced combustion, primary and secondary heat exchangers. An air-source heat pump or an electric heater may be used to provide winter heating instead of a gas furnace.
- Low- and medium-efficiency filters.
- DX coil connected with an outdoor condensing unit.
- Supply and return ducts, diffusers, and return inlets.
- Space, functional, and safety controls.
- A heating element humidifier and an exhaust/relief/return fan (both are optional).

Supply Volume Flow Rate, and Cooling and Heating Coil Loads

For a single-zone, constant-volume packaged system, the supply volume flow rate $\dot{V}_s$, in cfm [$m^3/(60\ s)$], required to offset the space sensible cooling load at summer design conditions can be

calculated from Eq. (20.69), as discussed in Sec. 20.17,

$$\dot{V}_s = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_s)} \tag{29.1}$$

The supply volume flow rate required to offset the space heating load at winter design conditions can be calculated from Eq. (20.70)

$$\dot{V}_s = \frac{Q_{rh}}{60\rho_s c_{pa}(T_s - T_r)} \tag{29.2}$$

Usually, the supply volume flow rate required to offset the summer design sensible cooling load is greater than the winter heating load. For convenience, the larger $\dot{V}_s$ is used for both summer and winter.

The DX coil load required at summer design conditions can be calculated from Eq. (20.48*b*)

$$Q_{cc} = 60\dot{V}_s\rho_s(h_{ae} - h_{cc}) \tag{29.3}$$

and the heating coil load required to offset the heating load at winter design conditions can be calculated from Eq. (20.39)

$$Q_{ch} = 60\dot{V}_s\rho_s c_{pa}(T_1 - T_2) \tag{29.4}$$

Controls

For a single-zone constant-volume system used in most residences in the United States, the primary control is the zone temperature control; and a dual thermostat with two set points, for separate cooling and heating mode operation during year-round zone temperature control, is used, as discussed in Sec. 20.19.

For single-zone constant-volume systems used in many industrial and commercial buildings where morning warm-up heating mode should be automatically changed over to cooling mode when the space is occupied, a single-set-point heating and cooling mode automatic changeover thermostat should be used. There is often a deadband of 1 to 3°F (0.6 to 1.7°C) between the heating and cooling modes, or vice versa.

In addition to the zone temperature control, there are specific controls such as ignition control, capacity conrols, head pressure controls, and safety controls. The burner is always energized first and then the supply fan. Cooling and heating capacity controls include on/off and two-speed; and several heating-stage step controls are often used.

Energy Use Intensities

According to DOE/EIA 1998 Commercial Buildings Consumption and Expenditure 1995, the energy use intensity of the single-zone constant-volume packaged systems (or residential central systems), in 1995 in the United States was 44 kBtu/ft$^2\cdot$yr (139 kWh/m$^2\cdot$yr). This EUI was only smaller than that in central air conditioning systems using chilled water as the cooling medium.

System Characteristics

System characteristics of a single-zone, constant-volume packaged system (SZCVPS) are listed in Table 29.1.

TABLE 29.1 System Characteristics of Packaged Systems

	SZCVPS	SZVAVPS	VAVCPS, VAVRPS, PHVAVPS
Zone thermal and sound control			
Control zone	Single-zone	Single-zone	Multizone
Control methods	Electric, two-stage thermostat	DDC, electric	DDC
Control modes	On/off	PI, PID, on/off	PI, PID, on/off
Heating-cooling mode changeover	Manual, automatic	Automatic	Automatic
Sound control	35–45 dBA	NC 30–45	NC 30–45
Indoor air quality (IAQ)			
Minimum ventilation control	Constant-volume	DCV, MPC	DCV, MPC
Filters	Low- or medium-efficiency	Low-, medium-, high-efficiency	Low-, medium-, high-efficiency
Humidity control	Optional	Optional	Optional
Types of humidity control	Heating element, wetted element	Heating element, wetted element	Heating element, wetted element
Air systems			
Types	Constant-volume air mixing	VAV, air mixing	VAV, air mixing
Supply fan	Forward-curved centrifugal	Centrifugal, forward, airfoil	Centrifugal, forward, airfoil
Fan total pressure	2.5 in. WC	4.0 in. WC	4.5 in. WC
Combined fan, motor, and drive efficiency	40%	40%	45%
Volume flow control	Constant-volume	Inlet vanes, variable-speed drive (VSD)	Inlet vanes, (VSD)
Relief/return fan total pressure	N/A	0.4–1.0 in. WC	0.4–1.0 in. WC
Combined fan, motor, drive efficiency		35%	45%
Economizer	Air, water	Air, water	Air, water
Cooling system			
Refrigeration compressor	Scroll, reciprocating	Scroll, reciprocating	Scroll, reciprocating
Capacity control	On/off cycling	On/off cycling, hot-gas bypass	Cycling, hot-gas bypass
Refrigerants	HCFC-22, HFC-407C, HFC-410A	HCFC-22, HFC-407C, HFC-410A	HCFC-22, HFC-407C, HFC 410A
Evaporator	DX coil	DX coil	DX coil
Condenser	Air-cooled	Air-cooled, evaporatively cooled	Air-cooled, evaporatively cooled
Refrigerant flow control	Expansion valve (TXV), capillary tube	TXV, restrictor	TXV, electric expansion valve
Energy performance	$<$ 5 tons, 9.3 EER		10–12 EER
Heating system			
Types of heating system	Gas furnace, heat pump	Gas furnace, heat pump, electric heating	VAVCPS N/A; VAVRPS and PHVAVPS: hot water, electric heating
AFUE	Gas furnace, conventional, 78%, condensing 93%	Gas furnace, conventional 78%, condensing 93%	Boiler, conventional 80%, condensing 93%
Maintenance			
Fault detection and diagnostics	Packaged unit	Packaged unit	Packaged unit

N/A means not available.

29.3 SINGLE-ZONE VAV PACKAGED SYSTEMS

System Description

In a single-zone VAV packaged system, zone temperature is maintained by a DDC controller through variation of the opening of the inlet guide vanes at the supply fan inlet or the supply fan speed through a variable-speed drive. Therefore, the supply volume flow rate is varied to match the zone load variation. A typical single-zone VAV packaged system is similar to that shown in Fig. 21.1 except for the following:

- The AHU is replaced by a packaged unit.
- The water-cooling coil is replaced by a DX coil.
- The heating coil is replaced by a gas furnace with primary and secondary heat exchangers.

An air economizer or a water economizer is often used to save energy. The cooling mode operation at design conditions and part-load operation and the heating mode operation at design and part-load operation are similar to that discussed in Sec. 21.2.

System Calculations

The supply volume flow rate to offset the zone sensible load at cooling mode design conditions can be calculated by Eq. (21.1) as

$$\dot{V}_s = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_s)} \tag{29.5}$$

The DX coil load required at cooling mode design conditions can be calculated by Eq. (21.2) as

$$Q_{cc} = 60\dot{V}_s\rho_s(h_{rn} - h_{cc}) \tag{29.6}$$

The heating capacity of the gas furnace required to offset the single-zone heating load during heating mode design conditions can be calculated by Eq. (21.3) as

$$Q_{ch} = 60\dot{V}_s\rho_s c_{pa}(T_{hc} - T_{en}) \tag{29.7}$$

Controls

Zone temperature control and sequence of operations are similar to those discussed in Sec. 21.2 except for the following:

- Cooling capacity is varied by cycling multiple compressors, or cylinder unloaders, or hot gas pass at low part-load operation. Cooling capacity control is discussed in later sections.
- Heating capacity is varied by controlling the gas valve in on/off or two-stage control.
- Discharge air temperature control is still required for a better control quality and energy saving. However, for a small packaged unit with only on/off cooling capacity control, zone temperature control in cooling mode operation is actually the same whether the single-zone VAV packaged system is employed with or without a discharge air temperature control.

Minimum ventilation control, head pressure control, and safety controls are similar to those discussed in Chap. 23. Shirey (1995) recommended an improved fan cycling strategy based on the field operating experience in the Salvador Dali Museum, St. Petersburg, Florida. The museum areas (storage

area and lobby) are primarily under thermostat control. Prior to the improvements, all indoor fans operated continuously, regardless of compressor operation, to maintain ventilation and air circulation for occupant comfort. Continuous fan operation resulted in large humidity fluctuations and increased indoor humidity levels at night when space load was reduced. Continuous fan operation also consumed more energy and needed additional compressor operation to offset fan heat. If there are wetted surfaces around the coil and condensate drain pan when the compressors are cycling off, moisture will be evaporated from these wetted surfaces, extracted by the indoor fan, and supplied to the conditioned space.

An alternative improved control strategy "automatic fan mode" was adopted. The indoor fan energized only when the compressors and heaters were operating. When the thermostat and humidistat set points are met, the indoor fan stops and the moisture drains from the packaged unit. The indoor fan that serves the storage room in the Salvador Dali Museum was changed to automatic fan mode control. The new control strategy provided a better storage indoor environment with significant energy savings. For the lobby, daytime continuous fan operation for constant air circulation was used when the lobby was open to the public. Automatic fan mode was used at nighttime.

System Characteristics

System characteristics of a single-zone VAV packaged system are listed in Table 29.1. In Table 29.1, DCV represents demand-controlled ventilation and MPC indicates mixing plenum pressure control.

29.4 COMPLIANCE WITH STANDARD 90.1–1999 THROUGH SIMPLIFIED APPROACH OPTION FOR SMALL AND MEDIUM HVAC&R SYSTEMS

Single-zone, constant-volume and single-zone, variable-air-volume packaged systems are the two widely used air conditioning systems in residential, commercial, and industrial buildings in the United States. For air conditioning systems in buildings two stories or less in height and with less than 25,000 ft^2 (11,800 m^2) gross floor area shall be considered in compliance with the requirements of HVAC in ASHRAE/IESNA Standard 90.1-1999 by means of a simplified approach if the following criteria are met:

1. The system serves a single zone.
2. A single or split-packaged unit is used which is either air-cooled or evaporatively cooled and meets minimum efficiency requirements listed.
3. An air economizer and control shall be installed as required in Sec. 21.2 with either barometric or powered relief sized to prevent space overpressurization.
4. Heating shall be provided by packaged heat pumps, PTHPs, gas-fired warm air furnaces, electrical heaters, or hot water heating systems and boilers with all applicable efficiency requirements met.
5. Outdoor air supplied by the system shall be equal to or less than 3000 cfm (1420/L/s) and less than 70 percent of the supply air quantity at minimum outdoor air design conditions unless an energy recovery ventilation system is provided.
6. The system is controlled by a manual changeover or dual set point thermostat.
7. Controls shall be provided to prevent auxiliary electric heater operation when the heat pump alone can meet the required heating load.
8. Controls shall not permit reheat or simultaneous heating and cooling.
9. For spaces other than those requiring continuous operation having a heating or cooling capacity greater than 65,000 Btu/h (30.7 kW), and having a supply fan motor power greater than 3/4 hp (0.56 kW), a timeclock shall be provided
   - ***a.*** To start and stop the system under different schedules,
   - ***b.*** That is capable of retaining programming and time setting during a loss of power of at least 10 hours
   - ***c.*** That has an accessible manual override allowing temporary operation up to two hours

d. That is capable of being set back down to 55°F (12.8°C) and up to 90°F (32.2°C) during off hours

10. HVAC&R piping, ducts, and plenums shall be insulated and meet the requirements listed in Secs. 7.2, 17.2, and 17.4.

11. Duct systems shall be air balanced to within 10 percent of design airflow rates.

12. When separate heating and cooling equipment serve the same control zone, thermostats shall be interlocked to prevent simultaneous heating and cooling

13. Exhausts with a volume flow rate over 300 cfm (140 L/s) that do not operate continuously shall be equipped with gravity or motorized dampers that will automatically shut when the exhaust systems are not in use.

14. Air systems with a design volume flow rate greater than 10,000 cfm (4720 L/s) shall have optimum start controls.

29.5 VAV COOLING PACKAGED SYSTEMS

System Description

A VAV cooling packaged system is a multizone air conditioning system that provides conditioned air without heating. It uses a packaged unit with DX cooling coils to condition the air and distribute it to various control zones through VAV boxes, ducts, distributing devices, and controls. The zone supply volume flow rate is modulated by the damper in the VAV box to match the variation of the zone sensible load to maintain a preset zone temperature. A typical VAV cooling packaged system with its system pressure characteristics is shown in Fig. 29.1. It consists of a packaged unit, many

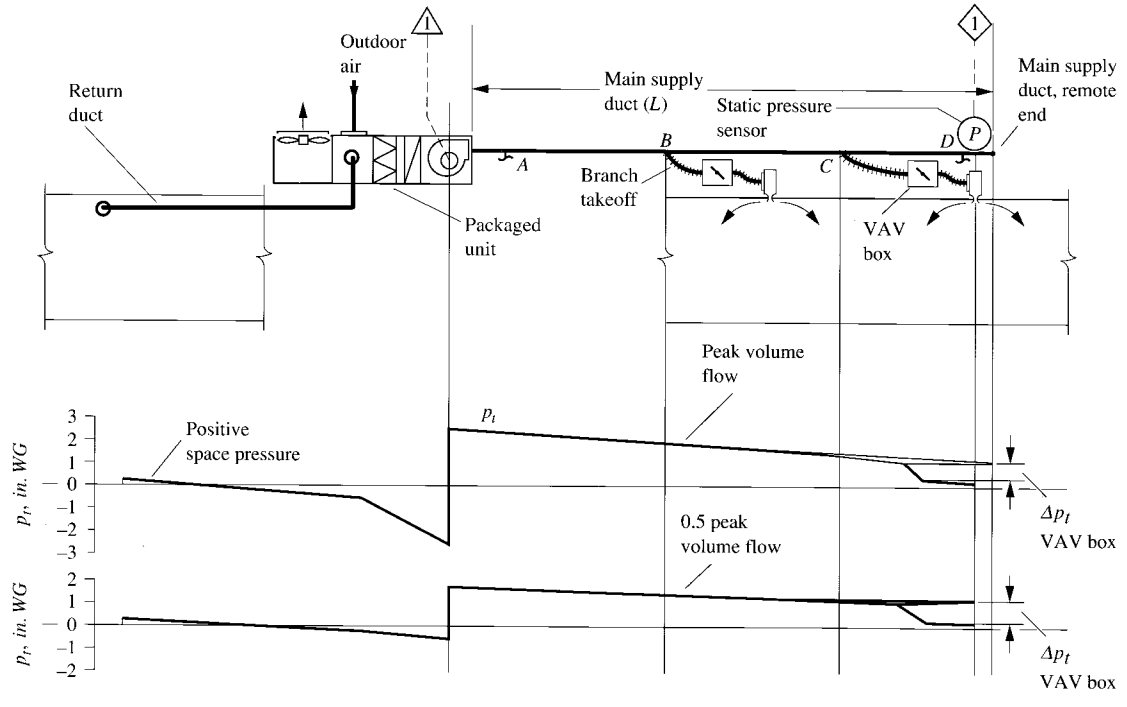


FIGURE 29.1 A VAV cooling packaged system.

VAV boxes, ducts, diffusers, and controls. The air conditioning cycle of a VAV cooling packaged system is the same as the air conditioning cycle in cooling mode operation for the interior zone in Fig. 21.5*a*. VAV cooling packaged systems are used for interior zones in buildings where the heating load is negligible.

Supply Volume Flow Rate and Coil Load

The supply volume flow rate for a control zone in the interior zone V_{sin}, in cfm [$m^3/(60\ s)$], or the supply volume flow rate of interior zone $\dot{V}_{si}$, in cfm [$m^3/(60\ s)$], can be calculated by Eq. (21.1) as

$$\dot{V}_{sin} = \frac{Q_{rin}}{60\rho_s c_{pa}(T_r - T_s)}$$
$$\dot{V}_{si} = \frac{Q_{ri}}{60\rho_s c_{pa}(T_r - T_s)} \tag{29.8}$$

where Q_{rin}, Q_{ri} = sensible cooling load of control zone n and interior zone, Btu/h (W). The DX coil load Q_{cc}, in Btu/h (W), of the packaged unit that serves the interior zone at cooling mode design conditions can be calculated by Eq. (21.2) as

$$Q_{cc} = 60\dot{V}_s\rho_s(h_m - h_{cc}) \tag{29.9}$$

Pressure Characteristics and Duct Static Pressure Control

In a VAV cooling packaged system with a system total pressure loss of 4.75 in. WC (1188 Pa), as shown in Fig. 29.1, the total pressure loss, in in. WC, between various sections of the system at peak supply volume flow may have values as follows:

	Flow, cfm	
	Peak flow	0.5 Peak flow
Return system	0.40	0.10
PU	2.25	0.56
Supply main duct	1.15	0.29
VAV box	0.60	0.90
Flexible duct and slot diffuser	0.40	0.10

The purpose of duct static pressure control is to achieve the following:

- Limit the maximum static pressure in the supply main ducts and maintain a preset static pressure at the point where the static pressure sensor is located.
- Provide the required volume flow rate of supply air for any branch takeoff at both design load and part-load conditions, and prevent starving of any VAV box.
- Minimize fan energy use at part load.

A starving VAV box is a VAV box with a supply volume flow rate lower than the amount required to offset the zone load. This is because the set point of the duct static pressure control is not properly set, the static pressure sensor is not properly located, or the supply duct and branch takeoffs are not properly designed and sized. If the static pressure sensor is located at the end of the main supply duct just before the last branch takeoff at point *D*, shown in Fig. 29.1, at peak supply

volume flow rate when the total pressure loss of the VAV box, flexible ducts, and slot diffuser in a branch takeoff $\Delta p = 1.0$ in. WC (250 Pa), the static pressure at point *D* is 1.0 in. WC (250 Pa). Point *C* is located upstream of point *D*. At peak supply volume flow rate, the mean air velocity inside the main supply duct at point *B*, denoted by v_B, is 1500 fpm (7.5 m/s), at point *C* the mean velocity $\boldsymbol{V}_c$ is 1200 fpm (6 m/s), and at point *D* the mean velocity v_D is 1000 fpm (5 m/s). The total pressure loss between points *B* and *C* along the main supply duct is 0.3 in. WC (75 Pa), and between points *C* and *D* it is 0.25 in. WC (63 Pa).

During part-load operation, when the volume flow rate has been reduced to 0.5 of peak supply volume flow, the static pressure at point *D* is maintained at 1.0 in. WC (250 Pa). From Eq. (17.11),

$$p_t = p_s + p_v \tag{29.10}$$

and from Eq. (17.12)

$$p_{t1} = p_{t2} + \Delta p_f \tag{29.11}$$

If the duct static pressure sensor is relocated at point *C*, then the pressure characteristics in the supply main duct at points *B*, *C*, and *D* at peak and 0.5 of peak supply volume flow rate, in in. WG (Pa), can be calculated as follows:

	Point *D*		Point *C*		Point *B*
Peak supply volume flow rate					
Static pressure p_s	0.78		1.00		1.25
Velocity pressure p_v	0.062		0.09		0.14
Total pressure $p_t = p_s + p_v$			1.09		
$p_{t1} = p_{t2} + \Delta p_f$	0.084				1.39
Total pressure loss Δp_f		0.25		0.30	
0.5 peak supply volume flow rate					
Static pressure p_s	0.944		1.00		1.062
Velocity pressure p_v	0.015		0.022		0.035
Total pressure $p_t = p_s + p_v$			1.022		
$p_{t1} = p_{t2} + \Delta p_f$	0.959				1.097
Total pressure loss between *C* and *D*, Δp_f		0.063		0.075	

From these analyses, the following can be noted:

- At peak (design) supply volume flow rate, points upstream of point *C* where the duct static pressure sensor is relocated (such as point *B*), both total pressure $p_{t,\text{up}}$ and static pressure $p_{s,\text{up}}$ of the airstream inside the duct, are higher than duct total pressure and static pressure at point *C*, denoted by p_{tC} and p_{sC}. Points downstream of point *C* (such as point *D*) $p_{t,\text{do}}$ are lower than p_{tC}, and $p_{s,\text{do}}$ are lower than p_{sC}.
- At part-load operation, such as 0.5 of peak supply volume flow rate, $p_{t,\text{up}} > p_{tC}$, $p_{s,\text{up}} > p_{sC}$, $p_{t,\text{do}} < p_{tC}$, and $p_{s,\text{do}} < p_{sC}$. But their differences are smaller than in design load.
- When duct static pressure sensor is located at point *C*, compared with the static pressure sensor located at point *D*, at part-load operation of 0.5 of peak supply volume flow rate, the duct static pressure at point *D* or at points downstream from point *D*, $p_{s,\text{do}} < 1.0$ in. WG. That is, the VAV box connected to points downstream from where the static pressure sensor is located may starve.
- It is the total pressure of the airstream inside the supply main duct (duct total pressure) that offsets the total pressure loss in the branch takeoff. However, duct static pressure is steadier than the duct total pressure, and duct static pressure is usually sensed in a supply main duct for duct static pressure control.

As discussed in Sec. 23.8, the set point of the duct static pressure control is the duct static pressure plus the associated velocity pressure, i.e., the total pressure required to overcome the

maximum total pressure loss in any of the branch takeoffs connected to the supply main duct. And 0.1 in. WG (25 Pa) should be added as a safety factor.

For a supply duct system with similar configuration in branch take-offs, the static pressure sensor of a duct static pressure control should be located near the remote end of the main duct as well as at a location where steady static pressure can be properly measured, as shown in Fig. 29.1.

If there are two or three supply main ducts, two or three static pressure sensors should be installed. Each should be located near the remote end of each main duct. A comparator is used so that the DDC controller can pick the lowest static pressure as the feedback value to modulate the variable-speed drive to provide a duct static pressure higher than the preset value in all main ducts.

System Characteristics

System characteristics of a VAV cooling packaged system are listed in Table 29.1.

29.6 VAV REHEAT PACKAGED SYSTEMS

System Description

A VAV reheat packaged system is a multizone system that uses a packaged unit with DX coil and filters to condition the air and that supplies the conditioned air to various control zones in the perimeter zone of a building through reheating VAV boxes, ducts, and diffusers, and to various control zones in the interior zones through VAV boxes, ducts, and diffusers. The zone supply volume flow rate of the cold supply air is modulated to match the variation of the zone loads during part-load operation.

Heating provided by the reheating coil is used to offset the zone heating load in winter in the perimeter zone as well as to prevent the zone temperature drops below a preset temperature when the zone supply volume flow rate has been reduced to the minimum setting during cooling mode part-load operation. A reheating coil is not required in the VAV boxes in the interior zone.

As discussed in Sec. 21.3, for energy saving, the zone supply volume flow rate must be reduced to a minimum setting, such as 30 percent of the peak supply volume flow rate during cooling mode part-load operation before the reheating coil is energized. Figure 29.2 shows a VAV reheat packaged system with a rooftop packaged unit. The air conditioning cycle for cooling mode operation is the same as that shown in Fig. 21.5*b*. Usually, the same rooftop packaged unit is used to supply conditioned air to both perimeter and interior zones. However, in winter, if two packaged units are used to serve the perimeter and interior zone separately, the mixing air temperature for the perimeter zone can be raised to a higher value, to save energy.

VAV reheat packaged systems are simple and effective. However, simultaneous cooling and heating processes should be minimized. VAV reheat packaged systems are used in many commercial buildings.

Supply Volume Flow Rate and Coil Load

In a VAV reheat packaged system, the zone peak supply volume flow rate $\dot{V}_{sn}$, in cfm [m^3/(60 s)], can be calculated from Eq. (21.4) as

$$\dot{V}_{sn} = \frac{Q_{rsn}}{60\rho_s c_{pa}(T_r - T_{sn})} \tag{29.12}$$

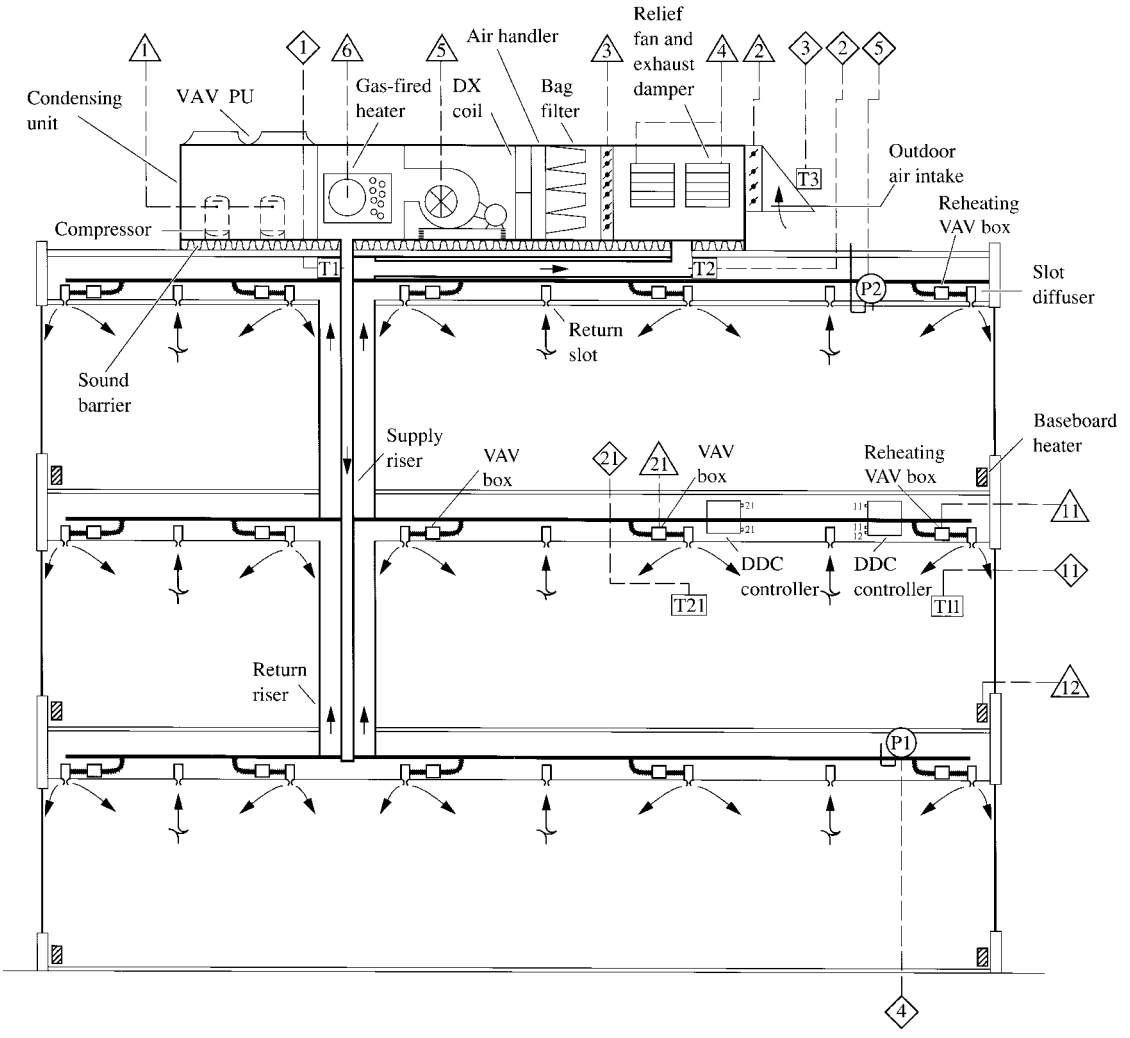


FIGURE 29.2 A VAV reheat packaged system.

From Eq. (21.5), the block supply volume flow rate for perimeter zone $\dot{V}_{sx}$, in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sx} = \frac{Q_{rsx}}{60\rho_s c_{pa}(T_r - T_{sx})} \tag{29.13}$$

From Eq. (21.6), the block supply volume flow rate for interior zone $\dot{V}_{si}$, in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{si} = \frac{Q_{rsi}}{60\rho_s c_{pa}(T_r - T_{si})} \tag{29.14}$$

From Eq. (21.8), the DX coil load Q_{cc}, in Btu/h (W), can be calculated as

$$Q_{cc} = 60\dot{V}_s\rho_s(h_m - h_{cc}) \tag{29.15}$$

The reheating coil load depends primarily on the winter zone heating load. From Eq. (21.10), the reheating coil load for a control zone in the perimeter zone Q_{chxn}, in Btu/h (W), can be calculated as

$$\begin{aligned} Q_{chxn} &= Q_{rhxn} + Q_{venxn} \\ &= Q_{rhxn} + 60\dot{V}_{sxn}\rho_s c_{pa}(T_{rxn} - T_{enn}) \end{aligned} \tag{29.16}$$

Night Setback and Morning Warm-up

In multizone VAV reheat rooftop packaged systems, the heating of the zone air during the occupied period in the perimeter zone in winter is often provided by zone electric reheating coils, or sometimes by water heating coils. The gas furnace, heat pump, or electric heater in the rooftop packaged unit is only used to maintain a night setback temperature and for morning warm-up purposes.

The reasons to supply warm air to maintain an indoor night setback temperature (such as 55°F, or 12.8°C) are

- To prevent freezing of water pipes and water surfaces in areas where the outdoor temperature at night is below 32°F (0°C)
- To provide an acceptable indoor temperature for emergency access
- To reduce the time required to warm up to a required temperature, say, 68 or 70°F (20.0 or 21.1°C), prior to an occupied period the next morning

During the night setback and morning warm-up periods, the following hold:

1. Outdoor dampers and exhaust dampers should be completely closed.
2. Recirculating dampers, inlet vanes, and inlet cones of the supply fan should be fully open; or the variable-speed drive should be running at full speed.
3. All VAV boxes or reheating boxes should be fully open.
4. Both the refrigeration compressor and the relief fan should be turned off.
5. When the zone temperature exceeds a certain limit, both the supply fan and the furnace, heat pump, or heater in the packaged unit can be turned off. The heating device in the packaged unit will be energized again when the zone temperature drops below a certain limit.
6. The termination of the night setback period is the beginning of the warm-up period. The warm air supply temperature from the rooftop packaged unit during the warm-up period is generally between 100 and 120°F (37.8 and 48.9°C).

Evenly Distributed Airflow at DX Coils

In a VAV reheat packaged system, if the air flowing through the DX coil is not evenly distributed over the entire coil surface, then liquid slugging of the reciprocating compressor, hunting of the thermostatic expansion valve, and a decrease in the DX coil capacity may all occur at the same time.

Refrigerant enters the various refrigerant circuits of the DX coil and the evaporator, typically as a mixture of 75 percent liquid and 25 percent vapor, after passing through the thermostatic expansion valve and the distributor tubes. If some of the refrigerant circuits have heavy refrigeration loads and others have only very light loads, the refrigerant in circuits with heavy loads expands rapidly to vapor, resulting in a greater vapor velocity and greater pressure loss. The refrigerant in circuits with very low loads remains in a liquid state and flows to the compressor in the form of

liquid slugging. Liquid slugging may cause reciprocating compressor failure. Therefore, an even airflow for the DX coil and thorough upstream air mixing must be maintained for both full- and part-load operations, especially using reciprocating compressors. Liquid refrigerant lowers the temperature of the sensing bulb of the thermostatic expansion valve and causes hunting.

Discharge Air Temperature Control for Packaged Systems

For a VAV reheat packaged system using reciprocating rooftop unit with a cooling capacity greater than 20 tons (70 kW), its compressors are often controlled by using four-step capacity controls. Multiple compressors are usually equipped in medium-size and large packaged units. Each compressor may have two or more cylinders, so that two can be loaded and two unloaded at the same time (see Fig. 11.23).

Air-Side Economizer Mode. For a typical DDC controller-activated discharge air temperature control, as shown in Fig. 5.6, if the set point of the discharge temperature T_{dis} is set at 53°F (11.7°C), which gives a supply temperature at the slot diffuser of $T_s = 55$°F (12.8°C), there will be a control band of 4°F (2.2°C). When a VAV reheat packaged system operates in an air economizer mode and $51 < T_{dis} < 55$°F ($10.6 < T_{dis} < 12.8$°C), the system will float within the control band by mixing outdoor air with recirculating air.

Initiation of Cooling Stages. If $T_{dis} > 53$°F (11.7°C) and due to the air economizer alone can no longer balance the sensible coil load, then T_{dis} floats to the upper limit of the control band. Once it reaches point 1, 55°F (12.8°C) in Fig. 5.6, the first-stage cooling is energized and T_{dis} drops below 55°F (12.8°C). First-stage cooling is most likely provided by the cooling capacity of two cylinders in a compressor. It cycles on and off, and T_{dis} floats within the control band, with proportional-integral (PI) control mode to maintain $T_{dis} = 53$°F (11.7°C) with the least deviation. When the first-stage cooling is turned off during cycling, it needs a time delay of at least 4 min before it can be turned on again, in order to prevent hunting and possible damage to the compressor motor. If, after the first-stage cooling has been energized, the DX coil capacity still cannot balance the sensible coil load, then T_{dis} continues to rise until it reaches point 2, which is 1°F (0.56°C) higher than the upper limit of the control band.

If the time interval between the time the first stage turns on and the instant when T_{dis} reaches point 2 is greater than 4 min, the first-stage cooling is then locked on and the second-stage cooling is energized to cycle on and off in an attempt to maintain T_{dis} at 53°F (11.7°C). As in the first stage, there must be a time delay of 4 min between the time the second-stage cooling turns off and the monent it can be turned on again. When T_{dis} floats within the control band, it may drop to point 3 because of the low sensible coil load. If the cooling capacity of having the first-stage cooling locked on and the second-stage cooling cycling still cannot offset the sensible coil load, T_{dis} will rise until it reaches a value 1°F (0.56°C) higher than the upper limit of the control band, such as point 4. At point 4, the third-stage cooling is cycling, and the first- and second-stage cooling is locked on.

In this type of packaged unit, the greatest cooling capacity is provided when the fourth-stage cooling is cycling and the first-, second-, and third-stage cooling is locked on.

As T_{dis} floats within the control band, any deviation from the set point of 53°F (11.7°C) is integrated over time as part of the proportional-integral control mode, in an attempt to reduce the deviation to zero, i.e., to maintain a near-constant temperature at the set point of 53°F (11.7°C), e.g., points 5 and 6 in Fig. 5.6.

If T_{dis} is being controlled in air economizer and cooling mode with refrigeration, the greater the deviation from the set point, the shorter the time needed for corrective action in order to provide control stability.

When a decrease in the coil load causes T_{dis} to drop to a value 1°F (0.56°C) lower than the lower limit of the control band (e.g., point 7), the currently cycled stage is locked off and the next-lower cooling stage becomes the cycling stage.

Successive decreases in T_{dis} to 1°F (0.56°C) below the lower limit of the control band cause repeated locking off of cooling stages until the second stage is locked off and the first stage becomes the cycling stage. When the first stage is cycled off, the liquid line solenoid valve is deenergized, but the compressor still operates to pump down the refrigerant to the condenser. When the suction pressure in the DX coil becomes lower than the limit of low-pressure control, it opens the compressor circuit, and both the compressor and the condenser fans are stopped. A fully intertwined DX coil will adjust its activating refrigerant circuits accordingly during step control of the compressor capacity.

At design load, the evaporating temperature inside the DX coil is usually between 40 and 45°F (4.4 and 7.2°C). Hot gas bypass is only used to prevent the DX coil from frosting in case T_{ev} falls below 32°F (0°C) because of a sudden load decrease.

Reset. For a DDC-controlled VAV packaged system that uses a proportional-integral control mode, T_{dis} can be reset based on either the space air temperature floating within the deadband or the outdoor temperature T_o. When T_{dis} is reset based on the space temperature, it is preferable to have several space temperature sensors connected in series. Their average temperature T_{rm}, in °F (°C), is used to reset T_{dis}. As T_{rm} floats within the deadband between 70 and 75°F (21.1 and 23.9°C), each 1°F (0.56°C) drop in T_{rm} corresponds to a 1 to 2°F (0.56 to 1.1°C) increase in T_{dis}.

During cooling mode, if T_{dis} is reset based on the outdoor temperature T_o, then T_{dis} will not be reset when $T_o \geq 70$°F (21.1°C). After T_o falls below 70°F (21.1°C), for every 5°F (2.8°C) that T_o drops, T_{dis} will increase 1°F (0.56°C). For instance, at an outdoor temperature $T_o = 50$°F (10.0°C), the set point of T_{dis} will be 57°F (13.9°C). For larger packaged units with multiple scroll compressors, cooling capacity is often divided into four capacity steps such as when reciprocating compressors are used.

Fan Modulation

Three types of supply fan modulation are widely used in large VAV packaged systems with rooftop packaged units to maintain a supply duct static pressure near the most remote branch takeoff in the supply main duct: (1) a forward-curved centrifugal fan with inlet vanes, (2) an airfoil centrifugal fan with a variable-speed drive, and (3) an airfoil centrifugal fan with inlet cone modulation. Because of the smaller fan inlet in an airfoil centrifugal fan, it is not recommended that inlet vanes be used, in order to prevent an extremely high inlet velocity, such as the actual maximum inlet velocity which exceeds 5000 fpm (25 m/s).

Both forward-curved centrifugal fans with inlet vane modulation and airfoil centrifugal fans with inlet cone modulation have a lower cost than airfoil fans that use a variable-speed drive. However, variable-speed drive fan modulation is more energy-efficient at part-load operations. A cost analysis is always helpful to determine whether having a variable-speed drive or having inlet vanes is optimum.

In a VAV packaged system using inlet vanes for its forward-curved centrifugal fan modulation, consider the following:

- Because all inlet vanes are subject to a certain amount of air leakage at the completely closed position, the fan volume flow rate will still be about 10 to 30 percent of the design flow when the inlet vanes are completely closed. A modulation range from 100 to about 30 percent is usually sufficient for a VAV packaged system.
- A forward-curved centrifugal fan has a smaller fan surge area at a lower fan total pressure and flow rate than an airfoil centrifugal fan.
- Fan modulation resulting in an external total pressure drop greater than 4 in. WC (100 Pag) in a VAV packaged unit may cause damage to the unit.
- A discharge damper should not be installed because closing the VAV boxes can provide the same shutoff function. A discharge damper may cause excessive noise due to overpressurization of the VAV boxes. It may also overpressurize the ducts.

A VAV packaged system is shut off during nighttime unoccupied hours in summer and off seasons. In locations where there is a cold winter, it operates to provide warm air and maintain a night setback indoor space temperature during unoccupied hours.

Air-Cooled, Water-Cooled, and Evaporatively Cooled Condensers

In rooftop packaged units using reciprocating compressors, the energy efficiency ratio (EER) for air-cooled, water-cooled, and evaporatively cooled condensers at an outdoor dry-bulb temperature of 95°F (35°C) and a wet-bulb temperature of 75°F (23.9°C) is as follows:

Air-cooled condenser	9 – 12
Water-cooled condenser	11 – 14
Evaporative-cooled condenser	14 – 16

The EER for VAV packaged units using scroll compressors is higher than that for those using reciprocating compressors. Air-cooled and evaporatively cooled condensers often have a higher installation cost than water-cooled condensers and cooling towers.

Although water-cooled and evaporatively cooled condensers are considerably more energy-efficient at an outdoor condition of 95°F (35°C) dry-bulb and 75°F (23.9°C) wet-bulb temperatures than air-cooled condensers, in many locations the annual average dry-bulb temperature is rather low. Shaffer (1987) used a manufacturer's computer program to compare the annual energy use of air-cooled and water-cooled reciprocating chillers for multistory buildings using VAV reheat systems in Boston and San Diego. The annual energy use of a water-cooled condenser and cooling tower is about 10 percent higher than that of an air-cooled condenser, mainly due to the energy use of condenser pumps. Comparing the energy use and costs of air-cooled, water-cooled, and evaporatively cooled condensers is recommended in order to make an energy-efficient and cost-effective selection.

Controls and Sound Problems for Rooftop Packaged Units

Microprocessor-based specific controls, safety controls, and diagnostics for a 1997 manufactured rooftop packaged unit with a cooling capacity between 20 to 130 tons (70 to 457 kW) are listed in in Sec. 16.5. Noise considerations for rooftop top packaged units are discussed in Sec. 19.9.

Case Study: VAV Reheat Packaged System for Precision Manufacturing

Desmone and Frank (1992) reported an addition of 10,000 ft^2 (930 m^2) for sophisticated precision manufacturing for Oberg Industries, a manufacturer of high-quality carbide and steel stamping dies in Freeport, Pennsylvania. The required HVAC&R criteria are as follows:

- Temperature is maintained at a constant level of 68 ± 1°F (20 ± 0.56°C) not only horizontally but also vertically from the floor to 13 ft (3.96 m) high.
- Humidity should be maintained at 50 percent.
- Airflow noise had to be limited and drafts and turbulence should be tightly constrained.

The selected HVAC&R system is a VAV reheat packaged system with two rooftop packaged units that feed cold air year-round into a medium-pressure duct loop. Machine oil duct exhaust, grinding duct exhaust, conditioned air return, medium-pressure supply duct, and the lowest VAV boxes and reheating hot water piping are arranged in layers to facilitate servicing.

There are 19 pneumatically controlled VAV boxes located throughout the extended plant with a two-stage modulation in sequence. The modulation of the supply air volume flow by the damper in

the VAV box provides the first-stage control. If the zone cooling load drops below the minimum setting of air delivery by the VAV box, the pneumatic controller opens the hot water valve, forcing the hot water to enter the reheating coil, and modulates the hot water flow to maintain the required exact zone temperature. The air economizer cycle is adopted to save energy. The HVAC&R system is designed with a 75 percent backup capacity for reliability.

System Characteristics

System characteristics of VAV reheat packaged systems are listed in Table 29.1.

29.7 PERIMETER-HEATING VAV PACKAGED SYSTEMS

A perimeter-heating VAV packaged system is a VAV multizone system that uses packaged units with DX coils to condition the air, During cooling mode operation, the conditioned air is supplied to the control zones in the perimeter and to interior zones through VAV boxes, ducts, and diffusers. In heating mode operation, heating is provided to the perimeter zone by a perimeter-heating system using either hot water finned-tube heaters or electric baseboard heaters to offset zone heating loads. Ventilation is provided by the cold ventilation air supplied from the VAV boxes at 30 percent of the peak supply volume flow rate minimum settting.

Perimeter VAV packaged systems are suitable for multizone commercial buildings in locations with long, cold winters.

Similar to that in Sec. 29.5, for VAV reheat packaged systems, the supply volume flow rate for control zones in the perimeter and in interior zones $\dot{V}_{sn}$, in cfm [m^3/(60 s)], can be calculated from Eqs. (29.12). The supply volume flow rates for perimeter or interior zone V_{sx} or V_{si}, in cfm [m/(60 s)], can be calculated from Eq. (29.13) or (29.14). The heating load for perimeter zone Q_{chx}, in Btu/h (W), can be calculated from Eq. (29.16).

System Characteristics

System characteristics of a perimeter-heating VAV packaged system are listed in Table 29.1.

29.8 FAN-POWERED VAV PACKAGED SYSTEMS

System Description

A fan-powered VAV packaged system (FPVAVPS) is a multizone system using a packaged unit to condition the air and to distribute the conditioned air through fan-powered VAV boxes, ducts, diffusers, and controls in the perimeter zone for an air system with conventional air distribution as shown in Fig. 21.13 except that a packaged unit is used to condition the air; or it distributes the conditioned air through the fan-powered VAV boxes, ducts, diffusers, and controls in either perimeter or interior zones for an air system with cold air distribution.

After the single-blade volume control damper has been closed to a minimum setting, such as 30 percent, the function of a fan-powered box is, first, to extract recirculating warm plenum air and mix it with the cold primary air and, second, to energize a reheating coil to maintain a preset zone temperature during part-load cooling mode operation. In winter heating mode, an electric heater in several capacity stages is controlled to maintain a preset zone temperature.

There are two types of fan-powered boxes: parallel fan-powered boxes and series fan-powered boxes. Parallel fan-powered boxes save fan energy more than series fan-powered boxes and are

more widely used. Fan-powered VAV packaged systems with parallel fan-powered boxes using an electric heater or hot water heating coils have been used in many commercial buildings.

Supply Volume Flow Rate and Coil Load

As discussed in Sec. 21.5, to calculate the zone peak supply volume flow rate at summer design conditions $\dot{V}_{sn,d}$, in cfm [$m^3/(60\ s)$], one must consider whether it is a control zone in the perimeter zone or in the interior zone, and whether a conventional air distribution is used or a cold air distribution is used.

- For conventional air distribution in the perimeter zone with a cooling supply temperature differential $(T_r - T_s) = 15$ to 20°F (8.3 to 11.1°C), from Eq. (21.18), the zone peak supply volume flow rate from the fan-powered box $\dot{V}_{sxn,d}$, in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sxn,d} = \frac{Q_{rsxn}}{60\rho_s c_{pa}(T_r - T_s)} \tag{29.17}$$

For conventional air distribution in interior zones, the zone peak supply volume flow rate from the VAV boxes $\dot{V}_{sin}$, in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sin} = \frac{Q_{rsin}}{60\rho_s c_{pa}(T_r - T_s)} \tag{29.18}$$

- For cold air distribution in the perimeter zone with a cooling supply temperature differential typically $T_r - T_{sc,\text{dis}} = 32$°F (17.8°C), the peak zone supply volume flow rate of the cold primary air from the fan-powered box $\dot{V}_{sxn,c}$, in cfm [$m^3/(60\ s)$], can be calculated as

$$\dot{V}_{sxn,c} = \frac{Q_{rsxn}}{60\rho_s c_{pa}(T_r - T_{sc,\text{dis}})} \tag{29.19}$$

As discussed in Sec. 21.5, the ratio between the recirculating plenum air from the fan-powered box and the peak zone supply volume flow rate $\dot{V}_{sxn,r}/\dot{V}_{sxn,c} = 0.67$. Also, $\dot{V}_{sxn,c} + \dot{V}_{sxn,r} = \dot{V}_{sxn}$. Therefore,

$$\dot{V}_{sxn,r} = \dot{V}_{sxn} - \dot{V}_{sxn,c} \tag{29.20}$$

where $\dot{V}_{sxn}$, $\dot{V}_{sxn,r}$ = peak zone supply volume flow rate and peak zone supply volume flow rate of recirculating plenum air of control zone *xn* in perimeter zone, cfm [$m^3/(60\ s)$]

- For cold air distribution in the perimeter zone during winter heating design conditions, the peak zone supply volume flow rate $\dot{V}_{sxn}$ should be calculated by Eq. (21.18) so that the supply temperature differential will be equal to or less than 15°F (8.3°C), to prevent excessive buoyancy and stratification.
- For cold air distribution in the interior zone, each control zone can be either with or without a fan-powered box, as discussed in Sec. 18.5. Supply of low-temperature air with high induction nozzle diffusers is recommended. When fan-powered boxes are used for cold air distribution in the interior zone, the supply volume flow rate of cold primary air $\dot{V}_{sin,c}$, in cfm [$m^3/(60\ s)$], can be similarly calculated by Eq. (29.19) except that the zone sensible cooling load in the perimeter zone Q_{rsxn} should be replaced by zone sensible cooling load in the interior zone Q_{rsin}, both in Btu/h (W). As the ratio $\dot{V}_{sin,r}/\dot{V}_{sin,c}$ is equal to 0.67, the zone peak supply volume flow rate of the recirculating air from the fan-powered box $\dot{V}_{sin,r}$, in cfm [$m^3/(60\ s)$], is therefore determined.
- From Eq. (29.13), the supply volume flow rate for the perimeter zone at summer design conditions $\dot{V}_{sx}$ can be calculated. From Eq. (29.14), the supply volume flow rate for the interior zone at summer design conditions $\dot{V}_{si}$ can be similarly calculated, both in cfm [$m^3/(60\ s)$].

- From Eq. (29.15), the DX coil load Q_{cc} in the packaged unit of a fan-powered VAV packaged system can be calculated, and from Eq. (29.16) the reheating coil load Q_{chxn} in each of the fan-powered boxes in the perimeter zone can be similarly calculated, both in Btu/h (W).

Controls

A fan-powered VAV packaged system has two air mixings: outdoor air and recirculating air mixing in the mixing plenum of the packaged unit, and the cold primary air and recirculating plenum air mixing in the fan-powered box. Minimum outdoor ventilation air control is critical. Both demand-controlled ventilation using CO_2 sensors and mixed plenum pressure control are discussed in Chap. 23 and should be carefully investigated during the peak supply volume flow rate as well as when the volume flow rates of the cold primary air in control zones have been reduced to minimum settings.

Zone controls and the sequence of operations of fan-powered boxes are discussed in Sec. 21.5. Microprocessor-based specific safety controls and diagnostics for a late-1990s manufacturered packaged unit are similar to those listed in Sec. 16.5.

Case Study: A Fan-Powered VAV Packaged System with Rooftop Packaged Unit

This case study is a retrofit project. In it, an HVAC&R system is used to serve a 48,000-ft^2 (4461-m^2) medical office building in Little Rock, Arkansas. The building was constructed in 1979. It is owned by a limited partnership of 16 physicians. The renovation of the HVAC&R system of this project won a 1992 ASHRAE Technology Award, second place, of existing commercial buildings.

As described by Tinsley et al. (1992), prior to the 1990 renovation, the HVAC&R system was a constant-volume electric terminal reheat rooftop packaged system. There were six rooftop packaged units with a total capacity of 120 tons (422 kW) and 162 electric duct heaters with an approximate total capacity of 450 kW.

For the renovation, a fan-powered VAV packaged system with a rooftop packaged unit was designed and installed. One rooftop packaged unit of 133 tons (468 kW) with 100 electric heated parallel fan-powered units was equipped to serve this medical office building.

The renovation mainly includes the following:

- The air economizer and the compressors are controlled in sequence by DDC discharge air temperature control in order to maintain a supply air temperature at 55°F (12.8°C). When the outdoor air temperature $T_o \leq 60$°F (15.6°C), all compressors are locked out.
- An energy-efficient evaporatively cooled condenser is used. The energy use of the compressors, tower fans, and spray pump at full load and at an outdoor wet-bulb temperature of 80°F (26.7°C) is 0.79 kW/ton (4.45 COP).
- An airfoil centrifugal fan with inlet cone modulation is used because it uses less fan energy than a forward-curved centrifugal fan with inlet vanes.
- A relief fan is used instead of a return fan. This relief fan operates only when the system is in air economizer mode.
- A lower duct static pressure control set point of 0.75 in. WG (188 Pag) is used.
- The fan-powered VAV packaged system operates only when any one of the 16 suites is occupied. When any suite is unoccupied, the primary air damper in the VAV box is completely closed. The fan and the electric heater in the parallel fan-powered box are controlled in sequence to maintain a zone temperature 10°F (5.6°C) lower than the zone temperature set point during unoccupied hours.
- All supply ducts have been sealed and insulated.

TABLE 29.2 System Characteristics of Fan-Powered VAV Packaged System and Desiccant-Based Air Conditioning Systems

	FPVAVPS	DBACS
Zone thermal and sound control		
Control zone	Multizone	Single-zone or multizone
Control methods	DDC or thermostat; high-medium-low fan speed	Thermostat, humidistat, or DDC
Control modes	On/off, PI, or PID	On/off, PI, or PID
Heating-cooling mode changeover	Automatic	Automatic, manual
Sound control	35–45 dBA	NC 30–45
Indoor air quality (IAQ)		
Minimum ventilation air control	Demand-controlled ventilation, or mixing plenum pressure control	Demand-controlled ventilation, or mixing plenum pressure control
Filters	Medium- or high-efficiency	Medium- or high-efficiency
Humidity control	Optional	Heating element humidifier and desiccant dehumidifier
Air systems		
Types	VAV air mixing	Constant-volume or VAV
Supply fan	Centrifugal, forward-curved or airfoil	Centrifugal, forward-curved or airfoil
Supply fan total pressure	4.5 in. WC	4.5–6 in. WC
Combined fan, motor, drive efficiency	40–45%	45–55%
Volume flow control	Inlet vanes, inlet cone, variable-speed drive (VSD)	N/A
Air economizer	Air, temperature, or enthalpy	N/A
Fan-powered box fan	Centrifugal, forward-curved	N/A
Fan total pressure	0.5 in. WC	
Combined fan, motor, drive efficiency	25%	
Relief/exhaust fan	Axial or centrifugal	Centrifugal
Fan total pressure	0.4–1.0 in. WC	1–1.5 in. WC
Combined fan, motor, drive efficiency	25–35%	30–40%
Cooling systems		
Refrigeration compressor	Scroll or reciprocating	Centrifugal, screw, scroll, or reciprocating
Refrigerants	HCFC-22, HFC-407C, HFC-410A	Centrifugal HCFC-123, screw HCFC-22, scroll HFC-407C, HFC-410A
Energy performance	10–12 EER	Centrifugal 0.5–0.55 kW/ton, screw 0.7–0.75 kW/ton, scroll 10–12 EER
Evaporator	DX coil	DX coil, liquid cooler
Condenser	Air-cooled, or evaporatively cooled	Air-cooled, water-cooled, or evaporatively cooled
Capacity control	On/off, multiple compressors, hot-gas bypass	Water flow, or multiple compressors
Heating system	Gas furnace, hot-water coil, or electric heater	Gas heater, electric heater, or hot-water coil
AFUE	Conventional 78%, condensing 93%	Conventional 78%, condensing 93%
Maintenance		
Fault detection and diagnostics	Packaged unit	

- The zone temperature can be maintained between 70 and 80°F (21.1 and 26.7°C) in any control zone by the DDC system.

The annual energy use for the HVAC&R system in this building before renovation was 169,150 Btu/ft^2·yr (533 kWh/m^2·yr), and the annual energy cost was \$2.92/ft^2·yr (\$31.4/m^2·yr). After renovation, the energy use intensity had dropped to 55,890 Btu/ft^2·yr (176 kWh/m^2·yr) and the energy cost was \$1.40/ft^2·yr (\$15.1/m^2·yr), drops of 67 and 52 percent, respectively.

System Characteristics

System characteristics of fan-powered VAV packaged systems are listed in Table 29.2.

29.9 DESICCANT-BASED AIR CONDITIONING SYSTEMS

Desiccant-Based Air Conditioning

A desiccant-based air conditioning process is a combination of desiccant dehumidification, evaporative cooling, supplementary compression refrigeration, and the regeneration or reactivation of the desiccant by means of waste heat or gas heating. A desiccant-based air conditioning system is a hybrid system of desiccant dehumidification, evaporative cooling, refrigeration, and regeneration systems to cool and dehumidify the space air and maintain it at a required temperature and relative humidity with adequate outdoor ventilation air, at the same time improving the efficiency of energy use. In 1983, ASHRAE awarded the Willis Carrier Prize to Nancy Banks, a 23-year-old engineer who proposed a desiccant dehumidification process with evaporative cooling and necessary supplementary refrigeration in commercial buildings. It was a new idea at that time that considerably improved the energy use of refrigeration and started a new era of commercial desiccant-based air conditioning.

Desiccant Dehumidification and Sensible Cooling

When moist air flows over a bed of either solid or liquid desiccant, absorption occurs at the bed of sorbents. Absorption is a moisture soption process associated with physical or chemical changes, while adsorption is one without physical or chemical changes. For liquid absorbents, such as liquid lithium chloride, the absorption of moisture is mainly caused by the vapor pressure difference between the moist air and the surface of the liquid. Solid adsorbents such as silica gels attract moisture because of the vapor pressure difference and the electric field at the desiccant's surface.

During the process of sorption, heat is released. This released heat, often called the heat of sorption, is the sum of the following:

- Latent heat due to the condensation of the absorbed water vapor into liquid
- Heat of wetting when the surface of the solid sorbent is wetted by the attached water molecules, or the heat of solution when moisture is absorbed by the liquid sorbents

Heat of wetting varies from a relatively large value when new desiccant first absorbs moisture to a very low value when it is nearly saturated.

During the desiccant dehumidification process, the humidity ratio of the leaving moist air is decreased. Although there is a heat loss through the outer casing of the dehumidifier to the ambient air, it is rather small compared with the release of the heat of sorption and any residual heat remaining after the regeneration process.

Regeneration uses heat to drive off the accumulated moisture so that the sorbent can be reused. The temperature of the dehumidified air is considerably increased because of the heat released

during sorption.The desiccant dehumidification process line deviates from the thermodynamic wet-bulb temperature line and is represented by line *di-do* on the psychrometric chart in Fig. 29.3.

The decrease in humidity ratio during the desiccant dehumidification process is given by

$$\text{Difference in humidity ratio} = w_{di} - w_{do} \tag{29.21}$$

where w_{di}, w_{do} = humidity ratio at dehumidifier inlet and outlet, lb/lb (kg/kg). The reduction of latent heat Q_l, in Btu/h (W), can be calculated as

$$Q_l = 60\dot{V}_o\rho_o(w_{di} - w_{do})h_{fg} \tag{29.22}$$

where $\dot{V}_o$ = volume flow rate of air entering dehumidifier, cfm [m^3/(60 s)]
ρ_o = air density of entering air, lb/ft^3 (kg/m^3)
h_{fg} = latent heat of vaporization, Btu/lb (J/kg)

Desiccant dehumidification is a dehumidification and heating process on the psychrometric chart, and it is illustrated by a straight line inclined downward, deviating slightly from the thermodynamic wet-bulb temperature line, as shown in Fig. 29.3. It has a flatter negative slope.

The acute angle between the desiccant humidification process and thermodynamic wet-bulb temperature depends mainly on the desiccant material and the water-holding capacity of the desiccant. When the desiccant approaches saturation, angle α is smaller than that when a new desiccant first absorbs moisture from the surrounding air. For a rotary desiccant dehumidifier impregnated with lithium chloride, if manufacturer's data are not available, angle α can be estimated at 5°.

The regeneration process removes the absorbed moisture from the desiccant. For the regeneration airstream, it is also a humidifying and cooling process. Molecular sieves require a higher regeneration temperature than silica gels and lithium chloride.

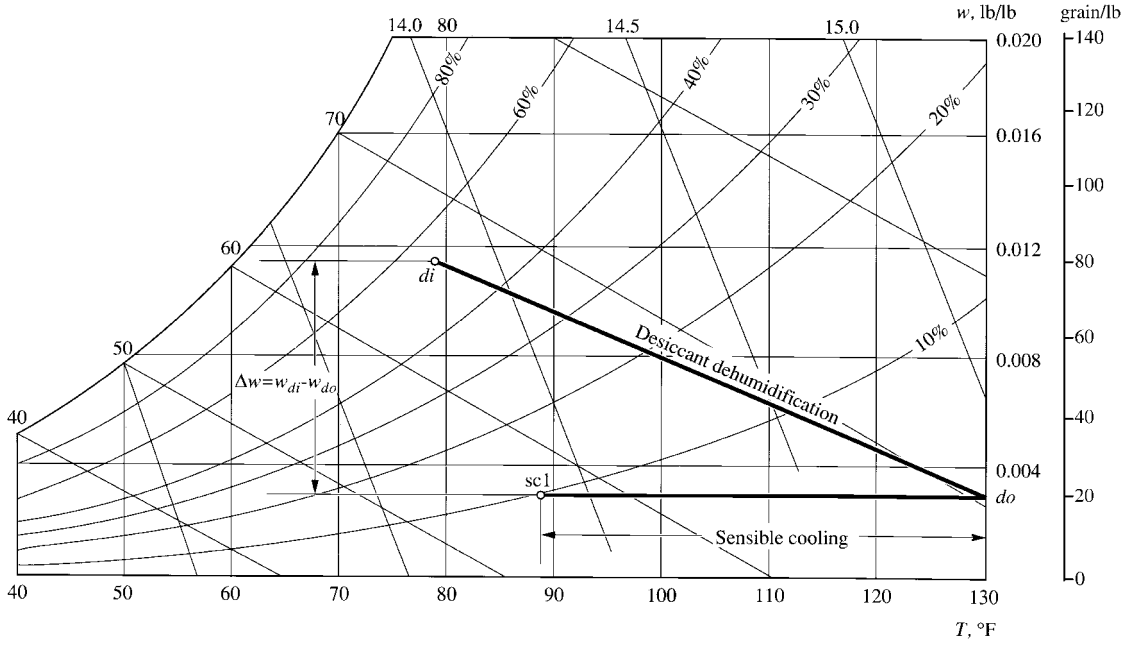


FIGURE 29.3 Desiccant dehumidification and sensible cooling.

After desiccant dehumidifcation, the extremely high-temperature air is usually sensibly cooled in two stages in order to save energy. This process is represented by line *do-sc*1 on the psychrometric chart in Fig. 29.3. The first stage uses an indirect evaporative cooling process, or is cooled by an outdoor airstream in an air-to-air heat exchanger, or is cooled by the evaporator end of a heat-pipe heat exchanger. In an indirect evaporative cooling process, the hot air is sensibly cooled by another evaporatively cooled airstream without direct contact.

The first-stage sensible cooling load Q_{sc1}, in Btu/h (W), can be calculated as

$$Q_{sc1} = 60\dot{V}_o\rho_o c_{pa}(T_{do} - T_{sc1}) \tag{29.23}$$

where T_{do}, T_{sc1} = temperature of dehumidified air entering and leaving indirect evaporative cooler, air-to-air heat exchanger, or heat pipe heat exchanger, °F (°C). The end state of first-stage sensible cooling *sc*1 can be determined according to local outdoor dry- and wet-bulb temperatures to use fully the indirect evaporative cooling to save refrigeration in second-stage sensible cooling.

In the second stage, dehumidified air may be mixed with the recirculated air and then enter a DX coil for further sensible cooling. The second-stage sensible cooling load Q_{sc2}, in Btu/h (W), can be calculated as

$$Q_{sc2} = 60\dot{V}_o\rho_o c_{pa}(T_{sc1} - T_{sc2}) \tag{29.24}$$

where T_{sc2} = temperature of dehumidified air leaving coil, °F (°C).

Desiccant-Based Air Conditioning Systems

In an air conditioning system, the coil load can be divided into sensible load and latent load. For comfort systems in commercial buildings, the latent load varies from 5 to 35 percent of the total coil load. In applications such as supermarkets, it may exceed 50 percent of the total coil load.

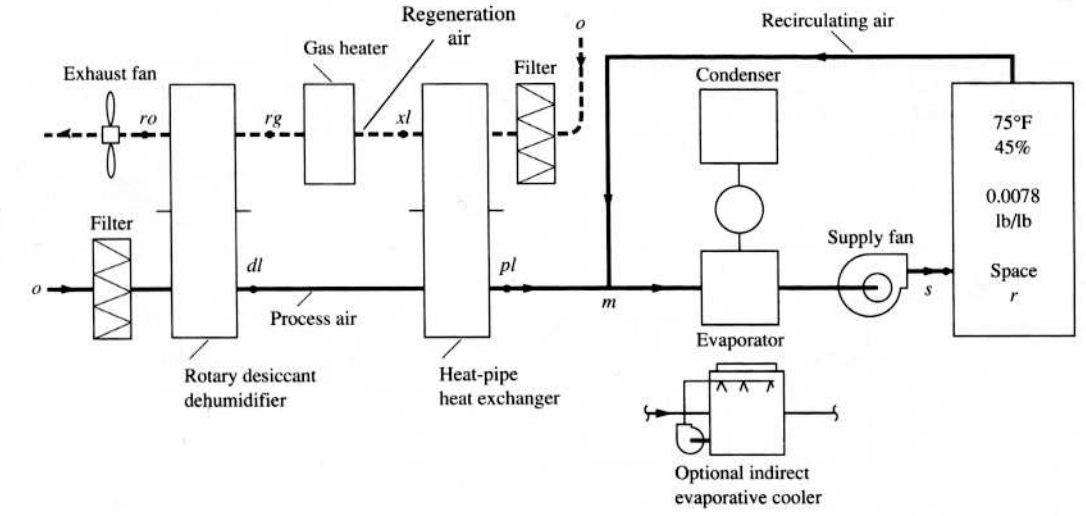
A desiccant-based air conditioning system is a system in which latent cooling is performed by desiccant dehumidification and sensible cooling by evaporative cooling or refrigeration, as shown in Fig. 29.4. Thus, a considerable part of the expensive vapor compression refrigeration is replaced by inexpensive evaporative cooling. There are two airstreams in a desiccant-based air conditioning system: a process airstream and a regenerative airstream. Process air can be all outdoor air or a mixture of outdoor and recirculating air. Process air is also the conditioning air supplied directly to the conditioned space or enclosed manufacturing process, or to the packaged unit or sometimes to the air-handling unit, or terminal for further treatment. A regenerative airstream is a high-temperature airstream used to reactivate the desiccant.

A typical desiccant-based air conditioning system, or simply a desiccant-based system, consists of mainly the following components: rotary desiccant dehumidifier, heat-pipe heat exchanger, indirect evaporative cooler, scroll or reciprocating vapor compression unit with DX coil or water-cooling coil, gas-fired heaters, fans, pumps, filters, controls, outer casing, ducts, and piping. Recently, the heat-pipe heat exchanger has been used instead of a rotary heat exchanger in the earlier desiccant-based systems because a heat-pipe heat exchanger has no cross-contamination or moving parts. Any of these components may be replaced by other components of similar function.

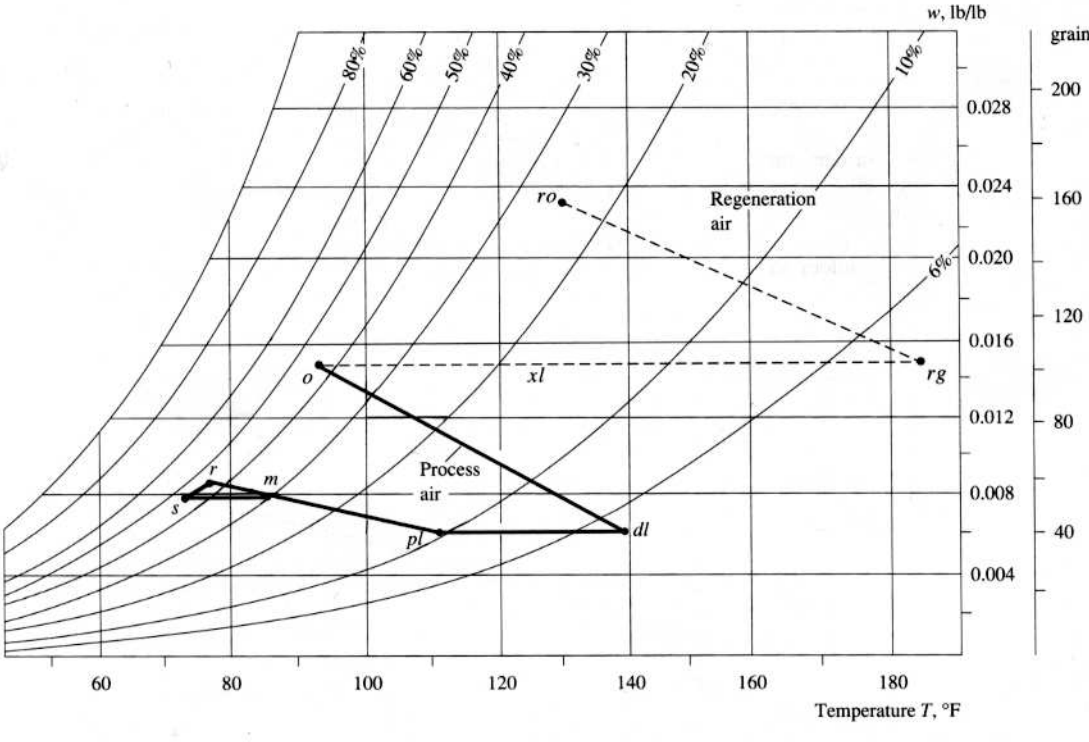
In addition to these components, a desiccant-based system may need a return air system, smoke control systems, and mechanical exhaust systems.

Desiccants

Either solid or liquid desiccant absorbs or releases moisture because of the difference in vapor pressure between the surface of the desiccants p_{des} and the surrounding air p_{sur}. In a desiccant dehumidification process, when $p_{\text{des}} < p_{\text{sur}}$, the desiccant absorbs moisture from the ambient air. In a regenerative process, $p_{\text{des}} > p_{\text{sur}}$, so moisture is released from the desiccant to the surrounding air.



(a)



(b)

FIGURE 29.4 A desiccant-based air conditioning system for a supermarket: (*a*) schematic diagram; (*b*) desiccant-based air conditioning cycle.

Desiccants can be classified as adsorbents, which absorbs moisture without accompanying physical and chemical changes, and absorbents, which absorb moisture accompanied by physical or chemical changes. Three kinds of desiccants are widely used in desiccant-based air conditioning systems: silica gel, lithium chloride, and molecular sieves.

Silica Gels. These are solid desiccants and adsorbents. Structurally, they contain numerous pores and capillaries in which water is condensed and contained. Silica gel has a high capacity to absorb moisture and releases it at a higher temperature. They are low in cost and available in sizes from $^{3}/_{16}$-in. (4.8-mm) beads to powderlike grains. Recently, titanium silica gel is used for its stable property.

Lithium Chloride (LiCl). This is an absorbent. It is in dry form when each LiCl molecule holds two water molecules. If each LiCl molecule holds more than two water molecules, it becomes a liquid and continues to absorb moisture. LiCl has a high capacity to absorb and to hold moisture. Lithium chloride is widely used in rotary wheel dehumidifiers.

Molecular Sieves. These are actually synthetic zeolites, a solid desiccant and an adsorbent in the form of crystalline aluminosilicates produced by a thermal process. Molecular sieves show physical stability and high moisture-releasing capacity at high regenerating temperatures of 248 to 428°F (120 to 220°C), and they are recommended in direct gas-fired applications.

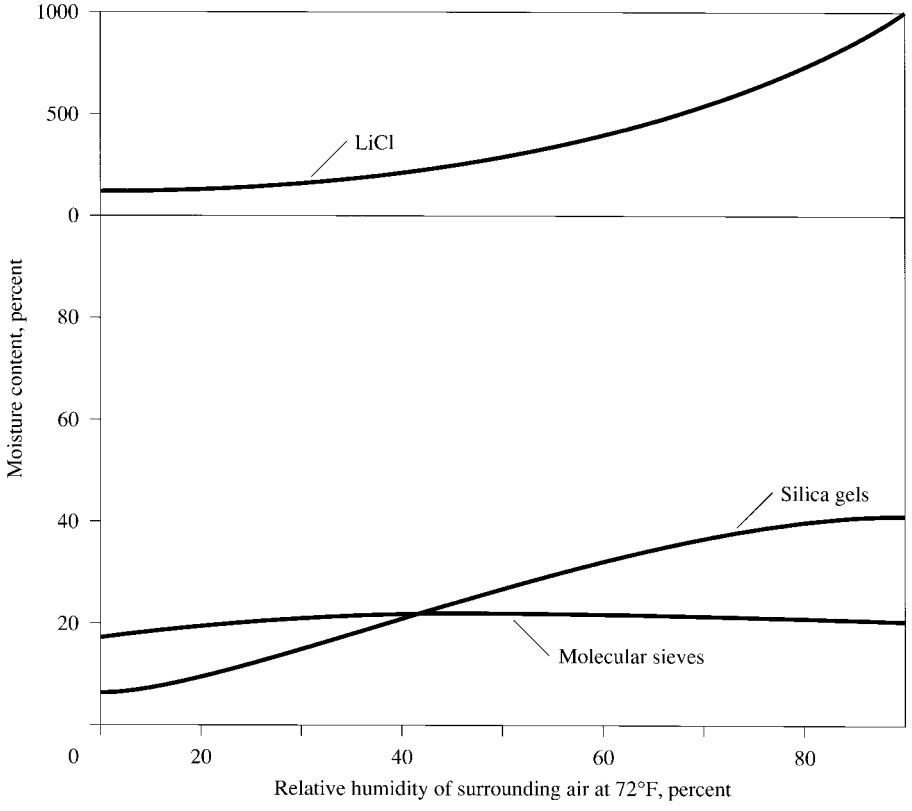


FIGURE 29.5 Sorption isotherms of some desiccants. (*Source: ASHRAE Handbook 1989, Fundamentals. Reprinted with permission.*)

As defined in Sec. 3.4, a sorption isotherm is a constant-temperature curve that indicates the relationship between the moisture content of the desiccant X_{des}, or moisture absorbed as a percentage of its dry mass, and the relative humidity of the surrounding air φ_{sur}. Figure 29.5 shows the sorption isotherms of silica gels, lithium chloride, and molecular sieves. LiCl has a much higher water-holding capacity than silica gels and molecular sieves.

Rotary Desiccant Dehumidifiers

A rotary desiccant dehumidifier is a rotary wheel that removes moisture from the airstream flowing through it. A rotary desiccant dehumidifier has similar construction and operating characteristics to the rotary total heat exchanger, as discussed in Sec. 12.5. Both are rotary wheels. The main differences between a rotary desiccant dehumidfier and a rotary total heat exchanger or rotary heat exchanger are as follows:

- The purpose of using a rotary desiccant dehumidifier is to remove moisture from an airstream, while the purpose of using a rotary total heat exchanger is to transfer heat energy from one airstream to another airstream.
- In a rotary desiccant dehumidifier, there is a process airstream and a regeneration airstream. The regeneration airstream has a very high temperature, whereas in a rotary total heat exchanger, there is an outdoor airstream and an exhaust airstream at indoor temperature.
- A rotary desiccant dehumidifier rotates at a speed of 6 and 10 r/h (revolutions per hour), whereas a rotary heat exchanger rotates at a speed of 10 to 25 r/min.

29.10 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR A SUPERMARKET

Loads in Supermarkets

A desiccant-based air conditioning system is operated at an open cycle when the process air is entirely outdoor air. If the process air at the inlet to the rotary desiccant dehumidifier is a mixture of recirculating air and outdoor air, the desiccant-based system is said to be operated at a closed cycle.

For supermarkets with many frozen food refrigerators, especially single-deck well types and single-deck island food refrigerators with considerable area exposed to indoor air, there are three types of loads:

- *Refrigeration load*. The refrigeration load of the frozen food refrigerator depends mainly on the temperature of the foods in it, the design ambient conditions (the dew point temperature of the conditioned space air), and the frozen surface area exposed to the space air. The higher the space temperature and space relative humidity, and therefore the dew point, the greater the amount of moisture that may condense on the frozen surface, the larger the accompanying latent heat of condensation, and the greater the refrigeration load.
- *Space cooling load*. A considerable portion of the sensible space cooling load is removed by the cooler surfaces of the frozen food refrigerator and the exfiltrated cold airstreams from the frozen refrigerators through their openings, which results in a space cooling load with a heavy latent load that may vary from 50 to 65 percent.
- *Coil load*. The coil load or refrigeration load of the desiccant-based air conditioning system in a packaged unit or sometimes an AHU to maintain a required space condition in the sales area is the sum of the space cooling load, outdoor ventilation air load, and system heat gain. During summer, the outdoor ventilation load is again mainly latent load, and it is affected by the space temperature and relative humidity.

In many supermarkets, 50 percent of the electric energy used annually is consumed by frozen food refrigerators, and another 15 percent is used for the air conditioning system to maintain the required space condition.

System Description

Consider a supermarket with an area of 30,000 ft^2 (2788 m^2) and an outdoor ventilation air requirement of 3000 cfm (1416 L/s). The condition of outdoor air is dry-bulb temperature of 95°F, relative humidity of 40 percent, and humidity ratio of 0.0142 lb/lb (100 gr/lb or 0.0142 kg/kg). The space cooling load is about 83,000 Btu/h (24,320 W). If a desiccant-based air conditioning system with an impregnated lithium chloride dehumidifier is used instead of a conventional vapor compression refrigeration system, it is possible to maintain an indoor space temperature of 75°F (23.9°C) and a relative humidity of 45 percent (0.0085 lb/lb or kg/kg, 60 gs/lb). Meanwhile, the supply air volume flow rate can be reduced from the 1 cfm/ft^2 (5 L/s·m^2) of a conventional system to 0.5 cfm/ft^2 (2.5 L/s·m^2) for a desiccant-based system. Thus, there is a significant reduction in refrigeration load of the frozen food refrigerators because of lower space relative humidity as well as the fan power consumption in the air conditioning system.

Figure 29.4*a* is schematic diagram of a desiccant-based air conditioning system for a 30,000 ft^2 (2788 m^2) supermarket. The process airstream of the desiccant-based air conditioning cycle is indicated on the psychrometric chart by the solid line in Fig. 29.4*b*. The state points of the process air at the exit of various components are as follows:

Outdoor air	*o*
Filter	*o*
Rotary desiccant dehumidifier	*dl*
Heat-pipe heat exchanger	*pl*
Mixing with recirculating air	*m*
Indirect evaporative cooler or refrigeration	*s*
Supply air after supply fan	*s*
Space air	*r*

The state points at the exits of the various components of the regeneration airstream are as follows:

Outdoor air	*o*
Filter	*o*
Heat-pipe heat exchanger	*xl*
Gas heater	*rg*
Rotary desiccant dehumidifier	*ro*
Exhaust fan to atmosphere	*ro*

Space Conditioning Line

For a supermarket of 30,000 ft^2 (2788 m^2), if 50 percent of the total space cooling load Q_{rc} is latent load Q_{rl}, then

$$Q_{rl} = 0.5 \times 83{,}000 = 41{,}500 \text{ Btu/h } (12{,}160 \text{ W})$$

If the supply air is at a temperature of 72.5°F (22.5°C), a relative humidity of 48 percent, and a humidity ratio of 0.0079 lb/lb (55 gr/lb or 0.0079 kg/kg), from Eq. (20.72), the supply air volume

flow rate of the mixture of the process air and the recirculating air is

$$\dot{V}_s = \frac{Q_{rl}}{60\rho_s(w_r - w_s)h_{fg\,32}}$$

$$= \frac{41{,}500}{60 \times 0.075(0.0085 - 0.0079)(1061)} = 14{,}487 \text{ cfm } (6836 \text{ L/s})$$

It is approximately equal to $0.5 \times 30{,}000 = 15{,}000$ cfm (7078 L/s). Similarly, from Eq. (20.69), for a sensible cooling load of about $0.5 \times 83{,}000 = 41{,}500$ Btu/h, the supply volume flow rate to maintain a space temperature of 75°F (23.9°C) is

$$\dot{V}_s = \frac{Q_{rs}}{60\rho_s c_{pa}(T_r - T_s)}$$

$$= \frac{41{,}500}{60 \times 0.075 \times (0.243)(75 - 72.5)} = 15{,}180 \text{ cfm } (7163 \text{ L/s})$$

The larger of these two calculated values should be the design supply volume flow rate.

Operating Parameters in Rotary Desiccant Dehumidifier

Assuming that the humidity ratio of the process air at the exit of the rotary desiccant dehumidifier after dehumidification is 0.006 lb/lb (0.006 kg/kg or 42 gr/lb), the difference during the dehumidification process in the rotary dehumidifier is therefore

$$w_o - w_{dl} = 0.0142 - 0.006 = 0.0082 \text{ lb/lb (kg/kg or 57.4 gr/lb)}$$

For the process airstream, the outdoor air (point *o*), enters the rotary dehumidifier at 95°F (35°C) and a humidity ratio of 0.0142 lb/lb (kg/kg or 100 gr/lb). On the psychrometric chart, draw a line *o-dl* from point *o* with an acute angle of 5° between *o-dl* and the thermodynamic wet-bulb temperature line. This line intersects the 0.006 lb/lb (kg/kg or 57.4 gr/lb) humidity ratio line at point *dl*, which is the state point of process air leaving the rotary desiccant dehumidifier. At point *dl*, temperature $T_{dl} = 139$°F (59.4°C).

For process air at a volume flow rate of 3000 cfm (1416 L/s), the dehumidifying capacity of the rotary dehumidifier is

$$\dot{m}_{\text{deh}} = 3000 \times 0.075 \times 0.0082 \times 60 = 110.7 \text{ lb/h } (50.2 \text{ kg/h})$$

If the volume flow rate of the regeneration air is also 3000 cfm (1416 L/s) and the temperature of the regeneration air entering the rotary desiccant dehumidifier is $T_{rg} = 185$°F (85.0°C) with a humidity ratio of 0.0142 lb/lb (kg/kg or 100 gr/lb), regeneration air leaves the rotary dehumidifier at $T_{ro} = 130$°F (54.4°C), and a humidity ratio of 0.023 lb/lb (kg/kg or 159 gr/lb).

Heat-Pipe Heat Exchanger

Assume that the effectiveness of the heat-pipe heat exchanger is 0.65. If the difference between the specific heat of the process air and regeneration air is ignored, and if the mean density of the process air flowing through the heat-pipe heat exchanger is $\rho_{\text{hp}} = 0.0685$ lb/ft^3 (1.096 kg/m^3) and for the regeneration stream mean density $\rho_{rg} = 0.070$ lb/ft^3(1.12 kg/m^3), then, from Eq. (15.35),

$$\varepsilon = \frac{\rho_p(T_{dl} - T_{pl})}{\rho_{rg}(T_{dl} - T_o)} = 0.65$$

The temperature of the process air after the heat-pipe heat exchanger is

$$T_{pl} = 139 - 0.65 \times \frac{0.070(139 - 95)}{0.0685} = 109.8°\text{F}(43.2°\text{C})$$

The temperature of regeneration air leaving the heat-pipe heat exchanger T_{xl} can then be evaluated as

$$T_{xl} = 95 + \frac{0.0685(139 - 109.8)}{0.070} = 123.5°\text{F}\ (50.9°\text{C})$$

Mixing of Process Air and Recirculating Air

Process air at a volume flow rate of 3000 cfm (1416 L/s) is mixed with recirculating air with a volume flow rate of 12,000 cfm (5663 L/s). The condition of the mixture point *m* can be determined from the psychrometric chart by drawing a line that connects the space air and the process air after the heat-pipe heat exchanger, *r-pl*, so that

$$\frac{rm}{r\text{-}pl} = \frac{3000}{15{,}000} = 0.2$$

From the psychrometric chart, the temperature of the mixture T_m is 81.7°F (27.6°C), and its humidity ratio w_m is 0.0078 lb/lb (kg/kg or 55 gr/lb) with a dew point of 50°F (10°C).

Indirect Evaporative Cooler or Refrigeration

If the desiccant-based air conditioning system is installed in a location in which summer 1 percent cumulative frequency of occurrence of summer outdoor design wet-bulb temperature is lower than 67°F (19.4°C), then an indirect evaporative cooler is recommended. Otherwise, a scroll or reciprocating vapor compressor refrigeration system should be used. For a supply volume flow rate of 15,180 cfm (7163 L/s), the sensible cooling capacity can be calculated as

$$Q_{cs} = 60\dot{V}_s\rho_s c_{ps}(T_m - T_s) = 60 \times 15{,}180 \times 0.075 \times 0.243(81.7 - 72.5)$$

$$= 169{,}313\ \text{Btu/h}\ (49{,}609\ \text{W})$$

The refrigeration system is mainly used for sensible cooling. This is only possible when the evaporating temperature T_{ev} in the DX coil is higher than 50°F (10°C). If T_{ev} is lower than 50°F, a certain degree of dehumidification exists in sensible cooling process *ms*. The required refrigeration load is therefore greater than Q_{cs}.

Gas Heater

The temperature of regeneration air to reactivate the desiccant LiCl is 185°F (85°C). A gas heater is used to heat the air from $T_{xl} = 123.5$ to 185°F (50.9 to 85°C). The heating capacity of the gas heater is

$$Q_h = 60\dot{V}_{rg}\rho_{rg}c_{pa}(T_{rg} - T_{xl})$$

$$= 60 \times 3000 \times 0.243 \times 0.0685(185 - 123.5) = 184{,}266\ \text{Btu/h}\ (53{,}990\ \text{W})$$

Operating Parameters of the Desiccant-Based Air Conditioning Cycle

From the above calculations, the operating parameters of the desiccant-based air conditioning cycle are as follows:

Point	Temperature, °F	Humidity ratio, lb/lb and kg/kg (gr/lb)
o Outdoor air	95	0.0142 (100)
dl After dehumidifier	139	0.006 (42)
pl After heat-pipe heat exchanger	109.8	0.006 (42)
m Mixture	81.7	0.0078 (55)
s Supply air	72.5	0.0078 (55)
r Space air	75, 45% RH	0.0083 (57)
Regeneration air		
xl After heat-pipe heat exchanger	123.5	0.0142 (100)
rg After gas heater	185	0.0142 (100)
ro After dehumidifier	130	0.023 (161)

Part-Load Operation and Controls

When the humidity ratio of the outdoor air drops or the space latent load falls during part-load operation, there are three methods of maintaining the space humidity ratio w_r and the space relative humidity φ_r if the space temperature remains constant:

- Modulation of the gas heating capacity and the temperature of the regeneration air varies the dehumidifying capacity of the rotary desiccant dehumidifier.
- In bypass control, a portion of outdoor air bypasses the rotary desiccant dehumidifier, so that the supply air has a higher humidity ratio.
- The rotational speed of the rotary desiccant dehumidifier can be modulated by using a variable-speed drive. Lower rotating speed of the dehumidifier means a smaller dehumidifying capacity of the rotary desiccant dehumidifier.

The space temperature is controlled by the vapor compression refrigeration system. If the sensible cooling load drops at part-load operation, a cylinder unloader or on/off control in multicompressor system can be used to maintain the space temperature within predetermined limits.

29.11 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR RETAIL STORES

System Description

Spears and Judge (1997) reported a field study of a large retail store, a new 188,000 ft^2 (17,465 m^2) Wal-Mart supercenter in Norfolk, Nebraska. The outdoor ventilation is provided by an outdoor-air unit in a gas-fired desiccant-based air conditioning system, as shown in Fig. 29.6. The sales area is served by two desiccant-based outdoor air units (OAUs) and the grocery area by one OAU. These OAUs supply the required amount of outdoor air to maintain the preset space relative humidity and CO_2 level. There are also rooftop packaged units (RPUs) to provide most of the sensible cooling capacity to offset sensible cooling load in the sales area. In the OAU, a DX coil also provides additional sensible cooling capacity to the space when needed. Each OAU supplies a constant 10,000 cfm (4719 L/s) to the sales area.

Operating Characteristics

During base ventilation operation, 60 percent of the supply air from the OAU, or 6000 cfm (2825 L/s) supplied from each of the OAUs is outdoor air. When the CO_2 sensor senses that the CO_2 level in the sales area exceeds 1000 ppm, the OAU shifts to high ventilation-air operation; all 10,000 cfm (4719 L/s) supplied from the OAU is outdoor air until the CO_2 level is brought back to an acceptable level.

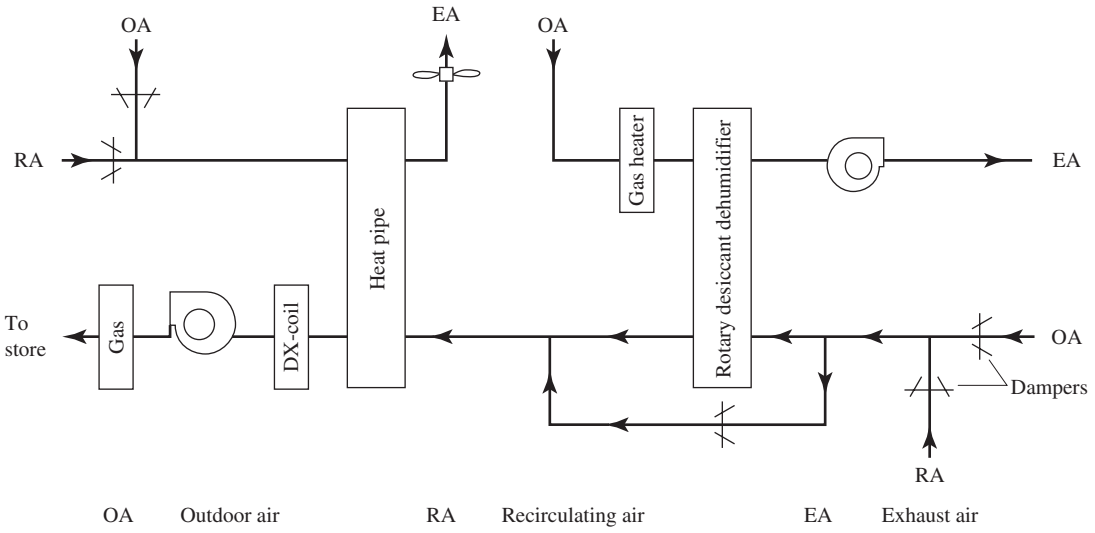


FIGURE 29.6 An outdoor air unit of a desiccant-based air conditioning system for a large retail store in Norfolk, Nebraska.

When the relative humidity sensor senses that the relative humidity of a representative zone exceeds a preset limit and calls for dehumidification, the entire 10,000 cfm (4719 L/s) of supply air in the OAU is dehumidified in the desiccant dehumidifier. When relative humidity drops to or below a preset limit and the dehumidification is not required, the entire 10,000 cfm (4719 L/s) bypasses the rotary desiccant wheel.

Performance

At the Norfolk store, the desiccant-based outdoor ventilation air system provided excellent control of indoor relative humidity and comfort, generally maintaining relative humidity within 5 percent of the 45 percent set point. The space temperature was maintained at 78°F (25.6°C). When the two OAUs were operated in base ventilation mode, the CO_2 level in the store never went above 1000 ppm. During high ventilation air mode, the desiccant wheel removes 0.0065 lb/lb (kg/kg or 42 gr/lb) from the outdoor air. Despite the fact that the ventilation rate had been raised from 0.15 to 0.30 cfm/ft^2 (2.73 to 5.46 m^3/h· m^2) in the Norfolk store, the energy cost increased only 2.6 percent more.

29.12 CASE STUDY: A DESICCANT-BASED AIR CONDITIONING SYSTEM FOR OPERATING ROOMS

Indoor Environment of Operating Rooms

Because the surgical staff are gowning more heavily to avoid the hazard of infectious diseases such as AIDS and at the same time there is a higher lighting and electronic equipment load, today surgeons need 65°F (18.3°C), 45 percent relative humidity, a humidity ratio of 0.0058 lb/lb (kg/kg or

40 gr/lb), and a dew point of 43°F (6.7°C). Using conventional dew point, there is a 22°F (12.2°C) reheat and a dedicated refrigeration system.

System Description

Busby (1996) reported a desiccant-based air conditioning system for a six-room operating suite at Straith Hospital, Southfield, Michigan, as shown in Fig. 29.7. The process airstream is an outdoor airstream of 7600 cfm (3586 L/s) at an inlet with a temperature of 95°F (35°C) with a humidity ratio of 0.0142 lb/lb (kg/kg or 100 gr/lb). It flows through a prefilter of medium efficiency and a water sensible cooling coil; the outdoor air is sensibly cooled to 80°F (26.7°C) and 0.0142 lb/lb (kg/kg or 100 gr/lb). Before entering the rotary desiccant dehumidifier, the process air is divided into two airstreams: a dehumidified airstream and a bypass airstream. At summer design conditions, the bypass air flows through the bypass dampers at 400 cfm (189 L/s). The desiccant used in the rotary dehumidifier is titanium silica gel. After dehumidification in the rotary dehumidifier, the temperature of the process air is raised to 138°F (58.9°C) and the humidity ratio reduced to 0.005 lb/lb (kg/kg or 36 gr/lb). Both the dehumidified and bypass airstreams are extracted by the supply fan, at the outlet of the supply fan, the process air temperature is increased to 140°F (60°C), and its humidity ratio is raised to 0.0057 lb/lb (kg/kg or 40 gr/lb).

The process air then flows through the evaporator end of the heat pipe, and its temperature drops to 95°F (35°C). When the process air flows through a water-cooling coil, its temperature is sensibly

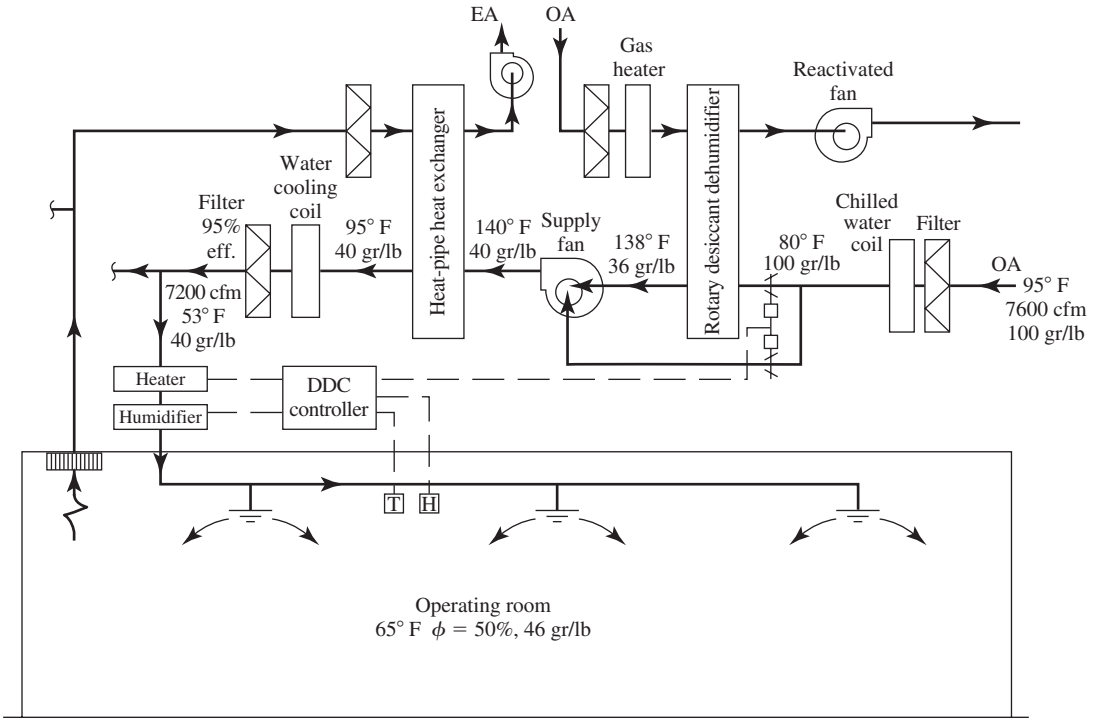


FIGURE 29.7 A desiccant-based air conditioning system for a six-room operating suite at Straith Hospital, Southfield, Michigan.

cooled further from 95 to 53°F (35 to 11.7°C) at a volume flow rate of 7200 cfm (3398 L/s). It is again flowing through a high-efficiency filter of MERV15. At the supply inlet, the supply air temperature is 53°F (11.7°C) with a humidity ratio of 0.0057 lb/lb (kg/kg or 40 gr/lb). The process air is then supplied to the operating room through an electric heater and a humidifier, so that the space temperature can be modulated by a DDC controller to maintain a preset indoor temperature and relative humidity according to sensed values.

There is an exhaust airstream of 7000 cfm (3303 L/s) to absorb the heat rejected from the condenser end of the heat pipe. There is also reactivating outdoor airstream of 250°F (121°C) heated by a gas heater. After it flows through the rotary dehumidifier, this reactivating airstream is exhausted to outdoors.

29.13 APPLICATIONS OF DESICCANT-BASED AIR CONDITIONING SYSTEMS

Comparision between Conventional Vapor Compression Refrigeration System and Desiccant-Based Air Conditioning System

Compared with air conditioning systems using vapor compression refrigeration systems, the desiccant-based air conditioning systems have the following benefits and savings:

- Thermal energy costs are lower. Desiccant-based systems use lower-cost evaporative cooling to replace more expensive electric power cost for vapor compression refrigeration, and use lower-cost gas heating to dry the desiccant instead of more expensive electric power cost to remove latent load by vapor compression refrigeration.
- Desiccant air conditioning system results in drier ductwork to prevent mold and bacterial growth.
- The all outdoor air process airstream provides a beneficial setup for demand-controlled minimum ventilation control. In a desiccant-based air conditioning system, it is more convenient to have a built-in heat recovery if the supply and return ducts are located close together.

The disadvantages of a desiccant-based air conditioning system are that it is more expensive and more complicated and needs more maintenance.

Conditions to Apply Desiccant-Based Air Conditioning Systems

Desiccant-based air conditioning systems are widely used in supermarkets, outdoor air makeup units, surgical operating rooms, and ice rinks. According to field experience, the following are the conditions under which a desiccant-based air conditioned system should be used:

- The dew point temperature of the indoor space temperature is equal to 40°F (4.4°C) or lower.
- There is a need of dehumidification of large amount of outdoor air.
- There is a need of low space relative humidity at lower space temperature.
- There is a low sensible heat ratio SHR_c in cooling and dehumidification process, such as $SHR \leq 0.7$.

ASHRAE/IESNA Standard 90.1-1999 specifies that where humidistatic controls, including dehumidification and humidification, are provided, such controls shall prevent reheating, mixing of heated and cold airstreams, or other simultaneous heating and cooling of the same airstream. Exceptions include:

- Systems serving spaces where humidity levels are required to satisfy process requirements, such as computer rooms, museums, surgical suites, supermarkets, refrigerated warehouses, and ice arenas.

- The system can reduce supply air volume to 50 percent or less of the design airflow rate or the minimal rate specified in ventilation requirements in ASHRAE Standard 62 before simultaneous heating and cooling takes place.
- The individual fan-coil cooling (refrigeration) unit has a design cooling capacity of 40,000 Btu/h (11.7 kW) or less, or a design cooling capacity of 80,000 Btu/h (23.4 kW) or less and can unload to 50 percent capacity before simultaneous heating and cooling takes place.
- 75 percent or more of the energy of reheating or for providing warm air in the mixing process is provided from site recovered heat or from a site solar energy source.
- Where the heat added to the airstream is the result of the use of a desiccant system and 75 percent of the heat added by the desiccant system is removed by a heat exchanger, either before or after the desiccant system, with heat recovery.

System Characteristics

System characteristics of a desiccant-based air conditioning system (DBACS) are listed in Table 29.2.

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CHAPTER 30

AIR CONDITIONING SYSTEMS: CENTRAL SYSTEMS AND CLEAN-ROOM SYSTEMS

30.1 CENTRAL SYSTEMS 30.2
30.2 FLOOR-BY-FLOOR CENTRAL SYSTEMS VERSUS CENTRAL SYSTEMS USING AIR SYSTEMS SERVING MANY FLOORS 30.2
- Size of Air System 30.2
- Separate Air Systems 30.2
- Floor-by-Floor Air System versus Air System Serving Many Floors 30.3

30.3 CONTROLS AND OPERATING CHARACTERISTICS OF CENTRAL SYSTEMS 30.4
- Controls at Part Load for Central Systems 30.4
- Controls of Water, Heating, and Refrigeration Systems 30.4
- Air and Water Temperature Differentials 30.5
- Influence of Inlet Vanes on Small Centrifugal Fans 30.6

30.4 TYPES OF VAV CENTRAL SYSTEMS 30.7
30.5 SINGLE-ZONE VAV CENTRAL SYSTEMS 30.7
- System Description 30.7
- Supply Volume Flow Rate, Coil Load, and Zone Temperature Controls 30.7
- System Characteristics 30.9

30.6 VAV COOLING CENTRAL SYSTEMS, VAV REHEAT CENTRAL SYSTEMS, AND PERIMETER-HEATING VAV CENTRAL SYSTEMS 30.9
- System Description 30.9
- Supply Volume Flow Rate and Coil Load 30.9
- Zone Temperature Controls 30.10
- System Characteristics 30.10

30.7 DUAL-DUCT VAV CENTRAL SYSTEMS 30.10
- System Description 30.10
- System Characteristics 30.11

30.8 FAN-POWERED VAV CENTRAL SYSTEMS 30.11
- System Description 30.11
- Zone Supply Volume Flow Rate and Coil Load 30.11
- Case Study: A Fan-Powered VAV Central System 30.12
- System Characteristics 30.13

30.9 CLEAN-ROOM SYSTEMS 30.14
- System Description 30.14
- Airflow 30.14
- Pressurization 30.16
- Temperature and Relative Humidities 30.16
- System Characteristics 30.16

30.10 CASE STUDY: CLEAN-ROOM SYSTEMS FOR SEMICONDUCTOR INTEGRATED-CIRCUIT FABRICATION 30.16
- Indoor Requirements 30.16
- Energy Use of Components 30.17
- System Description 30.17
- Operating Characteristics 30.18
- Summer Mode Operation 30.19
- Part-Load Operation and Controls 30.19
- Winter Mode Operation and Controls 30.20
- System Pressure 30.21
- Effect of Filter Final-Initial Pressure Drop Difference on System Performance 30.23
- Design Considerations 30.24

REFERENCES 30.24

30.1 CENTRAL SYSTEMS

A central air conditioning system consists of: a central plant in which a boiler and chillers are located, a water system to transport hot and chilled water from the central plant to the AHUs, and an air system often with AHUs to condition the mixture of outdoor and recirculating air and distribute the conditioned supply air to the conditioned space, as discussed in Sec. 1.4. Also, a return air system for energy saving, smoke control systems for multistory buildings according to fire codes, and mechanical exhaust systems may be required.

A central air conditioning system, or simply a central system, is either a single-zone or a multizone air conditioning system. In a central system, air is heated or cooled by the hot or chilled water in coils in air-handling units (AHUs). Conditioned air is then distributed to various control zones through ducts, terminals, and diffusers. Hot water and chilled water are heated by the boilers and cooled by chillers in the central plant.

In a central system, heating and cooling capacities can be accurately modulated by the water flow; air contaminants can be effectively removed by high-efficiency, even HEPA and ULPA, filters. Fan rooms and central plants can be located remotely from sensitive areas. Also there is significantly less field and space HVAC&R maintenance required; therefore, central systems are always used in the quietest space, the cleanest space, the most precision-oriented space, and the most demanding space.

30.2 FLOOR-BY-FLOOR CENTRAL SYSTEMS VERSUS CENTRAL SYSTEMS USING AIR SYSTEMS SERVING MANY FLOORS

Size of Air System

For a central system, the size of the air systems depends mainly on the following:

- Building occupancy and layout
- The smaller the air system, the less the fan system total pressure with lower energy use, the smaller the diameter of the main duct, and the greater the number of the AHUs required, resulting in a higher initial cost
- Advantages to the operation and maintenance of the air system
- Advantages to the compartmentation of fire protection

If the air-handling units are installed indoors, their optimum sizes are usually 15,000 to 25,000 cfm (7079 to 11,798 L/s). Above 25,000 cfm (11,798 L/s), the headroom available in the fan room to install an AHU and ducts is usually not adequate. AHUs having a supply volume flow rate below 10,000 cfm (4719 L/s) are often more expensive per cfm (L/s) volume flow than larger sizes.

To increase the net rentable floor area, weatherproof rooftop AHUs are often used instead of indoor units mounted in the fan room for buildings of only a few stories. In such instances, the size of rooftop AHUs is limited by two things: (1) the products currently available, such as the fact that the largest rooftop AHU has a volume flow rate of 63,000 cfm (29,730 L/s), and (2) the fact that too large an air system always results in a higher system total pressure loss and, therefore, a greater energy cost.

Separate Air Systems

In a control zone with special requirements and a floor area greater than 1000 ft^2 (472 m^2), it may be energy-efficient and cost-effective to use a separate air system. Special requirements include the following:

- Special process temperature and humidity requirements
- Clean rooms or clean space
- Special health care requirements
- Special operating characteristics, such as after-hours operation

Floor-by-Floor Air System versus Air System Serving Many Floors

In a floor-by-floor air system, AHUs are installed at least one for each floor in fan rooms or mechanical rooms. For an air system serving several or many floors, conditioned air is supplied or returned through vertical risers from and to the rooftop units, or from the AHUs installed in the fan rooms at the middle-level mechanical floor of a high-rise building, as shown in Figs. 1.1 and 1.2, or from the basement. A high-rise building is a multistory building of four or more floors. A low-rise building has three floors or less.

Let us compare a floor-by-floor air system and an air system serving many floors, both employing conventional air distribution.

- Normally, a floor-by-floor air system needs a smaller duct system and less fan energy and is more decentralized than an air system serving many floors, which is more centralized. Because the peak loads for all the control zones in a system do not occur simultaneously, a centralized system always has the benefit of a smaller load diversity factor than a decentralized system. Therefore, its total capacity is smaller.
- An air system serving many floors needs supply and return risers, which reduces the amount of available rental space. The air velocity in the return riser is always lower than that in the supply riser. Linford and Taylor (1989) recommended that rule-of-thumb estimate for the riser's area required of 1.3 ft^2 (0.12 m^2) per 1000 ft^2 (472 m^2) of conditioned area served.
- For a VAV system, it is far simpler to use a floor-by-floor air system as the supply, and return air is more easily balanced in each floor when both supply and return volume flow rates are varied at part load.
- Each floor in a floor-by-floor air system is a separate fire compartment, which avoids supply and return duct penetration between floors for fire safety. If local codes require smoke control and purging, the supply and return duct risers for a more centralized system can be easily used as part of the smoke control system during a building fire.
- The fans or compressors in a more centralized system are often located farther away from the occupied space than those in a floor-by-floor air system. The duct-borne noise is often the primary source to be attenuated by sound traps and the longer distances of inner-lined ductwork.
- A floor-by-floor air system has better redundancy, which confines any malfunction and shutdown to the individual floor.
- A floor-by-floor air system has greater flexibility than a more centralized system for future development, after-hours access for overtime workers, easier and more accurate tenant metering, and staged completion and rental of building space.
- A more centralized system always has larger fans and compressors and, therefore, more efficient equipment. It is often able to use high-efficiency airfoil fans and adjustable-frequency variable-speed drives.
- Operating and maintenance of a more centralized system are easier.
- The overall building cost of a more centralized system is often less than that for a floor-by-floor air system for low-rise buildings of two or three floors. For high-rise buildings, a detailed life-cycle cost analysis should be undertaken to determine the optimum choice.

30.3 CONTROLS AND OPERATING CHARACTERISTICS OF CENTRAL SYSTEMS

Controls at Part Load for Central Systems

For a multizone VAV central system, the zone thermal control actions in the air system, water system, and refrigeration system during summer cooling mode part-load operation are as follows:

- In any of the control zones, when a drop in zone temperature T_{rn} is detected by the temperature sensor, the DDC controller adjusts the position of the damper in the VAV box so that the volume flow rate of zone supply air is modulated to match the reduced sensible cooling load in that control zone, to maintain a preset zone temperature.
- As the dampers in the VAV boxes served by the same AHU are closed to smaller openings, the duct static pressure sensor detects the rise in static pressure. The DDC controller then closes the inlet vanes, or varies the speed of the supply fan, until the reduced supply volume flow rate of the AHU is matched by the reduced sensible cooling load of that area served by that AHU.
- When the supply volume flow rate of the AHU is reduced at part-load operation, the discharge air temperature T_{dis} tends to drop to a lower value. As this signal is sensed by the discharge temperature sensor, the DDC controller modulates the two-way valve of the water-cooling coil and reduces the flow rate of the chilled water so as to maintain a nearly constant preset temperature, or a reset T_{dis} at part load.
- As the drop in the coil load, represented by the product of the chilled water flow rate and the temperature difference between the chilled water entering and leaving the coil $T_{wl} - T_{we}$, is sensed by a Btu meter, the DDC controller modulates the inlet vanes of the centrifugal compressor or varies the compressor speed to reduce the refrigeration capacity to match the fall in the coil load at the cooling coils, to maintain a nearly constant or reset chilled water temperature leaving the chiller.
- The capacities of the air, water, and refrigeration systems of a central system should be equal or nearly equal to each other so that an equilibrium between the space temperature, discharge air temperature, and the chilled water temperature leaving the chiller can be maintained at a specific part load.
- Another important characteristic of the capacity control of central systems is that the control actions in all air, water, and refrigeration systems, except multiple constant-speed pumps used in building loops, are usually stepless, continuous modulation controls.

Controls in Water, Heating, and Refrigeration Systems

For most VAV central systems discussed in this chapter, the controls for water, heating, and refrigeration systems are similar.

- *Water system*. The most widely used water system is the plant-building loop system. There are four controls for the plant-building loop system: (1) coil discharge air temperature control, (2) chilled water temperature leaving chiller control, (3) staging control, and (4) pressure differential control. Water system controls are discussed in Sec. 7.11.
- *Heating system*. Low-temperature hot water heating system with two-pipe individual loop is widely used in central systems. Its controls include multiple-boilers staging control, hot water leaving boiler temperature reset, zone hot water temperature control, and safety controls. These are discussed in Secs. 8.3 and 8.7.
- *Refrigeration system*. Centrifugal and screw chillers are widely used in central systems. Leaving chilled water temperature control and reset, optimizing operation of multiple chillers, and safety controls are discussed in Secs. 11.11, 13.6, and 13.7.

Air and Water Temperature Differentials

Both air and water temperature differentials exist in a central air conditioning system. For a central system in cooling mode operation, the supply air temperature differential indicates the difference between the space air temperature T_r and the supply air temperature T_s. The discharge air temperature T_{dis} from the AHU after the draw-through supply fan is about 3°F (1.7°C) lower than T_s because of the supply duct heat gain. The temperature of the conditioned air leaving the cooling coil T_{cc} is about 5°F (2.8°C) lower than T_s as a result of the supply fan heat gain and duct heat gain.

The implications of a greater supply air temperature differential $T_r - T_s$, with respect to maintaining a specified space temperature and relative humidity, include the following:

- A lower supply volume flow rate $\dot{V}_s$ and, therefore, less investment for ducts, VAV boxes, flexible ducts, and diffusers
- A lower conditioned air off-coil temperature T_{cc}
- A greater risk of surface condensation due to a lower T_s
- A lower fan energy use because of the lower $\dot{V}_s$
- A higher compressor power input because of the lower T_{cc}.

The fan energy saving is often greater than the increase in compressor energy input when $T_r - T_s$ is increased.

In current practice, the supply air temperature differential is divided into two categories:

Conventional Air Distribution. This category has the following operating parameters:

- A supply air temperature differential $T_r - T_s$ of 15 to 24°F (8.3 to 13.3°C)
- A supply air temperature of 52 to 58°F (11.1 to 14.4°C), typically 55°F (12.8°C)
- A lowest temperature of chilled water leaving the evaporator as low as 37°F (2.8°C) because of the improvements in freeze protection control in evaporators
- A space temperature of 75 to 78°F (23.9 to 25.6°C) and a space relative humidity between 45 and 50 percent

Neither ice storage systems nor glycol is used in this category.

Cold Air Distribution. This category includes the following operating features:

- A supply air temperature differential $T_r - T_s$ of 30 to 36°F (16.7 to 20.0°C)
- A supply air temperature of 42 to 47°F (5.6 to 8.3°C), typically 44°F (6.7°C)
- A supply of water to the water coil at a temperature of 34 to 38°F (1.1 to 3.3°C)
- A space temperature of 75 to 78°F (23.9 to 25.5°C) and a relative humidity between 35 and 45 percent

Cold air distribution is always used with an ice storage system as the brine melts the ice and is thus cooled to 34°F (1.1°C). Fan-powered units are often used to blend the cold primary air at 44°F (6.7°C) with the plenum air to produce a supply temperature of 55°F (12.8°C). Ethylene glycol or propylene glycol mixed with water, or brine, is needed for freeze protection. Adequate insulation must be provided for ducts, terminals, and diffusers to prevent surface condensation.

As described in Sec. 7.1, under currently accepted procedures, the chilled water temperature differential, i.e., the difference between the chilled water temperatures entering and leaving the evaporator $T_{ee} - T_{el}$, is between 10 and 18°F (5.6 and 10.0°C) for water systems of the plant-building loop. A large water temperature differential saves pump power and reduces the pipe size but requires a lower T_{el}. For a variable-flow building water loop, although the chilled water temperature return from the coils T_{wl} may be different from T_{ee} at part load, T_{ee} is nearly equal to T_{wl} at design

load. If glycol is not blended with the chilled water, the lowest chilled water temperature should not be lower than 37°F (2.8°C) to protect against freezing.

Influence of Inlet Vanes on Small Centrifugal Fans

Inlet vanes mounted at the inlet of the centrifugal fan block a certain percentage of the air passage. For a small centrifugal fan, these vanes have a considerable effect on fan performance compared with similar types and sizes of fans without inlet vanes, even when the inlet vanes are fully open.

- The ratio of the inlet diameter D_1 to the impeller diameter D_2, D_1/D_2, is different for forward-curved and backward-curved centrifugal fans. For forward-curved fans, D_1/D_2 varies from 0.8 to 0.9, and for backward fans, it varies from 0.65 to 0.8.
- The percentage of blocked air passage for small centrifugal fans may be between 15 and 25 percent.
- Although the blocked area may be only 15 percent of the total air passage, the eddies and turbulences after the inlet vanes, even if they are fully opened, result in a greater energy loss.

The following is a comparison of fan performance between backward-curved and forward-curved centrifugal fans with inlet vanes when they are fully opened and fans without inlet vanes. These data are taken from the manufacturer's catalog of vertical modular AHUs (1990).

	Without inlet vanes			With inlet vanes		
Blade	BC	BC	FC	BC	BC	FC
D_2, in.	20	22.25	15	20	22.25	15
$\dot{V}$, cfm	10,400	13,000	6000	10,400	13,000	6000
v_f, fpm	2063	2063	2143	2063	2063	2143
Δp_s, in WC	3.75	3.75	3.75	2.25	3.17	3.50
Δp_t, in. WC	4.01	4.01	4.04	2.51	3.43	3.79
bhp, hp	10.93	12.66	5.79	10.38	14.45	5.95
rpm	1841	1645	1296	1839	1646	1291
$p_{t,i}/p_{t,o}$	0.63	0.86	0.94			

BC = backward-inclined or backward-curved centrifugal fan
FC = forward-curved centrifugal fan
$\dot{V}$ = volume flow rate, in cfm (L/s)
v_f = air velocity at fan outlet, fpm (m/s)
Δp_s, Δp_t = fan static pressure and fan total pressure, respectively, in. WC (Pa)
bhp = brake horsepower, hp (kW)
rpm = revolutions per minute
$\Delta p_{t,i}$, $\Delta p_{t,o}$ = fan total pressure with and without inlet vanes, respectively, in. WC (Pa)

If the ratio D_1/D_2 for a backward-curved centrifugal fan of impeller diameter $D_2 = 20$ in. (508 mm) is taken as 0.75 and the percentage of blocked area at the inlet is 20 percent, the velocity at the fan inlet when the inlet vanes are fully opened is

$$v_{\text{inlet}} = \frac{\dot{V}}{A_{\text{free}}} = \frac{10{,}400}{0.8\pi[(20 \times 0.75)/(12 \times 2)]^2}$$

$$= 10{,}590 \text{ fpm } (53 \text{ m/s})$$

For such an extremely high velocity, the ratio of fan total pressure of this BC centrifugal fan of 20-in. (508-mm) impeller diameter when the inlet vanes are fully opened is $p_{t,i}/p_{t,o} = 0.63$. More than one-third of the fan energy output is lost as a result of the installation of inlet vanes. Such a decrease in fan total pressure due to an extremely high velocity v_{inlet} was verified in a field installation

of a VAV system using inlet vane modulation. A fan total pressure of less than 3 in. WC (750 Pa) was available. After the inlet vanes had been dismantled, the fan total pressure increased to more than 4 in. WC (1000 Pa).

For a BC fan with a 22.25-in.- (565-mm-) diameter impeller, the ratio $p_{t,i}/p_{t,o} = 0.86$; and for an FC fan of 15-in.- (381-mm-) diameter impeller, the ratio $p_{t,i}/p_{t,o} = 0.94$.

During a cost comparison between inlet vanes and adjustable-frequency variable-speed drives, the total pressure loss due to the inlet vanes when they are fully opened must be taken into consideration. Therefore, for centrifugal fans in either AHUs or PUs, installation of inlet vanes is not recommended in backward-curved centrifugal fans with impeller diameters smaller than 25 in. (635 mm).

30.4 TYPES OF VAV CENTRAL SYSTEMS

Similar to packaged systems, VAV central systems can be subdivided into the following air conditioning systems according to their configuration and operating characteristics:

- Single-zone VAV central system (SZVAVCS)
- VAV cooling central system (VAVCCS)
- VAV reheat central system (VAVRCS)
- Perimeter-heating VAV central system (PHVAVCS)
- Dual-duct VAV central system (DDVAVCS)
- Fan-powered VAV central system (FPVAVCS)

The dual-duct VAV central system is the only system with a dual-duct VAV air system. Regarding the operating characteristics of the air system, the difference between a VAV central system and a VAV packaged system is mainly due to the fact that water heating and cooling coils in an AHU are used to heat and to cool the supply air in the central system instead of gas or electric heater and DX coils in a PU to heat and cool the supply air in packaged systems. Their differences are discussed in detail in Sec. 29.1.

30.5 SINGLE-ZONE VAV CENTRAL SYSTEMS

System Description

A single-zone variable-air-volume central system is an air conditioning system that has a central plant and water systems to supply hot and chilled water to the water heating and cooling coils in AHUs to heat and cool the supply air, and distribution to the conditioned space through ducts and diffusers, as shown in Fig. 21.1. A relief or return fan is employed to extract the recirculating air to the AHU, or it is exhausted outdoors during the air economizer cycle. The zone air is maintained at a preset temperature or relative humidity by modulating the water flow rate to the coils by means of a sensor and a DDC controller. Hot water is heated in the boiler, and the chilled water is cooled in the chiller in a central plant and transported to the AHU through hot and chilled water systems. Single-zone VAV systems are widely used in arenas, indoor stadiums, airport terminals, and many industrial applications.

Supply Volume Flow Rate, Coil Load, and Zone Temperature Controls

The supply volume flow rate $\dot{V}_s$, in cfm [m/(60s)], of a single-zone VAV central system based on the zone sensible cooling load at summer design conditions can be calculated from Eq. (29.1); and

the supply volume flow rate based on the zone heating load can be calculated from Eq. (29.2). Usually, the supply volume flow rate required to offset the zone sensible cooling load is the greater of the two and is the adopted supply volume flow rate for the single-zone VAV central system.

The cooling coil load Q_{cc}, in Btu/h (W), can be calculated from Eq. (29.3), and the heating coil load Q_{ch}, in Btu/h (W), can be calculated from Eq. (29.4).

Zone temperature controls for a single-zone VAV central system are similar to those discussed in Sec. 21.2.

TABLE 30.1 System Characteristics of VAV Central Systems

	SZVAVCS	VAVCCS/VAVRCS, PHVAVCS	DDVAVCS
Zone thermal and sound control			
Control zone	Single-zone	Multizone	Multizone
Control methods	DDC, electric	DDC	DDC
Control modes	PI, PID	PI, PID	PI, PID
Heating-cooling modes changeover	Automatic	Automatic	Automatic
Humidity control	Optional	Optional	Optional
Sound control	NC 25–40	NC 25–40	NC 25–40
Indoor air quality			
Minimum ventilation control	DCV or MPC	DCV or MPC	DCV or MPC
Filter efficiency	Medium or high	Medium or high	Medium or high
Air system			
Types	VAV, air mixing	VAV, air mixing	VAV, air mixing, dual supply duct
Supply fan (SF)	Forward, airfoil, centrifugal	Forward, airfoil, centrifugal	Single- or dual-fan, forward, airfoil, centrifugal
Supply fan total pressure	4–5 in. WC	4.5–5.5 in. WC	4.5–6 in. WC
Combined fan-motor-drive efficiency	50%	55%	55%
Relief/return fan	Centrifugal or axial	Centrifugal or axial	Centrifugal or axial
Relief/return fan total pressure	Relief 0.6 in. WC, return 0.5–1 in. WC	Relief 0.6 in. WC, return 0.5–1 in. WC	Relief 0.6 in. WC, return 0.5–1.5 in. WC
Combined fan-motor-drive efficiency	35%	35–40%	35–40%
Air economizer	Fixed or differential dry-bulb or enthalpy	Fixed or differential dry-bulb or enthalpy	Fixed or differential dry-bulb or enthalpy
Cooling systems			
Refrigeration compressor	Centrifugal or screw	Centrifugal or screw	Centrifugal or screw
Refrigerants	HCFC-123, HCFC-22, HFC-134a	HCFC-123, HCFC-22, HFC-134a	HCFC-123, HCFC-22 HFC-134a
Evaporator	Flooded—liquid cooler	Flooded—liquid cooler	Flooded—liquid cooler
Condenser	Water or evaporatively cooled	Water or evaporatively cooled	Water or evaporatively cooled
Energy performance, compressor	0.5–0.7 kW/ton	0.5–0.7 kW/ton	0.5–0.7 kW/ton
Heating systems			
Type	Hot water coil, electric heater	Hot water coil, electric heater PHVAVCS low-temperature hot water baseboard heater	Hot water coil, electric heater
Boiler AFUE	Conventional 78%, condensing 93%	Conventional 78% condensing 93%	Conventional 78% condensing 93%
Maintenance			
Fault detection and diagnostics	AHU, chiller	AHU, chiller	AHU, chiller

System Characterisics

System characteristics of a single-zone VAV central system (SZVAVCS) are listed in Table 30.1.

30.6 VAV COOLING CENTRAL SYSTEMS, VAV REHEAT CENTRAL SYSTEMS, AND PERIMETER-HEATING VAV CENTRAL SYSTEMS

System Description

A VAV cooling central system is a multizone system that has a central plant and chilled water systems to supply chilled water to the AHUs and water-cooling coils to cool and dehumidify the supply air in the AHUs and distribute it to various control zones through ducts, VAV boxes, diffusers, and controls in the interior zone of a building. It is similar to that shown in Fig. 29.1 except the packaged unit should be replaced by an AHU.

A VAV reheat central system is a multizone system that has a central plant and water systems to supply hot water to the reheating VAV boxes, or an electric heater to heat the air in the reheating coil, and chilled water to the water-cooling coils to cool the supply air in AHUs; it distributes the conditioned air to various control zones in the perimeter zone of a building through ducts, reheating VAV boxes, diffusers, and controls, and to various control zones in the interior zone of a building through ducts, VAV boxes, diffusers, and controls. A VAV reheat central system is similar to that shown in Fig. 29.2 except the rooftop packaged unit should be replaced by an AHU.

A perimeter-heating VAV central system is a multizone system that has a central plant and water systems to supply hot water to the perimeter hot water heating system and chilled water to the cooling coils in the AHUs to condition the supply air and distribute it to various control zones in the perimeter and interior zones through ducts, VAV boxes, diffusers, and controls, as shown in Fig. 21.6. In the perimeter zone, winter heating is provided by a low-temperature hot water heating system, such as basebord finned-tube heaters. During winter heating, the VAV boxes supply cold primary air at minimum setting to various control zones in the perimeter zone for the required minimum outdoor ventilation air.

Supply Volume Flow Rate and Coil Load

For VAV cooling central systems, VAV reheat central systems, and perimeter-heating VAV central systems, the supply volume flow rate $\dot{V}_{sn}$, in cfm [$m^3/(60\ s)$], of any control zone in the perimeter or interior zone at summer design conditions can be calculated from Eq. (29.12) as

$$\dot{V}_{sn} = \frac{Q_{rsn}}{60\rho_s c_{pa}(T_r - T_{sn})} \tag{30.1}$$

The cooling coil load of the AHU that has a supply volume flow rate of $\dot{V}_s$ can be calculated from Eq. (29.15) as

$$Q_{cc} = \dot{V}_s \rho_s (h_m - h_{cc}) \tag{30.2}$$

In a VAV reheat central system, for any control zone in the perimeter zone, the reheating coil load of a reheating VAV box can be calculated from Eq. (29.16) as

$$\begin{aligned} Q_{chn} &= Q_{rhxn} + Q_{venxn} \\ &= Q_{rhxn} + 60\dot{V}_{sxn}\rho_s c_{pa}(T_{rxn} - T_{enn}) \end{aligned} \tag{30.3}$$

Zone Temperature Controls

In a VAV cooling central system, as the temperature sensor detects that the zone temperature T_{rn} is falling below a preset value (set point) in the interior zone during part-load operation, a DDC controller closes the damper opening in the VAV box until the minimum setting, such as 30 percent, is reached. When the VAV box damper is at minimum setting, if the zone temperature T_{rn} drops 2 or 3°F (1.1 or 1.7°C) lower than the set point, the DDC controller resets the discharge air temperature of the AHU T_{dis} by 2 or 3°F (1.1 to 1.7°C) higher in order to maintain a higher zone temperature T_{rn}.

In a VAV reheat central system, VAV boxes are installed in each of the control zones in the interior zone.

- Since in the interior zone cold air is often still required to offset the zone cooling load in the winter season, zone temperature control is the same as that in a VAV cooling central system. For each control zone in the perimeter zone, a reheating VAV box is installed.
- As discussed in Sec. 21.3, when the zone temperature drops within the range $72.0 < T_{rxn} \leq 75.0$°F ($22.2 < T_{rxn} \leq 23.9$°C), zone temperature control is in deadband mode. The damper in the reheating VAV box is at 30 percent minimum setting, and the reheating coil is deenergized.
- When the zone temperature $T_{rxn} \leq 72$°F (22.2°C), zone temperature control is in heating mode. The DDC controller opens the two-way valve of the reheating coil and modulates its entering water mass flow rate in a reverse-acting mode to maintain a preset zone temperature. The cold primary air is still at 30 percent minimum setting to provide outdoor ventilation air to the control zone.

In a perimeter-heating VAV central system, the zone temperature controls of the control zones in the interior zone during the cooling mode design load and part-load operations are similar to those in the VAV reheat central system except that a VAV box is installed in each of the control zones in the perimeter zone.

The operating mode of the zone temperature control in control zones of the perimeter zone is often automatically changed over from cooling mode to heating mode and vice versa. There is also a deadband mode between the cooling and heating modes. The width of the deadband mode is often limited to 1 to 3°F (0.6 to 1.7°C).

When the zone temperature of a control zone in the perimeter zone $T_{rxn} \leq 72$°F (22.2°C), zone temperature control of the perimeter-heating VAV central system is in heating mode operation. In heating mode:

- A DDC controller actuates the reverse action relay.
- The cold primary air supplied from the VAV box in the perimeter zone is still at 30 percent minimum setting.
- The hot water temperature leaving the boiler is reset according to the outdoor air temperature.
- The controller opens and modulates the mass flow rate of hot water entering the finned-tube baseboard heater in the perimeter zone to maintain a preset temperature of 72°F (22.2°C).

System Characteristics

System characteristics for VAV cooling central systems, VAV reheat central systems, and perimeter-heating VAV central systems are listed in Table 30.1.

30.7 DUAL-DUCT VAV CENTRAL SYSTEMS

System Description

A dual-duct VAV central system is a multizone system that has a central plant and water systems to supply hot and chilled water to the water heating and cooling coils in the AHUs to condition

the supply air; it distributes to various control zones in the perimeter zone by using a warm air duct and a cold air duct through mixing VAV boxes, diffusers, and controls, and to various control zones in the interior zone using only a cold duct through VAV boxes, diffusers, and controls.

Only central systems have field-built dual-duct VAV air systems. The number of supply fans, operating characteristics of mixing VAV boxes, zone temperature control and sequence of operations, discharge air temperature control, and zone supply volume flow rate of a dual-duct VAV central system are the same as those discussed in Sec. 21.4. The dual-fan dual-duct (cold and warm duct has its own supply fan) VAV central system allows the use of different ratio of outdoor air to supply air, simplifies its control and is more energy-efficient than a single-supply-fan dual-duct VAV central system. A dual-fan dual-duct VAV central system is shown in Fig. 21.9.

The supply volume flow rate, cooling coil load, and the heating coil load can be calculated in the same way as discussed in Section 21.4.

System Characteristics

System characteristics of a dual-duct VAV central system are listed in Table 30.1.

30.8 FAN-POWERED VAV CENTRAL SYSTEMS

System Description

A fan-powered VAV central system is a multizone system that has a central plant and water systems to supply hot and chilled water to the AHUs and VAV boxes, water-cooling coils in the AHUs, and reheating coils in the fan-powered VAV boxes to cool and heat the supply air and distribute it to various control zones in the perimeter zone through ducts, fan-powered VAV boxes, diffusers, and controls, and to various control zones in the interior zones through ducts, VAV boxes, diffusers, and controls. There are two types of fan-powered VAV box: parallel fan-powered VAV box and series fan-powered VAV box, as discussed in Sec. 21.5. Parallel fan-powered VAV boxes are more widely used because they are more energy-efficient. A fan-powered VAV central system with parallel fan-powered VAV boxes is similar to that shown in Fig. 21.13.

The fan energy use, fan characteristics, zone temperature control, and design considerations of a fan-powered VAV central system are similar to those discussed in Sec. 21.5.

Zone Supply Volume Flow Rate and Coil Load

Zone supply volume flow rates of a fan-powered VAV central system are the same as those in a fan-powered VAV packaged system, as discussed in Secs. 21.5 and 29.7.

For conventional air distribution in the perimeter zone, zone peak supply volume flow rate $\dot{V}_{sxn,d}$ can be calculated from Eq. (29.17). For conventional air distribution in interior zone, zone peak supply volume flow rate $\dot{V}_{sin,d}$ can be calculated from Eq. (29.18). For cold air distribution in perimeter zone, zone supply volume flow rate of cold primary air $\dot{V}_{sxn,c}$ can be calculated from Eq. (29.19) For cold air distribution, the volume flow rate of the recirculating plenum air can be calculated from Eq. (29.20) as

$$\dot{V}_{sxn,r} = \dot{V}_{sxn} - \dot{V}_{sxn,c}$$

The reheating coil load in the fan-powered VAV box Q_{chxn} can be calculated from Eq. (29.16). The cooling coil load of the AHU at summer design conditions Q_{cc} can be calculated from Eq. (29.15).

Case Study: A Fan-Powered VAV Central System

A fan-powered VAV central system was designed and constructed for the 34-story office building of the Taipei World Trade Center, Taipei, Taiwan. It has a gross floor space of 1.05 million ft^2 (97,580 m^2) and a total air conditioned floor space of 880,310 ft^2 (81,810 m^2). The peak refrigeration load of the building is about 2530 tons (8890 kW). The local utility rate structure strongly favors the use of the refrigeration system during off-peak hours. This project won first place in the 1991 ASHRAE Technology Awards for HVAC&R system designs for commercial buildings. The entrants were Hsing-Chung Yu and Carl E. Claus.

Refrigeration System. The ice storage system was a partial storage one (ice storage systems are discussed in the next chapter). A separate flooded liquid cooler was connected to the screw compressors to provide direct cooling during on-peak hours as well as during after-hours operation, to offset the significant nighttime load. During direct cooling, the compressors were controlled to operate at a higher suction temperature of 33°F (0.56°C) to conserve energy.

A liquid overfeed system was used instead of direct expansion during the ice-making period because of its higher heat transfer. About 20 percent of the refrigerant coil surface in the ice builders could be saved. During ice melting, chilled water was supplied directly through a closed-loop water circuit to the storage tanks, and the tanks were pressurized. Such an arrangement obviates the use of a heat exchanger and thus avoids the corresponding chilled water temperature rise.

Condensing heat was efficiently rejected through evaporatively cooled condensers.

Air Systems. Because of the lower chilled water temperature from the storage tank during ice melting, cold primary air at 45°F (7.2°C) was introduced at the parallel fan-powered unit. It was mixed with the induced plenum air to produce a supply air of 56°F (13.3°C) at the diffuser. Such a lower primary air temperature had the following consequences:

- Summer space relative humidity dropped to 40 to 45 percent. The thermal comfort of the occupants was greatly improved.
- Cold primary air volume flow rate was reduced to 40 percent compared with conventional air distribution.

Two built-up centralized AHUs were installed in the basement. Each was used to supply 200,000 cfm (94,380 L/s) of primary air from the bottom to the top of the entire building. Compared with the originally proposed conventional system with additional equipment floors at mid-level, this air distribution system saved considerable rental space.

The centralized air system, having floor-by-floor automatic shutoff dampers, was closely matched with the roof-mounted smoke control system. In case of a building fire, the return fan would be stopped and the rooftop exhaust fan would be started to purge the smoke from the fire floor through the opening and closing of dampers connected to the smoke exhaust system. At the same time, air would be supplied to the two floors above the fire floor as well as one floor below to pressurize these floors.

The air system also did the following:

- Provided 15 cfm (7.1 L/s) of outdoor air for each occupant
- Included cartridge-type filters and upstream prefilters with a dust spot efficiency of 85 percent
- Used controllable-pitch axial fans for supply and return air
- Included electric heaters for winter heating

A microprocessor-base DDC system was used for zone temperature; AHU operating parameters; ice-charging, discharging, and direct cooling; and emergency smoke controls.

Cost and Electric Demands. The actual bidding price of the HVAC&R system for this project was approximately $9 million, or $8.57 per ft^2 ($92.2 per m^2) gross area. Compared with the originally proposed conventional air conditioning system using centrifugal chillers, the air system power input was reduced from 1679 to 1136 kW, and the electricity demand dropped from 5960 to 4330 kW. The estimated total annual energy consumption per ft^2 gross area was 33,190 Btu/ft^2·yr (105 kWh/m^2·yr).

System Characteristics

System charateristics of fan-powered VAV central systems are listed in Table 30.2.

TABLE 30.2 System Characteristics for Fan-Powered VAV Central Systems (FPVAVCS) and Clean-Room Systems

	FPVAVCS	CRS
Zone thermal and sound control		
Control zone	Multizone	Single or multizone
Control methods	DDC	DDC, or electric
Control modes	PI or PID	PI or PID
Heating-cooling mode changeover	Automatic	Cooling mode year-round, summer-winter mode changeover, automatic
Sound control	35–45 dBA	NC 50–65
Indoor air quality (IAQ)		
Minimum ventilation control	DCV or MPC	Constant-volume
Filters	Medium to high efficiency	HEPA or ULPA with prefilters
Humidity control	Optional	Dew point control
Humidifiers		Steam, or heating element
Air system		
Types	VAV air mixing	MAU and RAU conditioning and mixing
Supply fan	Forward, or airfoil centrifugal	Centrifugal or vane-axial
Supply fan total pressure	4.5–5.5 in. WC	MAU 4–4.5 in. WC, RAU 2–2.5 in. WC
Combined fan-motor-drive efficiency	55%	MAU 55%, RAU 55%
Relief/return fan	Centrifugal or axial	Centrifugal or axial
Relief/return fan total pressure	Relief 0.6 in. WC, return 0.5–1 in. WC	Return 0.5–1 in. WC
Combined fan-motor-drive efficiency	35–40%	35–40%
Fan-powered VAV box fan	Centrifugal, forward	N/A
Box fan total pressure	0.5 in. WC	
Combined fan-motor-drive efficiency	25%	
Air economizer	Fixed or differential dry-bulb or enthalpy	N/A
Cooling systems		
Refrigeration compressor	Centrifugal or screw	Centrifugal or screw
Refrigerants	HCFC-123, HCFC-22, HFC-134a	HCFC-123, HCFC-22, HFC-134a
Evaporator	Flooded liquid cooler	Flooded liquid cooler
Condenser	Water cooled or evaporatively cooled	Water cooled or evaporatively cooled
Energy performance (compressor)	0.5–0.7 kW/ton	0.5–0.7 kW/ton
Heating system	Hot water coil, or electric heater	Space, year-round cooling, preheating—outdoor air, reheating—part load; hot water, electric heating
Boiler AFUE	Conventional 78%, condensing 93%	Conventional 78%, condensing 93%
Maintenance		
Fault detection and diagnostics	AHU, chiller	AHU, chiller, filters

30.9 CLEAN-ROOM SYSTEMS

System Description

A clean-room or clean space central air conditioning system, or simply a clean-room system (CRS) or clean space system, has a central plant and water systems to supply hot and chilled water to the AHUs, uses HEPA and ULPA filters with prefilters and water coils in the AHU to remove air contaminants, and conditions the air. The conditioned air is supplied to the conditioned space through ducts, terminals, and air distribution devices in order to provide an indoor environment strictly controlled with the required cleanliness, temperature, relative humidity, airflow pattern, pressurization, and noise.

A clean-room system is shown in Fig. 30.1 and has the following characteristics:

- Because of the complexity of the system configuration and the higher requirements in the control of indoor environments, a clean-room system is a custom-built central system with AHUs and water-cooling and heating coils to condition the supply air.
- A clean-room system is required to provide airflow of specific velocity to reduce lateral air contamination. Therefore, a clean-room system is a constant-volume system.
- A clean-room system can be either a single-zone system or a multizone system. In a multizone clean-room system, the zone reheating coil is used to compensate for the variation in zone sensible load to maintain a nearly constant preset zone temperature.
- A clean-room system always has a separate makeup air unit (MAU) to condition the outdoor air, and a recirculating air unit (RAU) recirculates the space air, filters it, cools it, and pressurizes the mixture of outdoor and recirculating air. Such an arrangement minimizes the cross-contamination of airstreams as well as consolidates the filtration of the outdoor air.

Clean rooms and clean spaces are widely used in semiconductor, pharmaceutical, aerospace, and health care industries and facilities.

Airflow

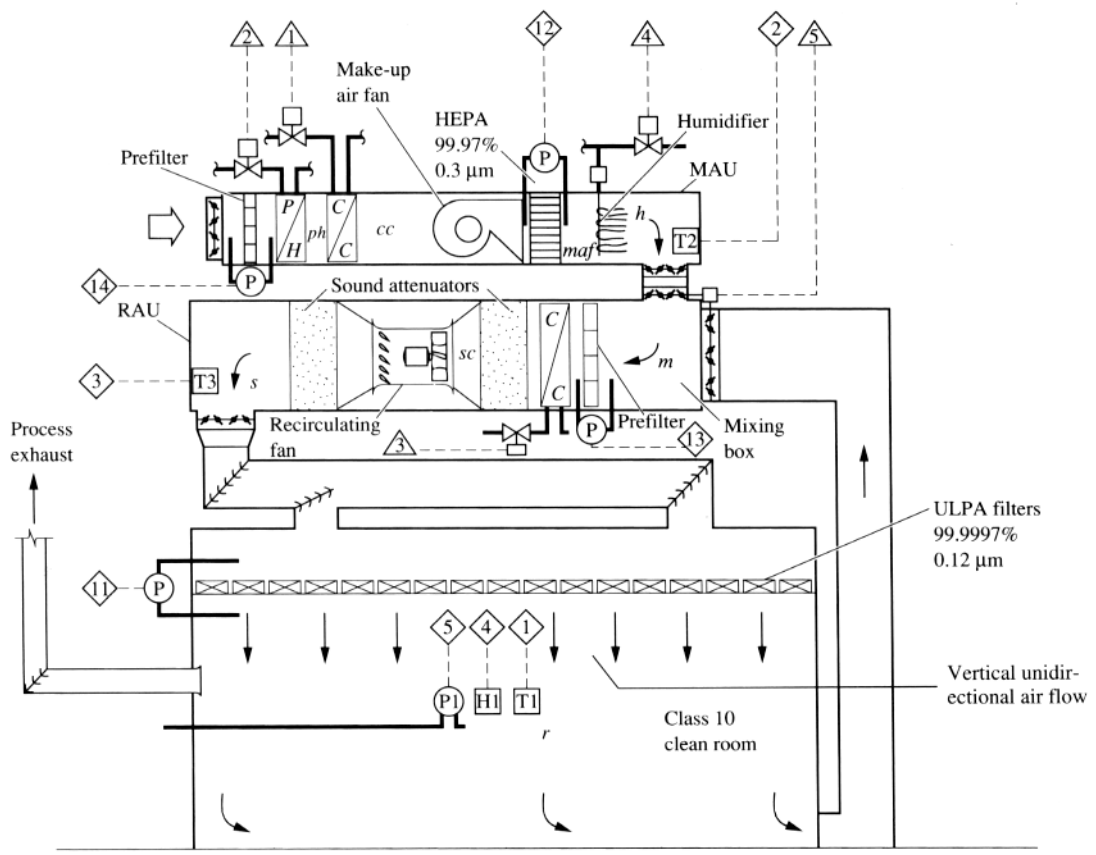
The volume flow rate of the cleaned and conditioned air supplied to the clean room depends on the desirable air velocity that must be provided in the working area of the clean room. As discussed in Sec. 20.17, the supply volume flow rate $\dot{V}_s$, in cfm (L/s), can be calculated from Eq. (20.73) as

$$\dot{V}_s = A_r v_r \tag{30.4}$$

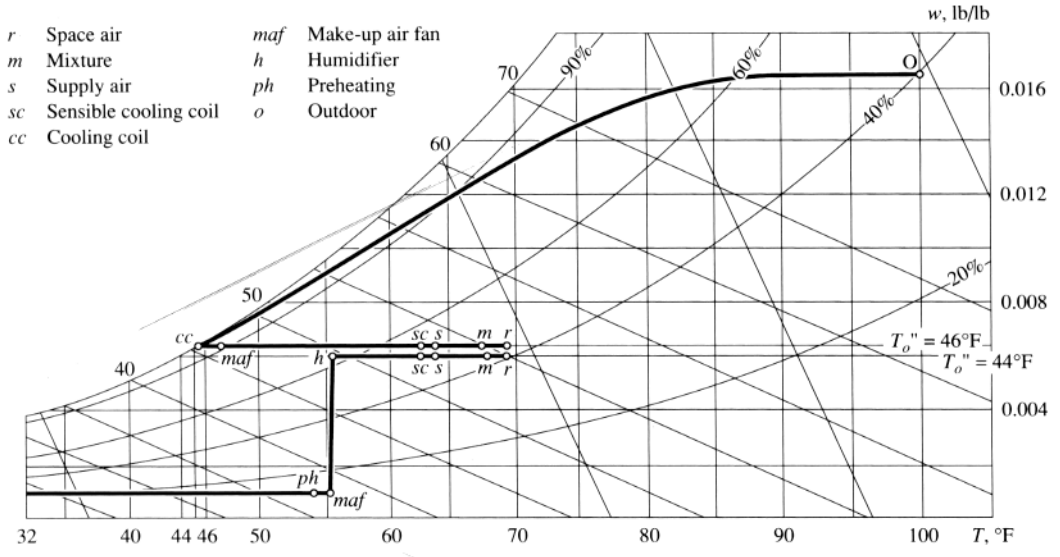
According to *ASHRAE Handbook 1999*, *HVAC Applications*, U.S. Federal Standard 209E does not specify velocity requirements. The 90 fpm (0.45 m/s) figure is still widely used in clean rooms. Current research suggests lower velocity may be possible if the required cleanliness levels can be maintained.

Proper airflow pattern is essential to predict the paths of the airstreams as well as to prevent contaminants from being deposited on critical surfaces in the working area. In clean rooms, there are two types of airflow pattern: unidirectional airflow, as shown in Fig. 30.1*a*, and nonunidirectional airflow. In a unidirectional airflow pattern, airstreams flow through the working area of the clean room in a single-pass, single direction of parallel airstreams. The unidirectional airflow can be subdivided into vertical unidirectional airflow and horizontal unidirectional airflow.

When the ceiling of a clean room is fully covered by HEPA or ULPA filters, the downward airflow produces a unidirectional flow of ultraclean air that covers the working area of the clean room. Baylie and Schultz (1994) reported for clean rooms with ceiling partly covered by HEPA or ULPA filters, a porous membrane that is added beneath the HEPA or ULPA filters forms a small plenum which equalizes the pressure across the face of the ceiling and minimizes the turbulence created by the larger grid required to support the HEPA filters and lighting fixtures. When a membrane ceiling



(a)



(b)

FIGURE 30.1 A clean-room system for a class 10 clean room: (*a*) schematic diagram; (*b*) air conditioning cycle.

is added and air is returned from the bottom inlets of the sidewall, the downward airflow underneath the membrane ceiling is in the form of unidirectional airflow.

For a clean room with a ceiling that is partly covered by HEPA or ULPA filters and without any porous membrane underneath the filters, the downward airflow in the clean room is in the form of nonunidirectional airflow.

Pressurization

Clean rooms and clean spaces always maintain a higher pressure than the surrounding less clean space to minimize the infiltration of air contaminants. The following pressure differentials are often used:

	Pressure diffferential, in. WC (Pa)
Between clean rooms or clean space and nonclean space	0.05 (12.5)
Between clean rooms and less clean rooms	0.02–0.03 (5–7.5)

Pressure control precision is typically between ± 0.01 and ± 0.03 in. WC (± 2.5 and ± 7.5 Pa). For a door or opening between two clean rooms of different cleanliness requirements, a minimum air velocity in the range of 15 to 50 fpm (0.07 to 0.25 m/s) should be maintained at the door (when it is opened) or the other openings.

Temperature and Relative Humidities

Because staff wear heavy gowns in clean rooms with stringent indoor environmental requirements, a temperature between 66 and 68°F (18.9 and 20°C) is to be maintained. In class 10,000 or class 100,000 clean rooms with less restrictive garments, 70 to 72°F (21.1 to 22.2°C) may be satisfactory. A tolerance of ±2°F (1.1°C) is adequate for comfort purposes.

In clean rooms, space relative humidity is usually controlled at 45 ± 5 percent.

System Characteristics

System characteristics of a clean-room system are listed in Table 30.2.

30.10 CASE STUDY: CLEAN-ROOM SYSTEMS FOR SEMICONDUCTOR INTEGRATED-CIRCUIT FABRICATION

Indoor Requirements

The fabrication of semiconductor integrated circuits requires a highly sophisticated combination of advanced technologies. Clean-room central system with vertical unidirectional airflow is one of these technologies in semiconductor wafer fabrication. It must meet stringent air quality requirements in air cleanliness, air temperature, humidity, airflow pattern, pressurization, lighting, noise, and vibration for the sake of successful manufacturing. Contaminated semiconductors result in inferior products. As discussed in Sec. 4.11, clean-room air cleanliness requirements for semiconductor fabrication include classes 1, 10, 100, 1000, and 10,000. Vertical unidirectional airflow with average air velocities from 60 to 90 fpm (0.3 to 0.45 m/s), typically at 90 fpm (0.45 m/s), is widely used.

Manufacturing an integrated circuit involves photolithography, etching, and diffusion processes. These processes may take place over many hours, even days. A stable temperature is extremely important. Because clean-room production personnel wear smocks that fully cover them, clean-room temperatures are controlled from 68 to 72°F (20 to 22.2°C) with tolerances of ±0.1°F (0.056°C), ±0.2°F(0.11°C), ±0.5°F (0.28°C), and 1.0°F (0.56°C). A closer tolerance is often maintained within the manufacturing process itself; e.g., wafer reticle writing by electron beam technology needs ±0.1°F (0.056°C). whereas ±1.0°F (0.56°C) is often used for the open-bay area.

In clean rooms, if the relative humidity is too low, static electricity is created and causes defective products. If the relative humidity is too high, some chemicals may expand and may cause equipment failure. For etching and diffusing areas, the humidity should be maintained at 40 to 45 percent with a tolerance of ±5 percent, whereas in the photolithography area, a relative humidity of 35 to 40 percent with a tolerance of ±2 percent is usually maintained.

The manufacture of metal oxide semiconductors requires large quantities of conditioned outdoor air as makeup air, to replace processing exhaust air and to maintain clean-room pressurization. A clean room is always maintained at a positive pressure to prevent the infiltration of contaminated air from surrounding spaces. For some clean rooms, the process exhaust airflow may be as high as 10 cfm/ft^2 (182 m^3/h· m^2) of floor area. The average process exhaust airflow for semiconductor fabrication may be between 2 and 3 cfm/ft^2 (36 to 54 m^3/h· m^2).

Energy Use of Components

In Naughton (1990a, 1990b), the breakdown of energy use in a typical clean-room central system with vertical unidirectional flow is approximately as follows:

Manufacturing equipment (75 W/ft^2 or 807 W/m^2)	50 percent
Air conditioning (HVAC&R)	40 percent
Electric lights	6 percent
Building envelope and others	4 percent

The chiller and fans each consume 45 percent of the total HVAC&R energy use. Both plant and building pumps use the remaining 10 percent.

Because of extremely high space sensible cooling load and very small space latent load, the sensible heat ratio of the space conditioning line SHR_s can often be taken as 0.99.

The operating cost of HVAC&R is only 5 to 20 percent of the total cost needed to produce an integrated circuit (semiconductor wafer). The air cleanliness, temperature, relative humidity, airflow, and pressurization required for successful fabrication are still extremely important goals of an HVAC&R system design. Because of high utility rates, however, a reduction in the operating cost of the clean-room system also becomes a very influential factor in clean-room design and operation.

System Description

Semiconductor clean rooms are of two types: open-bay design or clean tunnel. The clean tunnel consists of narrow modular clean rooms which may be isolated from one another. Open-bay design includes large open-construction clean rooms. A typical clean-room air conditioning system, or simply clean-room system, for a class 10 clean room of the open-bay configuration is shown in Fig. 30.1. This system consists of the following components:

Recirculating Air Unit (RAU). The function of a recirculating air unit is to recirculate the space air; to filter it, to cool it, to pressurize the mixture of recirculating and conditioned makeup air, and to force the mixture to flow through the ULPA filters and the clean-room working area. An RAU

comprises the following components:

- A mixing box to mix recirculating and conditioned makeup air
- A prefilter with a dust spot efficiency of 30 percent
- A chilled water sensible cooling coil
- A recirculating fan
- Makeup air and recirculating air dampers
- Two sound attenuators, one located immediately before the fan inlet and the other after the fan outlet

Usually an axial fan is used as the recirculating fan because of its higher fan total efficiency (between 75 and 82 percent) and its better operating characteristics. An unhoused centrifugal fan, often called a cabinet fan, with a fan total efficiency of 58 to 63 percent is sometimes used because of its lower sound power level.

The chilled water entering the sensible cooling coil in the RAU is often at a temperature of 50°F (10°C).

Makeup Air Unit (MAU). The function of an MAU is to supply outdoor air to the clean room for process exhaust and pressurization, to condition it, and to control the humidity of the clean room by cooling and dehumidifying, or heating and humidifying, the makeup air. The system components in an MAU include the following:

- An outdoor air damper
- A prefilter with a MERV 8 (dust spot efficiency of 30 percent)
- A preheating coil
- A chilled water cooling coil
- A makeup air centrifugal fan
- A HEPA filter of 99.97 percent DOP efficiency
- A humidifier, most often a steam humidifier
- MAU shutoff damper

Chilled water entering the cooling coil in the MAU for cooling and dehumidifying is at a temperature of 40°F (4.4°C). It is more energy-efficient to have a separate chiller to provide chilled water for an MAU.

ULPA Filters and the Unidirectional Airflow Clean Room. The function of ULPA filters is to provide ultraclean air for the clean room. The pressurized plenum or ducted ULPA filter modules are often used for an even distribution of unidirectional downward air flow.

For class 1, 10, and 100 clean rooms, ULPA filters with a DOP efficiency of 99.9997 percent of 0.12-μm particles and unidirectional airflow are used. For class 1000 through class 100,000, HEPA filters with a DOP efficiency of 99.97 percent of 0.3-μm particles and nonunidirectional airflow may provide satisfactory contamination control.

Unidirectional downward airflow produces a uniform air shower of ultraclean air. Internally generated contaminants will not move laterally against the 90 fpm (0.45 m/s) airflow and will be carried away by predictable parallel airstreams. Recirculating air enters either the bottom side return inlets directly or the perforated raised floor panels. It is then returned to the RAU to mix with makeup air again.

Operating Characteristics

The following temperature and relative humidity are to be maintained in a class 10 clean room:

Room temperature, year-round	69 ± 1.0°F (20.6 ± 0.56°C)
Space humidity, year-round	42.5 ± 2.5 percent

For a class 10 clean room with an area of 1000 ft^2 (472 m^2), a supply volume flow rate of 90,000 cfm (42,470 m^2) is required to provide an air velocity of 90 fpm (0.45 m/s) in order to produce a unidirectional flow in this clean room. The outdoor air intake for process exhaust, space pressurization, and occupants is typically 6000 cfm (2830 L/s).

Summer Mode Operation

In Mandelbaum (1991), the operating modes of clean rooms of classes 1 through 1000 are divided into summer and winter modes. When the dew-point temperature of the outdoor air T_o'' is 46°F (7.8°C) and above, the clean-room system is in summer mode operation.

Let us consider a hot summer day. Outdoor air at a summer design temperature of 100°F (37.8°C) and a wet-bulb temperature of 78°F (25.6°C) enters the MAU, as shown in Fig. 30.1*a* and *b*. It is cooled and dehumidified at the cooling coil to a leaving coil condition of air temperature T_{cc} = 46°F (7.8°C), relative humidity φ_r = 99 percent, and dew-point temperature ≈ 46°F (7.8°C). After the conditioned outdoor air absorbs the fan heat of the MAU, the air enters the RAU at a discharge temperature T_{dis} of 47.5°F (8.6°C) and a relative humidity of 92 percent.

In the RAU, conditioned air from the MAU is mixed with the recirculating air from the clean room at a temperature of 69°F (20.6°C), a relative humidity of 42.5 percent, and a dew-point temperature of 46°F (7.8°C). The ratio of volume flow of recirculating air to makeup air is 12:1. The mixture m enters the sensible cooling coil at temperature T_m = 67°F (19.4°C) and a dew point of 46°F (7.8°C). It is then sensibly cooled to a temperature T_{sc}.

If the maximum space sensible cooling load is 563,000 Btu/h (265,680 L/s) and if the density of supply air ρ_s = 0.078 lb/ft^3 (1.248 kg/m^3), the temperature of supply air T_s can be calculated as

$$T_s = T_r - \frac{Q_{rs}}{60\dot{V}_s\rho_s c_{pa}}$$

$$= 69 - \frac{563{,}000}{60 \times 90{,}000 \times 0.078 \times 0.243} = 63.5°\text{F (17.5°C)} \tag{30.5}$$

Usually the temperature rise due to fan heat in the RAU is 1°F (0.56°C). The temperature of the air leaving the sensible cooling coil is then T_{sc} = 63.5 − 1 = 62.5°F (16.9°C). Its dew-point temperature is still 46°F (7.8°C). Because of the short supply duct, large volume flow, and the fact that the surrounding space is conditioned, the duct heat gain is usually ignored.

Because the SHR$_s$ of the space conditioning line is 0.99, after the air has absorbed the space sensible cooling load and a very small amount of latent load, the space temperature is then maintained at 69°F (20.6°C). The space relative humidity is 42.5 percent, and the dew point is 46°F (7.8°C).

To cool and dehumidify the makeup air at the cooling coil to a leaving dew-point temperature of 46°F (7.8°C), the chilled water entering the cooling coil should be provided at a temperature T_{we} = 40°F (4.4°C) without using glycol for freeze protection. The chilled water leaving the chiller in the plant loop is usually 1°F (0.56°C) lower, or 39°F (3.8°C).

Part-Load Operation and Controls

Two controls maintain the required space temperature at 69°F (20.6°C) and the relative humidity within acceptable limits during summer mode part-load operation: the discharge air temperature control incorporating dew point control of the MAU and the space (zone) air temperature control.

When the dry- and wet-bulb temperatures of the outdoor air T_o and T_o' drop but the dew-point

temperature is still at 46°F (7.8°C) or above, the discharge air temperature sensor T2 detects the fall in T_{dis} and the DDC controller modulates the two-way valve of the chilled water cooling coil in the MAU to reduce its water flow until a 47.5°F (8.6°C) discharge air temperature and a discharge air dew point of 46°F (7.8°C) are maintained.

If the space sensible cooling load falls below the design load, the space temperature drops accordingly. As the space temperature sensor T1 detects such a fall in temperature, the DDC controller modulates the two-way valve of the sensible cooling coil in the RAU, reduces its water flow, and tends to maintain a constant space temperature of $T_r = 69$°F (20.6°C).

In addition to the MAU discharge air temperature and space temperature controls, there are controls for space humidity, space pressurization, filter pressure drop monitoring, and smoke detection.

Because the discharge air dew point from the MAU is 46°F (7.8°C), the space dew point is also 46°F (7.8°C), and the mixture of recirculating and makeup air still has a dew point of 46°F (7.8°C). After sensible cooling in the RAU, the dew point of supply air, point *s*, remains at 46°F (7.8°C). Because the sensible heat ratio of the space conditioning line $\text{SHR}_s = 0.99$, the space relative humidity will be always around 42.5 percent if the space temperature T_r is maintained at 69°F (20.6°C).

For a space like a clean room having only a negligible latent load, supply air dew-point control can always maintain both space temperature and relative humidity within required limits by means of sensible cooling and reheating when the space sensible cooling load is varied.

In Fig. 30.1*a*, there is a pressure sensor P1 in the clean room. This sensor measures the pressure differential Δp_r between the clean room and the surrounding area. This pressure differential is generally maintained at 0.05 in. WC (12.5 Pa) to prevent infiltration of contaminated air.

According to the signal from P1, the DDC controller modulates the opening of the interlocked makeup air and recirculating dampers. When there is an increase in makeup air and a decrease in recirculating air in the mixing box of RAU, the space pressure differential tends to rise. A decrease in makeup air and an increase in recirculating air cause a drop in space pressure differential. The supply of makeup air is balanced by the process exhaust air and the exfiltrated air from door gaps due to the pressure differential.

Pressure sensors are also used for MERV 8 prefilters, HEPA filters, and ULPA filters. If the pressure drop exceeds a predetermined limit, an indicating lamp flashes or an alarm is energized. Both signals call for replacement.

When the smoke detector is energized, a signal is sent to the building fire control system. The fire alarm and the smoke control system are energized accordingly.

Winter Mode Operation and Controls

When the dew point of the outdoor air T_o'' is 44 °F (6.7°C) or below, the clean-room system is in winter mode operation. If $T_o'' \leq 44$°F (6.7°C) and $T_o < 53.5$°F (11.9°C), the outdoor air is preheated at the preheating coil to a temperature $T_{ph} = 53.5$°F (11.9°C). After absorbing the fan heat of the MAU, it enters the RAU at a discharge temperature $T_{\text{dis}} = 55$°F (12.8°C). If $T_o'' \leq 44$°F (6.7°C) and $T_o \geq 53.5$°F (11.9°C), the outdoor air is sensibly cooled to 53.5°F (11.9°C). Again, it is discharged to the RAU at $T_{\text{dis}} = 55$°F (12.8°C) after absorbing fan heat.

If makeup air at a temperature of 55°F (12.8°C) and a dew point of 44°F (6.7°C) is mixed with winter mode recirculating air of temperature 69°F (20.6°C) and 40 percent relative humidity (a dew point of 44°F, or 6.7°C), the resulting mixture will be at a temperature of 67.5°F (19.7°C). The dew point will still be 44°F (6.7°C). If the mixture is cooled at the sensible cooling coil to 62.5°F (16.9°C) according to the signal of the temperature sensor T1, and if it is supplied to the clean room at a temperature of 63.5°F (17.5°C) and a dew point of 44°F (6.7°C) after absorbing the fan heat of RAU, then it will maintain a space temperature $T_r = 69$°F (20.6°C), $\varphi_r = 40$ percent, and a dew point of 44°F (6.7°C) at a maximum space sensible cooling load of 563,000 Btu/h (164,960 W). A lower space sensible cooling load results in a decrease in the space temperature T_r. As the sensible cooling capacity at the cooling coil is modulated and reduced by the DDC controller, the space temperature will still be maintained at 69°F (20.6°C).

If $T_o'' < 44$°F (6.7°C), space relative humidity φ_r may be less than 40 percent. When the space

humidity sensor detects such a shortage, the DDC controller energizes the steam humidifier in the MAU to humidify the makeup air until $\varphi_r = 40$ percent. In winter mode operation, φ_r will always be maintained around 40 percent.

During winter mode operation, the chilled water temperature entering the cooling coil in the MAU is reset from 40 to 45°F (4.4 to 7.2°C) for a more energy-efficient operation. Also, the preheating coil and the cooling coil in the MAU are sequentially controlled; they are not energized simultaneously.

In winter operation, outdoor air is preheated in the MAU, and the mixture of outdoor and recirculating air is again sensibly cooled in the RAU. Such simultaneous heating and cooling are mainly a result of the requirement for steam humidification. It is not appropriate to humidify the air to near saturation. Uneven distribution may cause surface condensation.

During winter mode operation, if the dew point of the outdoor air T_o'' is between 44 and 46°F (6.7 and 7.8°C), the operation mode remains in winter mode. When $T_o'' \geq 46$°F (7.8°C), it will change over to summer mode operation. Similarly, if $44 < T_o'' < 46$°F ($6.7 < T_o'' < 7.8$°C), the operating mode remains in summer mode until $T_o'' \leq 44$°F (6.7°C). Then it changes over to winter mode operation. Such a control strategy prevents hunting between summer and winter mode operation during intermediate seasons.

System Pressure

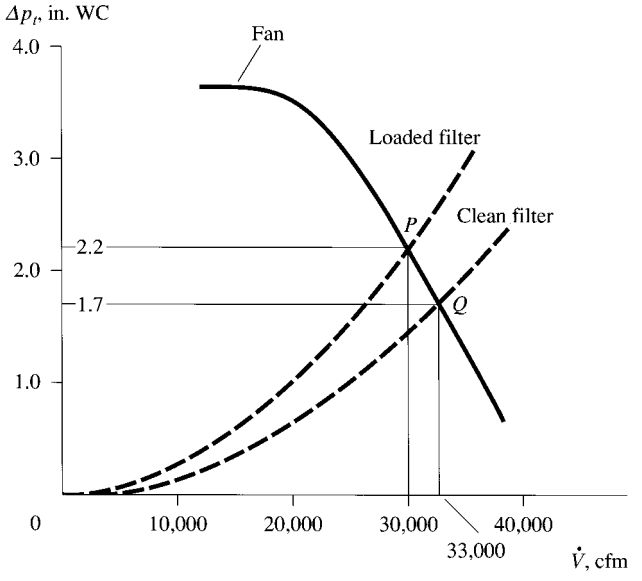
According to Naughton (1990a, 1990b) and Hunt et al. (1990), the system pressure loss of an MAU is about 3.8 to 4 in. WC (950 to 1000 Pa). For an RAU, the system pressure loss is usually between 2.0 and 2.50 in. WC (500 and 625 Pa). A typical breakdown of the fan total pressure of the recirculating air fan in an RAU is as follows:

System components	Pressure drop, in. WC (Pa)
RAU	
Prefilter	0.25 (62)
Cooling coil	0.35 (87)
Inlet sound attenuator	0.10 (25)
Discharge sound attenuator	0.15 (38)
Discharge elbow	0.10 (25)
External pressure drop	
HEPA filters, final	0.80 (200)
Distribution duct work	0.20 (50)
Perforated raised floor panel	0.10 (25)
Return path	0.15 (38)
Total	2.20 (550)

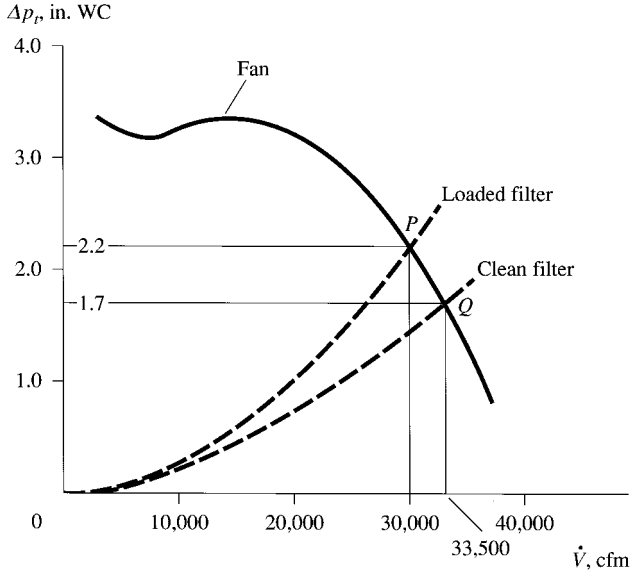
HEPA filters of lower pressure drop, such as a final loaded pressure drop of 0.50 in. WC (125 Pa), are also available. However, they are expensive and require more space. This fact should be carefully analyzed and considered by the production engineer and architect.

To reduce the pressure drop across coils, the most effective method is to reduce their face velocities to 300 to 400 fpm (1.5 to 2 m/s). The result is a larger MAU and RAU. Again, the compromise is between initial and operating costs.

All duct fittings along the airflow must be carefully designed to reduce pressure losses. Square elbows should always be installed with splitter vanes. High-efficiency fans and pumps should be used, along with high-COP and -EER chillers. According to Naughton (1990a, 1990b), the energy use of fans, chillers, and pumps in a clean-room system for a class 1 clean room may be around 50 W/ft^2 (538 W/m^2) of floor area.

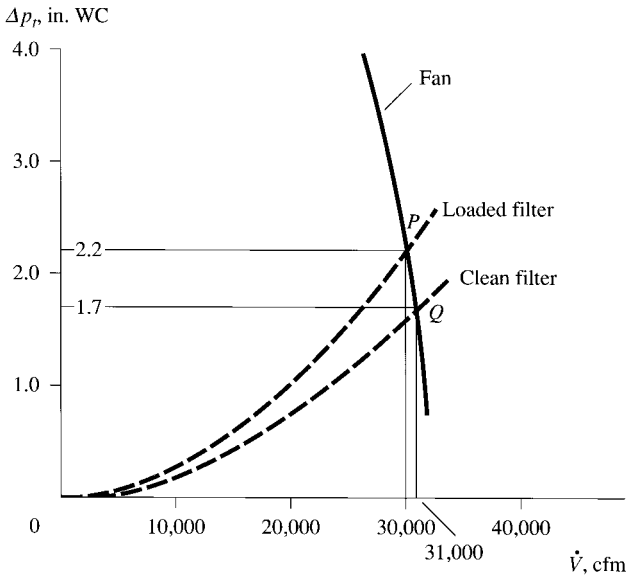


(a)



(b)

FIGURE 30.2 Difference in final and initial pressure drops of filters affects system performance. (*a*) Airfoil blade centrifugal fan; (*b*) forward-curved centrifugal fan; (*c*) vane-axial fan.



(c)

FIGURE 30.2 *(Continued)*

Effect of Filter Final-Initial Pressure Drop Difference on System Performance

During the calculation of the system pressure loss of an air system for the purpose of selecting fans and AHUs, one should use the final pressure drop of the filter $\Delta p_{f,f}$, in in. WC (Pa), to provide the required volume flow rate whenever filters are either clean or loaded with specified capacity. For purposes of energy estimation, the average pressure drop of the filters during the working period $\Delta p_{f,m}$, in in. WG, should be employed instead of final or initial pressure drops.

For most prefilter and ultrahigh-efficiency air filter assemblies used in AHUs, the difference between the final pressure drop and the initial pressure drop of the filter assembly is 0.4 to 0.8 in. WC (100 to 200 Pa). Let us consider an air system whose design volume flow rate $\dot{V}$ is 30,000 cfm (14,160 L/s), and whose system pressure loss when filters are loaded Δp_{sy} is 2.2 in. WC (550 Pa). At $\dot{V} = 30{,}000$ cfm (14,157 L/s), the difference between the final and initial pressure drops is 0.5 in. WC (125 Pa).

The effect of the difference in final and initial pressure drops of a filter on system performance is illustrated in Fig. 30.2. In Fig. 30.2*a*, an airfoil-blade centrifugal fan is used. The fan and AHU are selected according to these criteria: $\dot{V} = 30{,}000$ cfm (14,160 L/s) and $\Delta p_{sy} = 2.2$ in. WC (550 Pa). If the filters are loaded, the operating point is *P*. If the filters are clean, the system pressure drops to 1.7 in. WG (425 Pa). The operating point moves along the selected fan curve to point *Q*, which has $\dot{V} = 33{,}000$ cfm (15,573 L/s) at $\Delta p = 1.7$ in. WC (425 Pa).

In Fig. 30.2*b*, a forward-curved centrifugal fan is used. A forward-curved fan has a slightly flatter fan curve at operating point *Q*, and the system volume flow rate increases to 33,500 cfm (15,810 L/s). For Fig. 30.2*c*, a vane-axial fan is used. A vane-axial fan has a very steep fan curve. Therefore, at point *Q* the system volume flow rate is increased to only about 31,000 cfm (14,630 L/s).

The air system in a clean-room system is always a constant-volume system. The fan power inputs at point *Q* for airfoil blade and vane-axial fans are approximately the same as at point *P*, whereas for forward-curved fans, their fan power input is comparatively greater.

During the selection of AHUs, if the fan performance of the AHU is expressed in volume flow $\dot{V}$ versus fan total pressure Δp_t, the designer should ascertain that the pressure drops of the loaded filters are included in Δp_t, not those of the clean ones.

Design Considerations

The successful construction of a class 1, 10, or 100 clean room depends on the combined effort of architects, mechanical engineers, production engineers, owners, contractors, HVAC&R equipment manufacturers, and many others. To maintain a temperature tolerance of ± 0.1°F (0.056°C) for a manufacturing process, it is important to stabilize all internal loads during the processing period. The process should be surrounded by conditioned space for which the tolerance is ± 0.5°F (0.3°C). Even movement of production personnel toward the processing area may cause radiant heat turbulence and temperature fluctuations.

Energy use per ft^2 (m^2) floor area of a clean-room system for class 1 through class 100 clean rooms is 5 to 10 times higher than that of an air conditioning system for a commercial building. Careful analyses and improvements based on experience with similar projects will provide a satisfactory indoor environment for the manufacturing process and also will reduce the energy use of the clean-room system. Semiconductor products that are being produced now may become obsolete within a few years. As with many other HVAC&R systems, air conditioning system design for clean rooms must incorporate flexibility for change and future development.

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CHAPTER 31
AIR CONDITIONING SYSTEMS: THERMAL STORAGE SYSTEMS

31.1 THERMAL STORAGE SYSTEMS 31.1
System Description 31.1
Building Energy Consumption and Thermal Storage Systems 31.2
Electric Deregulation and the Impact on Thermal Storage Systems 31.2
Benefits and Drawbacks of Thermal Storage Systems 31.2
Full Storage and Partial Storage 31.3
Ice Storage and Chilled Water Storage 31.5
31.2 ICE-ON-COIL, INTERNAL-MELT ICE STORAGE SYSTEMS 31.6
System Description 31.6
Brine and Glycol Solution 31.6
Ice Storage Tank 31.7
Case Study: Operating Modes of Ice-on-Coil Ice Storage System 31.7
System Characteristics 31.9
31.3 ICE-ON-COIL, EXTERNAL-MELT ICE STORAGE SYSTEMS 31.10
System Description 31.10
Ice Builders 31.11
Refrigerant Feed 31.11
Ice-Charging Control 31.11
System Characteristics 31.11
Case-Study: An Ice-on-Coil External-Melt Ice Storage System 31.13
31.4 ENCAPSULATED ICE STORAGE SYSTEMS 31.13
System Description 31.13
Location of Chiller and Storage Tank 31.14
Controls 31.14
Charging and Discharging 31.15
System Characteristics 31.15
31.5 ICE-HARVESTING ICE STORAGE SYSTEMS 31.15
System Description 31.15
Ice Making or Charging 31.16
Chiller Operation 31.17
System Characteristics 31.17
31.6 COMPARISON OF VARIOUS ICE STORAGE SYSTEMS 31.17
31.7 STRATIFIED CHILLED WATER STORAGE SYSTEMS 31.18
System Description 31.18
Basics for Chilled Water Storage 31.18
Storage Tanks 31.19
Stratified Tanks 31.19
Temperature Gradient and Thermocline 31.20
Diffusers 31.20
Charging and Discharging Temperature versus Tank Volume 31.22
System Characteristics 31.23
31.8 CASE STUDY: A STRATIFIED CHILLED WATER STORAGE SYSTEM 31.23
Chilled Water Storage System 31.23
Concentric Double-Octagon Diffusers 31.24
Charging Process 31.26
Discharging Process 31.26
Part-Load Operation 31.27
System Performance 31.28
REFERENCES 31.28

31.1 THERMAL STORAGE SYSTEMS

System Description

A thermal energy storage air conditioning system, or simply a thermal storage system, consists of a central plant, a chilled water or brine system incorporated with a thermal storage system, a hot water system, an air system including AHUs , terminals, return air system, smoke control systems, and

mechanical exhaust systems. In addition, the electric-driven refrigeration compressors in the central plant are operated at off-peak or at off-peak and on-peak hours. Stored ice or chilled water in tanks is used to provide cooling in buildings during on-peak hours when there is a higher electric rate. Hot water from the boilers in the central plant is used to provide heating in winter. Air is conditioned in the AHUs and terminals. The conditioned air is then distributed to various control zones through ducts, terminals, diffusers, and controls.

A thermal storage system is

- Always a central system that uses stored chilled water or brine from the central plant to cool the air
- Often a multizone VAV system
- Often a cooling storage system

Building Energy Consumption and Thermal Storage Systems

Lorsch (1993) analyzed the energy consumption for a 264,000 ft^2 (24,536 m^2) building using the Ontario Hydro electric rate structure. The chiller equipment needed 31 percent of the on-peak demand but consumed only 8 percent of the annual kWh. The annual load factor for this building is only 37 percent. Commercial buildings have poor load profiles of electric power demand. Electric utilities are a capital-intensive industry. To improve their load factor and to sell more kWh, electric utilities tend to shift daytime, more energy-inefficient diesel and gas-turbine plant operation to nighttime base-load highly efficient coal and nuclear plant operation. Electric utilities offer a higher energy rate in daytime on-peak hours and a lower energy rate in off-peak hours. A thermal storage system shifts a part of the electric power demand of the HVAC&R system from the daytime on-peak hours, usually from noon to 8 p.m., to nighttime off-peak hours.

A thermal storage system often consumes approximately the same amount of electricity as a conventional air conditioning system; i.e., a thermal storage system does not necessarily save energy. However, a thermal storage system significantly reduces energy cost.

Electric Deregulation and the Impact on Thermal Storage Systems

Because of the electric deregulation and the use of real-time pricing (RTP) and other electricity rate structures instead of the time-of-use (TOU) rate structure, as discussed in Sec. 25.7, open-market competition creates pressure on utilities to cut costs. According to Silvetti and MacCracken (1998) and the analyses from EIA, the following trend may dominate:

- There is a conflict between the interests of the generation-owning utilities in higher-generation prices and the effects of some demand side management (DSM) programs to reduce demand and possibly to help hold down competitive prices for generation. DSM programs are causing problems.
- Regarding energy-efficient equipment-related incentive plans, something is wrong when a power producer gives money to a customer to purchase less power, particularly if the financial incentive might eventually benefit a competitor.
- Thermal storage systems benefit the power provider, the customer, the power user, and the industry setting the price for that energy.

Benefits and Drawbacks of Thermal Storage Systems

Silvetti and MacCracken (1998) noted why a thermal storage system is attractive:

- Thermal storage systems are one of the few legitimate tools which shift the higher electric demand/electric rate for HVAC&R fully or partially from on-peak hours to a lower electric demand/electric rate in off-peak hours, and therefore lower operating costs.

- Thermal storage systems reduce the equipment size and save initial cost.
- Thermal storage systems do not negatively impact the building's indoor environmental control operations, as the load shedding or some load control programs have done.
- As a thermal storage system is a central system, it uses chilled water from the central plant to cool the air in the AHUs. Thermal storage systems are easily adaptable to the water system of a central air conditioning system. As discussed in Sec. 1.5, central systems served 24 percent of the floor space of commercial buildings in 1995.
- Real-time pricing and other rate structures are available as the power providers maneuver to offer the most competitive structure in the electric deregulation environment.
- Thermal storage systems have longer operating hours of compressors and pumps, and chillers and cooling tower at full-load operation, at lower outdoor temperatures at nighttime, and a backup source for cooling during emergencies.

Drawbacks of thermal storage systems include high initial cost and complicated operation, maintenance, and control.

Full Storage and Partial Storage

The ton-hour, or ton · h (kWh), is the unit of stored refrigeration. One ton-hour is the refrigeration or heat absorption of 12,000 Btu (3.516 kWh) performed by a refrigeration system during a 1-h period. The aim of thermal storage strategy is to incur the lowest possible energy cost and initial investment so that life-cycle costs are minimized. The economic benefit of a thermal storage system can also be assessed by calculating the simple payback period of its cost. A simple payback or life-cycle cost analysis of the building load profile, utility electric rate structure, and system characteristics is always necessary.

Determination of the optimum size of a thermal storage system is based on the utility's electric rate structure (difference between on-peak and off-peak unit charges) and the building refrigeration load profile.

Direct cooling denotes the process by which compressors produce chilled water to cool the building directly. When the cost of the direct cooling by a refrigeration system is lower than the cost of stored energy, the operation of the thermal storage system is said to be at chiller priority. On the other hand, if the cost of the direct cooling is higher than the cost of stored energy, the operation is said to be at storage priority. Construction cost and the utility's incentive payments should be considered. If the energy cost difference between on-peak and off-peak hours is great, the full use of stored energy during on-peak hours may be most economical.

There are two kinds of thermal storage: full storage and partial storage. Figure 31.1 shows the load-time diagrams for full storage and partial storage. For a full-storage, or load shift thermal storage, system, all refrigeration compressors cease to operate during on-peak hours, and the building refrigeration load during that period is entirely offset by the chilled water supplied from the thermal storage tank, as shown in Fig. 31.1*a*.

Partial storage, or load leveling, can be either in load-leveling mode, in which refrigeration compressors are operated at full capacity during on-peak hours, as shown in Fig. 31.1*b*, or in demand-limited mode, in which building electric demand limits only part of the refrigeration compressors operated, as shown in Fig. 31.1*c*.

A utility's demand charge is the building total demand charge, which is the sum of the demand charges for HVAC&R systems and other uses including the electricity for lighting, escalators, computers, and electric appliances.

The following is a comparison of different storage strategies of a thermal storage system with heat recovery and cold air distribution for an office building in Dallas, Texas, as described by Tackett (1989).

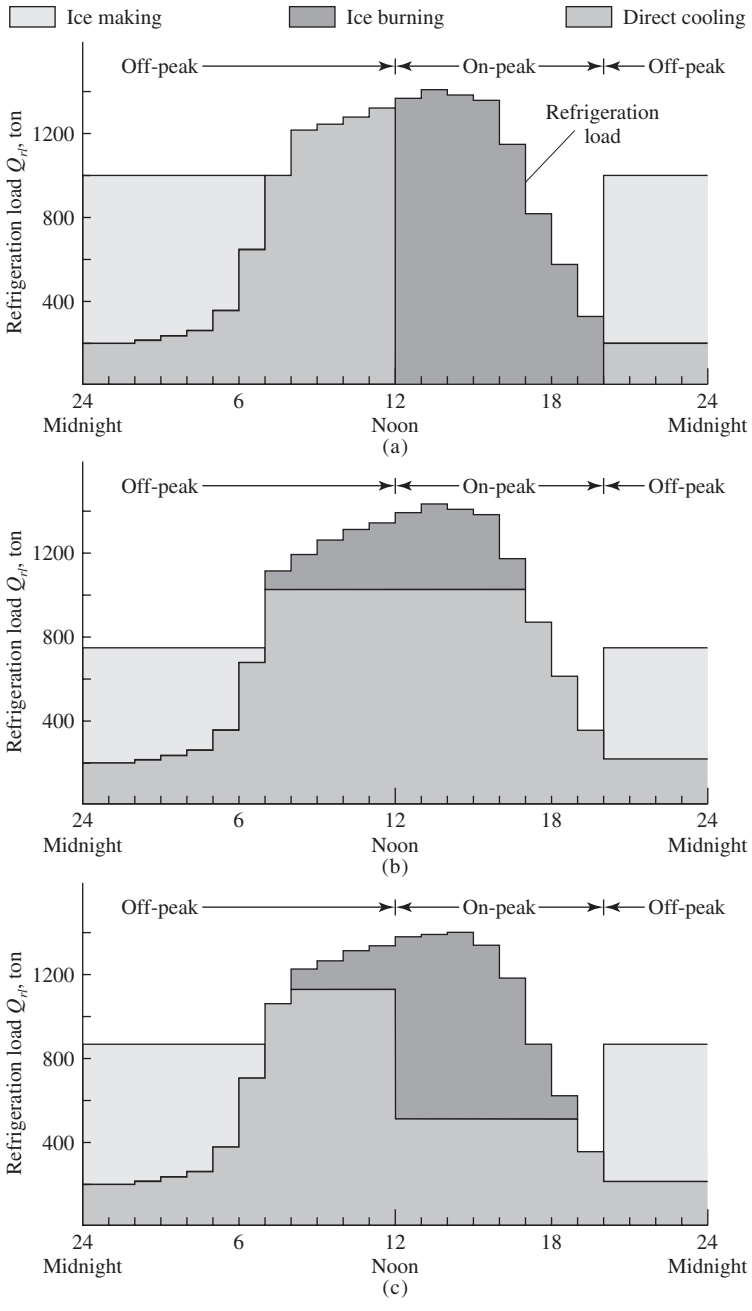


FIGURE 31.1 Full and partial storage: (*a*) full storage; (*b*) partial storage, all compressors operating; (*c*) partial storage, 50 percent of compressors operating.

	Full storage	Partial storage	Demand limited
Utility incentive, $	288,750	132,500	215,000
Net incremental cost, $	291,750	21,750	124,250
Demand limit saving, kW	1,200	400	800
Total saving, $	124,250	64,000	92,750
Simple payback, years	2.35	0.34	1.34

The larger the thermal storage system and the capacity of refrigeration compressors, the greater the savings. Actually, selection of the size of the storage system is better determined according to life-cycle cost, and system reliability should also be considered.

Ice Storage and Chilled Water Storage

In the late 1970s, engineers began experimental thermal storage applications in buildings. Throughout the late 1990s, there are several thousand thermal storage systems operated in various commercial buildings including offices, shopping centers, schools, and hospitals and industrial applications in the United States.

Two thermal storage media are widely used for air conditioning systems: ice storage and chilled water storage. At a temperature difference of 18°F (10°C), 1 lb (2.2 kg) of chilled water can store $18 \times 1 = 18$ Btu (19 kJ) of thermal energy, whereas 1 lb of ice can store $1 \times 144 + 60 - 35 = 169$ Btu (178 kJ). If the density of water is 62.3 lb/ft^3 (997 kg/m^3) and the density of ice is 57.5 lb/ft^3 (920 kg/m^3), for the same stored cooling capacity, the storage volume for ice is only about 0.12 that of the chilled water. In addition, ice storage systems generally provide chilled water at a temperature of 34 to 35°F (1.1 to 1.7°C) to produce cold supply air between 42 and 49°F (5.6 and 9.4°C). Ice storage systems incorporating cold air distribution significantly reduce the volume flow rate of supply air, so air-side fan energy consumption and initial investment drop accordingly.

Generally, ice storage systems have a rather low incremental capital cost compared with conventional air conditioning systems without thermal storage. Ice storage systems can be easily incorporated with cold air distribution. A lower incremental capital cost than that of chilled water storage systems and the incorporation with cold air distribution are two main benefits of the ice storage system.

In addition to the ice storage and chilled water storage systems, phase-change material storage systems are sometimes used. The most common phase-change material used for cool thermal storage is a mixture of inorganic salts, water, and nucleating and stabilizing agents which melts and freezes at 47°F (8.3°C). Phase-change materials have high discharge temperatures and a high storage tank volume of 6.0 ft^3/ton·h (0.048 m^3/kWh) instead of 2.4 to 3.3 ft/ton·h (0.019 to 0.027 m^3/kWh) for other ice storage systems; therefore, they have limited applications in commercial buildings.

Currently, ice storage and chilled water storage systems can be classified into the following categories:

- Ice-on-coil, internal-melt ice storage system (IMISS)
- Ice-on-coil, external-melt ice storage system (EMISS)
- Encapsulated ice storage system (EISS)
- Ice-harvesting ice storage system (IHISS)
- Stratified chilled water storage system (SCWSS)

Chilled water storage systems are often large-capacity storage systems. In newly installed chilled water storage systems, the stratified chilled water storage system is the most widely used.

31.2 ICE-ON-COIL, INTERNAL-MELT ICE STORAGE SYSTEMS

System Description

An ice-on-coil, internal-melt ice storage system uses brine flowing inside coils to make ice and to melt ice in the water that surrounds the coil. The central plant (cooling) and a chilled water or brine-incorporated ice storage system of an ice-on-coil, internal-melt ice storage system consists of the following components: chillers, ice storage tanks, chiller pumps, building pumps, controls, piping, and fittings as well as AHUs, terminals, return air system, smoke control systems, and mechanical exhaust systems.

Centrifugal, screw, and reciprocating chillers are usually used in ice-on-coil internal-melt ice storage systems depending on the size of the plant and types of condenser (water-cooled, air-cooled, or evaporatively cooled) used. In locations where the outdoor air temperature during nighttime off-peak hours drops 20°F (11.3°C) lower than the daytime maximum temperature, air-cooled chillers may sometimes be more efficient than water-cooled chillers.

Figure 31.2 is a schematic diagram of a typical ice-on-coil, internal-melt ice storage system for an office building near Dallas, Texas.

Brine and Glycol Solution

Brine is a salt solution or an aqueous glycol solution used as a heat-transfer medium. Its freezing point is lower than that of water, and depends on the concentration of salt or glycol in solution.

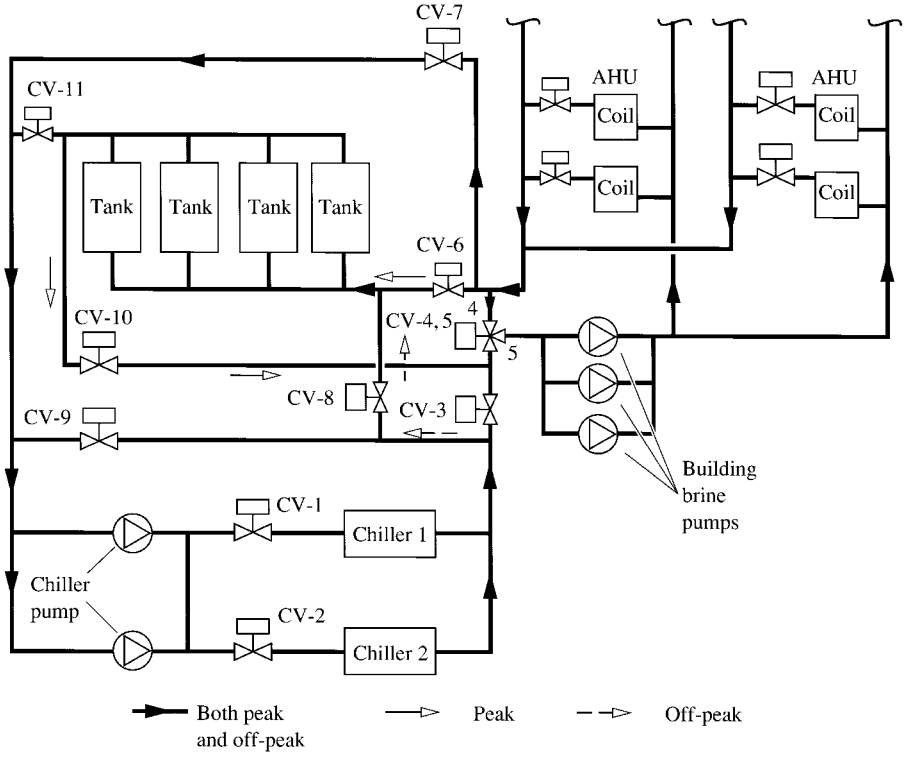


FIGURE 31.2 Schematic diagram of ice-on-coil, internal-melt ice storage system.

Brine is also used as a liquid coolant to absorb or to store heat energy in refrigeration and thermal storage systems.

Ethylene glycol and propylene glycol are brines. They are colorless, nearly odorless liquids. They are often mixed with water at various concentrations and used as freezing point depressants to lower the freezing point of water. Inhibitors must be added to ethylene and propylene glycols to prevent metal corrosion.

The freezing point of an aqueous ethylene glycol solution with a concentration of 25 percent by mass drops to 10°F (−12.2°C), and its rate of heat transfer is about 5 percent less than that of water. The freezing point of a propylene glycol solution with a concentration of 25 percent by mass drops to 15°F (−9.4°C).

The physical properties of aqueous ethylene glycol solution are more appropriate for thermal storage systems than those of aqueous propylene glycol solution. In certain applications, toxicity considerations may be dictated by federal EPA requirements and local codes and regulations.

Ice Storage Tank

In an ice storage system, ice making or charging is the process in which compressors are used to produce ice. Ice burning (ice melting or discharging) means that ice in the storage system is melted in order to produce chilled water to offset the required refrigeration load.

In an ice-on-coil, internal-melt ice storage system, ice is produced, or charged, in multiple storage tanks where closely spaced multicircuited polyethylene or plastic tubes are surrounded by water, as shown in Fig. 31.3. Brine, an aqueous ethylene glycol solution with 25 to 30 percent ethylene glycol and 70 to 75 percent water, circulates inside the tubes at about 24°F (− 4.4°C). The water surrounding the tubes freezes into ice up to a thickness of about 0.5 in. (12.7 mm). Tubes containing glycol solution entering and leaving the tank are arranged side by side alternately to provide more uniform heat transfer.

Brine typically leaves the storage tank at 30°F (− 1.1°C). Plastic tubes occupy about one-tenth of the tank volume, and another one-tenth is left empty to accommodate the expansion of ice during ice making. Multiple ice storage tanks are always connected in parallel. During ice burning or ice melting, brine returns from the cooling coils in the air-handling units at a temperature of 46°F (7.8°C) or higher. It melts the ice on the outer surface of the tubes and is thus cooled to 34 to 36°F (1.1 to 2.2°C). The brine is then pumped to the air-handling units to cool the air again.

In the storage tank, ice is stored and the high-pressure brine inside the tubes is separated from the water, usually at atmospheric pressure, surrounding the tubes in the storage tank.

Case Study: Operating Modes of Ice-on-Coil Ice Storage System

In a typical ice storage system using ice-on-coil, internal-melt storage tanks in a 550,000 ft^2 (51,115 m^2) office building near Dallas, Texas, as described by Tackett (1989), there are two centrifugal chillers. Ethylene glycol is used as the coolant. Each chiller has a refrigeration capacity of 568 tons (1997 kW) when it produces 34°F (1.1°C) brine at a power consumption of 0.77 kW/ton (COP 4.56). If the brine leaves the chiller at 24°F (− 4.4°C), the refrigeration capacity then drops to 425 tons (1494 kW) with a power consumption of 0.83 kW/ton (COP 4.24).

A demand-limited partial-storage strategy is used; i.e., one chiller is operated during on-peak hours, as shown in Fig. 31.1*c*. Meanwhile, ice is also burned during on-peak hours to reduce the demand charge. Ice is charged during off-peak hours to reduce energy costs. The system uses 90 ice-on-coil, internal-melt storage tanks.

For summer cooling, the daily 24-h operating cycle can be divided into three periods: off-peak, direct cooling, and on-peak.

Off-Peak. This is the period from 8 p.m. until the air-handling units start the next morning. During this period, the primary operating mode is ice making, at a maximum capacity of 650 tons

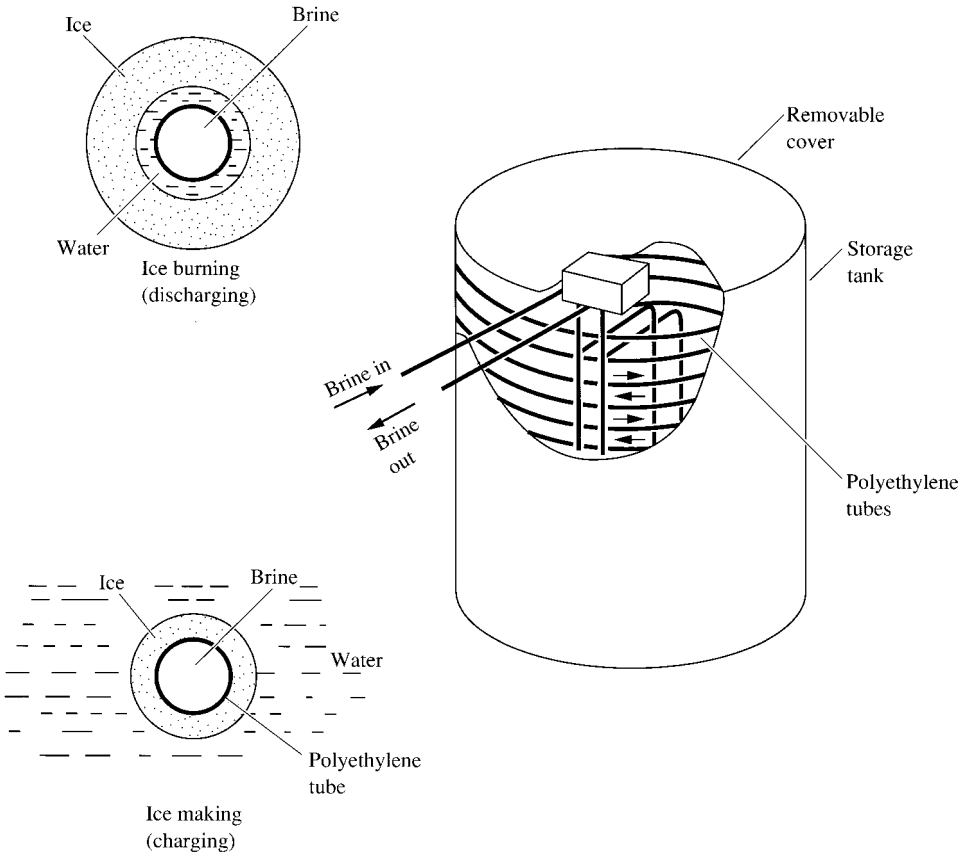


FIGURE 31.3 Ice-on-coil, internal-melt ice storage tank.

(2285 kW). The chillers also provide direct cooling at a capacity less than 200 tons for refrigeration loads that operate 24 h.

In this operating mode, the ice-on-coil, internal-melt storage tanks are charged. At the same time, a 34°F (1.1°C) ethylene glycol solution is supplied to the air-handling units for nighttime cooling. The DDC controller controls the ice storage system in the following operating sequence, as shown in Fig. 31.2:

1. Open control valves CV-1, CV-2, CV-5, CV-6, CV-8, and CV-11; and close control valves CV-3, CV-4, CV-6, CV-7, CV-9, and CV-10.
2. Reset the temperature of the glycol solution leaving the chiller T_{el} to 22°F (− 5.6°C).
3. Reset the load limit of both chillers to 100 percent.
4. Start the chiller and condenser pumps. Chiller pumps operate at high speeds during ice making to provide a higher flow rate and a high rate of heat transfer in the storage tanks as well as a greater head to overcome the pressure drop for both the evaporator and the coils in the ice storage tanks. Chiller pumps operate at low speeds in direct cooling mode.
5. Start chillers 1 and 2 following the lead/lag sequence.
6. After chillers are started, open control valves CV-3 and CV-7.
7. Start the building chilled water circulating pumps in sequence.

8. Modulate control valves 4 and 5, and maintain a 34°F (1.1°C) glycol solution supply temperature to the air-handling units.

If the ice-on-coil, internal-melt ice storage tanks are all 100 percent charged, the ice storage capacity is 7500 ton·h (26,370 kWh).

When the sensors detect that the ice storage tanks are 100 percent charged, the ice-making mode is terminated. If nighttime after-hours cooling is not needed, the ice storage system shuts down. If the ice inventory (the amount of stored ice in the tanks) falls below 90 percent, ice making starts again.

There are two additional operating modes during this period: ice making without direct cooling for after-hours use and ice burning for after-hours use with all chillers shut off.

Direct Cooling. Direct-cooling operation lasts from the start of the air-handling units until noon on weekdays. This period has two operating modes:

- *Direct cooling mode*. In this operating mode, chillers are operating and are reset to 34°F (1.1°C).
- *Direct cooling with ice-burning or ice-melting mode*. In this mode, both chillers are turned on. When the required refrigeration load exceeds both chillers' capacity, some ice storage will be discharged to supplement the chillers.

On-Peak. On-peak hours are from noon until 8 p.m. weekdays. Ice-burning mode, with or without chiller operation, is used in this period. In ice-burning or ice-melting mode, one chiller is operated at the demand limit. This is the primary operating mode during summer cooling. The operating sequence is as follows:

1. Open control valves CV-3, CV-6, CV-7, and CV-10; and close CV-8, CV-9, and CV-11.
2. Open control valve CV-1 and close CV-2 if chiller 1 is required to operate. Open CV-2 and close CV-1 if chiller 2 is required to operate.
3. Modulate control valves 4 and 5 at normal open positions.
4. Reset chilled water temperature leaving the chiller to 32°F (0°C).
5. Set the load limit of the operating chiller to 400 kW.
6. Start one condenser pump.
7. Start chiller pumps 1 and 2 at low speed. Both pumps will operate during ice burning.
8. Start one chiller according to the lead/lag sequence.
9. Modulate control valves 4, 5, 6, and 7 to maintain a 34°F (1.1°C) chilled water supply temperature to the air-handling units.
10. Start the brine circulating pumps in sequence. During on-peak hours, the brine circulating pump needs a greater head to overcome the pressure drop of the coil in the AHU as well as the pressure drop of the coil in the ice storage tanks.

System Characteristics

An ice-on-coil, internal-melt ice storage system is a modular system consisting of many closely packed storage tanks connected in parallel. Such an ice storage system is more flexible during the installation of storage tanks, especially for retrofit projects.

In ice-on-coil, internal-melt ice storage systems, off-peak cooling of the building can be provided by direct cooling from the chillers. During ice burning, melted water separates the tube and ice. Water has a much lower thermal conductivity (0.35 Btu/h·ft·°F, or 0.61 W/m·°C) than that of ice (1.3 Btu/h·ft·°F, or 2.25 W/m·°C), so the capacity of the ice-on-coil, internal-melt ice storage system is dominated by the rate of ice burning or melting.

Other system characteristics of ice-on-coil, internal-melt ice storage systems are listed in Table 31.1.

TABLE 31.1 System Characteristics of Ice-on-Coil, Internal-Melt Ice Storage Systems

	IMISS/EMISS, EISS	IHISS SCWSS
Zone thermal and sound control		
Control zone	Single or multizone	Single or multizone
Conrol methods	DDC or electric	DDC or electric
Control modes	PI, PID, or on/off	PI, PID, or on/off
Sound control	NC 25–40	NC 25–40
Indoor air quality		
Minimum ventilation control	DCV or MPC	DCV or MPC
Filter efficiency	Medium- to high-	Medium- to high-
Humidity control	Optional	Optional
Air system		
Types	VAV air mixing or CV	VAV air mixing or CV
Supply fan	Centrifugal	Centrifugal
Supply fan total pressure	4.5–5.5 in. WC	4.5–5.5 in. WC
Combined fan-motor-drive efficiency	55%	55%
Air economizer	Fixed or differential dry bulb or enthalpy	Fixed or differential dry bulb or enthalpy
Return/relief fan	Centrifugal or axial	Centrifugal or axial
Relief/return fan total pressure	Relief 0.6 in. WC, return 0.5–1 in. WC	Relief 0.6 in. WC, return 0.5–1 in. WC
Combined fan-motor-drive efficiency	35–40%	35–40%
Cooling systems		
Refrigeration compressor	Centrifugal or screw	Centrifugal or screw
Refrigerants	HCFC-123, HCFC-22, HFC-134a	HCFC-123, HCFC-22, HFC-134a
Evaporator	Flooded liquid cooler	IHISS: harvester; SCWISS: flooded liquid cooler
Condenser	Water or evaporatively cooled	Water or evaporatively cooled
Energy performance	IMISS: 0.85–1.2 kW/ton, EMISS: 0.85–1.4 kW/ton, EISS: 0.85–1.2 kW/ton	IHISS: 0.95–1.3 kW/ton, SCWSS: FOM 0.85 to 0.92
Water pumps	Centrifugal	Centrifugal
Combined pump-motor efficiency	65%	65%
Heating systems	Hot water coil or electric heater	Hot water coil or electric heater
Boiler AFUE	Conventional 78%, condensing 93%	Conventional 78%, condensing 93%
Maintenance		
Fault detection and diagnostics	AHU, chiller	AHU, chiller

FOM indicates figure of merit. Refer to Sec. 31.7 for details.

31.3 ICE-ON-COIL, EXTERNAL-MELT ICE STORAGE SYSTEMS

System Description

In an ice-on-coil, external-melt ice storage system, ice builds up on the outer surface of coils or tube banks, which are submerged in water in a storage tank. The refrigerant flows and evaporates inside the tubes. When the ice melts, it cools the water at a temperature between 34 and 38°F (1.1 and 3.3°C) for cooling in the AHUs. The central plant (cooling) and chilled water incorporated ice storage system of an ice-on-coil, external-melt system consists of chillers, evaporating coils, stor-

age tanks, condenser, heat exchanger, refrigerant pumps, chilled water pumps, air system controls, piping, and fittings. A typical ice-on-coil, external-melt ice storage system is shown in Fig. 31.4.

Screw and centrifugal compressors are often used because of their higher efficiency. For an ice-on-coil, external-melt ice storage system with a capacity less than 2400 ton·h (8438 kWh), a reciprocating compressor may also be used. Evaporatively cooled condensers have a higher system energy efficiency ratio (EER) and are often used in many new projects.

Ice Builders

Ice builders are large, well-insulated steel tanks containing many serpentine coils, usually made of steel pipes of 1- to $1\frac{1}{4}$-in. (25- to 31-mm) diameter. HCFC-22 is often used as the refrigerant. The refrigerant-filled serpentine coils are submerged in water in the ice builder and function as evaporators. The ice build up on the coil is between 1 and 2.5 in. (25 and 64 mm) thick. When ice builds up on the coil, the suction temperature of the compressor falls to 22 to 24°F (− 5.6 to − 4.5°C).

Ice is melted by the water circulating over it. The steel tubes of the serpentine coil should be spaced so that the built-up ice cylinders do not bridge each other. If the cylinders are bridged, the paths of water circulation are blocked. Baffle plates are sometimes added to guide the water flow and provide a secondary heat-transfer surface between the refrigerant and water.

The storage tanks containing refrigerant coils are usually located at a lower level or on a grade because of their weight. Because the chilled water system in a multistory building is always under a static head at lower levels, a heat exchanger, usually plate-and-frame type, is used to isolate the storage tank brine system from the chilled water system connected to the AHUs. An alternative is to supply chilled water directly to the storage tanks and pressurize the tanks. This arrangement obviates the use of a heat exchanger and a corresponding increase in brine temperature of about 3°F (1.7°C).

Refrigerant Feed

Two kinds of refrigerant feed are widely used in ice-on-coil, external-melt ice storage systems: direct expansion and liquid overfeed. Direct expansion (DX) uses the pressure difference between the receiver at the high-pressure side and the suction pressure to force the refrigerant to flow through the ice builder. A suction-liquid heat exchanger can be used to cool the liquid refrigerant from the condenser for better efficiency. Direct expansion is simple, and no refrigeration pump is required. Its main drawback is that 15 to 20 percent of the coil surface is used for superheat and is not available for ice buildup.

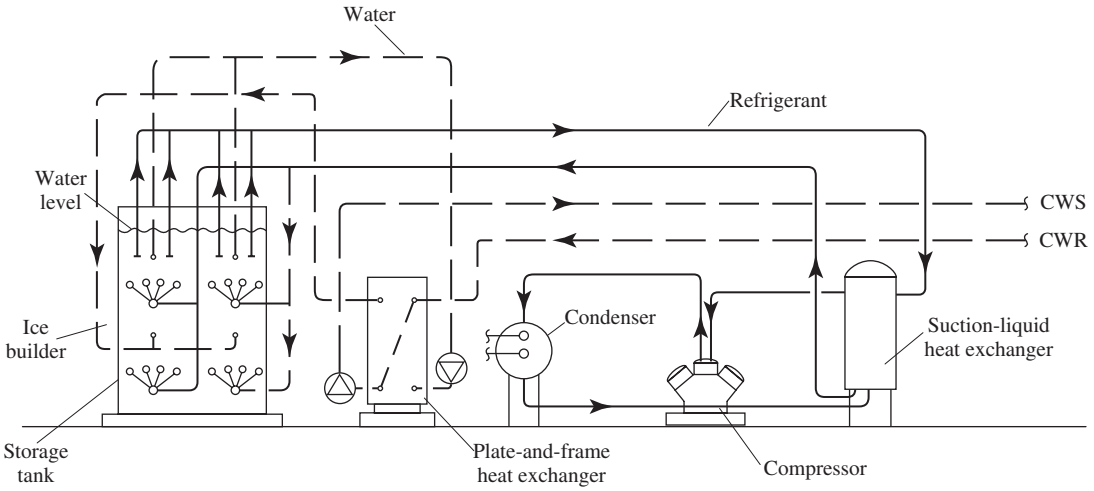
As described in Secs. 10.1 and 10.4, liquid overfeed uses a refrigerant pump to feed ice-builder coils about 3 times the evaporation rate they need. Because the liquid refrigerant wets the inner surface of the ice-builder coils, it has a higher heat-transfer coefficient than direct expansion.

Ice-Charging Control

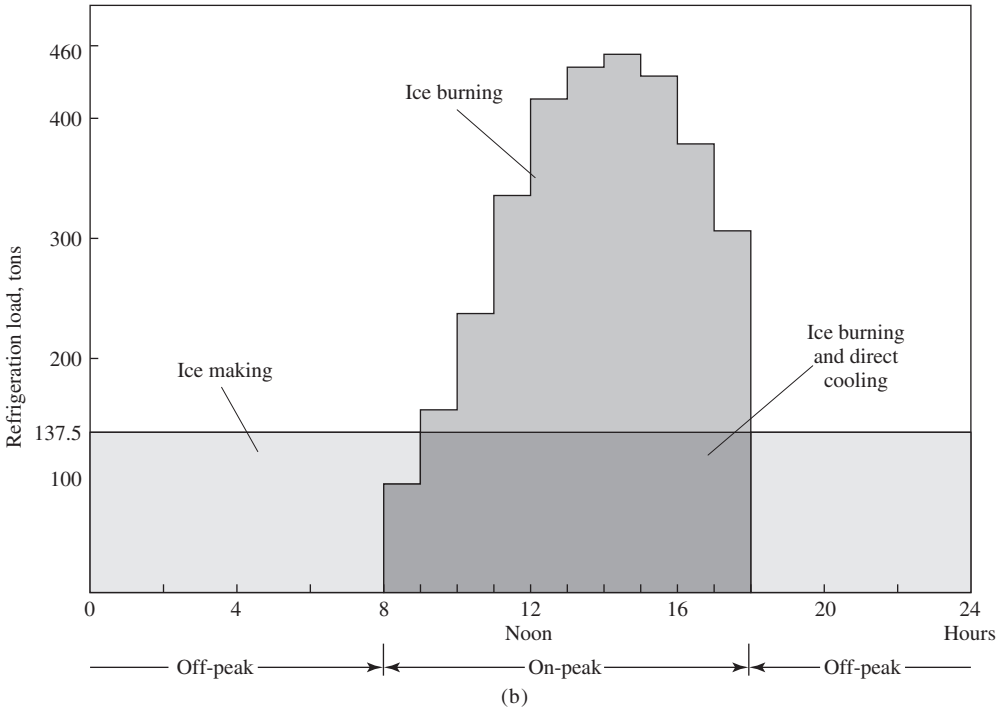
The thicker the ice built up on the coils, the greater the amount of ice stored in the tank. The thickness of ice on the coil should be measured to meet ice-burning requirements during on-peak operating hours or in the direct cooling period. Because ice has a higher volume than water, as the ice builder is charged (i.e., as ice builds up on the coil), the water level rises. An electric probe can sense the water level in the tank and thereby determine the amount of ice stored in the tank.

System Characteristics

The ice-on-coil, external-melt ice storage system is the oldest type of ice storage system. It may be costly and complex. In the storage tank, stored ice occupies only about one-half the volume of the tank, so the ice builder must be larger and heavier.



(a)



(b)

FIGURE 31.4 A typical ice-on-coil, external-melt ice storage system: (*a*) schematic diagram; (*b*) operating diagram.

Case Study: An Ice-on-Coil, External-Melt Ice Storage System

Gilbertson and Jandu (1984) presented an ice-on-coil, external-melt ice storage system for a 24-story office building with an area of 265,000 ft^2 (24,628 m^2) in San Francisco, California. This project was completed in the early 1980s. A schematic diagram of this system is shown in Fig. 31.4*a*. This office tower had a peak refrigeration load of 460 tons (1617 kW), and an ice storage capacity of 3300 ton·h (11,603 kWh) of refrigeration over 24 h. It required a 137.5-ton (4843-kW) refrigeration capacity for the ice storage system for a hot summer day at summer outdoor design conditions. For redundancy, two identical refrigeration systems using HCFC-22 as refrigerant were installed. Each had a 70-nominal-ton (246-kW) compressor, an ice builder of 960 ton·h (3375 kWh), a water-cooled condenser, and other accessories. (Nominal ton is the refrigeration capacity, in tons or kW, at rating conditions.) Direct-expansion refrigerant coils with a suction-liquid heat exchanger were used. When the thickness of built-up ice was about 2 in. (50 mm), the ice storage capacity reached 960 ton·h (3375 kWh).

A plate-and-frame heat exchanger was used to isolate the storage tank from the tower chilled water system. The 34°F (1.1°C) water from the ice builder entering the heat exchanger with an 8°F (4.4°C) temperature increase cooled the chilled water from 54 to 38°F (12.2 to 3.3°C). This was a partial-storage system. The operating modes during a 24-h cycle for this 24-story office tower, shown in Fig. 31.4*b*, are as follows:

Time period	Operating mode
6.00 p.m. to 8.00 a.m.	Off-peak, ice making
8.00 a.m. to 9.00 a.m.	On-peak, ice making and direct cooling
9.00 a.m. to 6.00 p.m.	On-peak, ice melting and direct cooling

The HVAC&R and plumbing system, including ice storage, for this office building cost $2.4 million, which was slightly less expensive than a conventional central system using centrifugal chillers.

System characteristics of ice-on-coil, external-melt ice storage systems are listed in Table 31.1.

31.4 ENCAPSULATED ICE STORAGE SYSTEMS

System Description

In an encapsulated ice storage system, plastic containers, filled with deionized water and ice-nucleating agent, are immersed in a secondary coolant ethylene glycol solution in a steel or concrete tank. Ice is charged and stored when the secondary coolant is at a temperature between 22 and 26°F (− 5.6 and − 3.3°C) circulated through the tank. Ice is melted when the warm coolant returned from the AHUs is circulated through the tank. Chillers can also provide direct cooling at a coolant temperature from 36 to 42°F (2.2 to 5.6°C). The central plant (cooling) and the chilled water incorporated ice storage system of an encapsulated ice storage system consist of the following components: chillers, steel tank, encapsulated containers, pumps, air system controls, piping, and accessories.

Two types of encapsulated ice containers are currently available in the United States: dimpled spheres of 4-in. (100-mm) diameter and rectangular containers approximately $1\frac{3}{8}$ by 12 by 30 in. (35 by 300 by 750 mm). The containers are made of high-density polyethylene and are designed to withstand the pressure due to the expansion during freezing.

When containers are put or stacked inside the storage tank, they allow free circulation of fluid and do not provide unwanted short-circuit fluid flow which causes degradation of performance. The storage tank can be an open, nonpressurized type or pressurized type. An open storage tank needs a barrier to keep the frozen containers submerged into the coolant.

Location of Chiller and Storage Tank

In encapsulated ice storage systems and ice-on-coil, internal-melt ice storage systems, the chillers and storage tanks are usually connected in series, as shown in Fig. 31.5. When partial storage is used, their relative location can be either in chiller upstream or chiller downstream arrangements. In a chiller upstream arrangement, as shown in Fig. 31.5*a*, the chilled water returned from AHUs at 46°F (7.8°C) is often first cooled in the chiller to 40°F (4.4°C), and then it enters the storage tank and is cooled down to 34°F (1.1°C). In the chiller upstream arrangement, since the chilled water cooled at the chiller is at a higher temperature, this results in a higher COP at the chiller. However, the usable portion of the total storage capacity will be reduced because of the lower storage tank discharge temperature.

In a chiller downstream arrangement, as shown in Fig. 31.5*b*, the chilled water returned from the AHUs at 46°F (7.8°C) is often first cooled in the storage tank to 40°F (4.4°C), and then it enters the chiller and is cooled down to 34°F (1.1°C). In a chiller downstream arrangement, the COP of the chiller is lower, and the usable portion of the total storage capacity of the ice storage tanks is increased.

Usually, during partial storage, the chiller upstream arrangement is often used for higher efficiency in the chiller. When ice storage capacity becomes a problem, an analysis should be undertaken to make an optimum choice.

Controls

According to Dorgan and Elleson (1993), ice-charging inventory in the storage tank is measured based on the displacement of water in the tank when the ice is formed inside the encapsulated

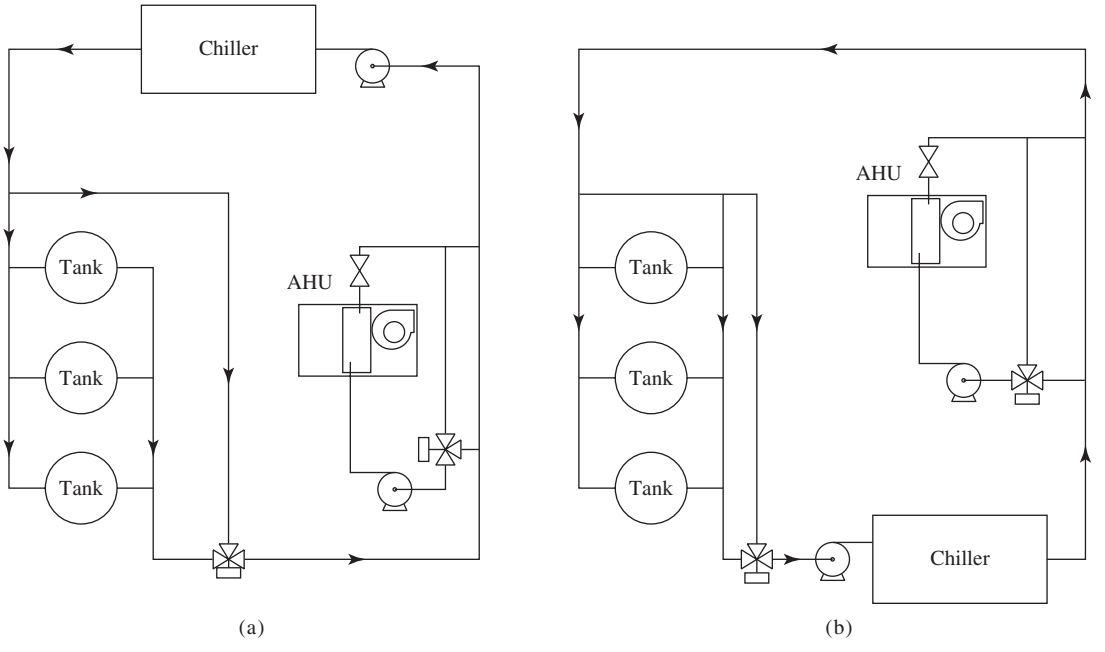


FIGURE 31.5 Relative location of chiller and storage tank: (*a*) chiller upstream; (*b*) chiller downstream.

containers. For open tanks, a static pressure transducer is often used to detect the water level in the storage tank. In pressurized tanks, the expansion of the frozen containers forces the secondary coolant overflowing into a separate inventory tank, and its water level is measured. A pump is used to pump the overflowing fluid in the inventory tank back into the storage tank after discharging. With nondimpled spherical containers that expand very little as the encapsulated ice freezes, storage inventory can be monitored based on the integrated flow and temperature measurements.

Encapsulated ice storage systems use a storage tank bypass three-way modulating valve to control the chilled water leaving temperature.

Encapsulated ice storage systems with a chiller upstream arrangement are well suited to chiller priority control. When the system refrigeration load is less than the chiller capacity, the chilled water bypasses the storage tanks completely. As soon as the system refrigeration load exceeds the chiller capacity and the chiller leaving temperature increases above the leaving set point, the control system diverts part of the chilled water flow through the storage tanks to maintain the required supply temperature to the AHUs.

Storage priority is more complicated to achieve. A required refrigeration load prediction algorithm to forecast the chiller cooling is needed each day. Chiller capacity is then limited by increasing the chilled water leaving setpoint, and most of or all the refrigeration load is then met by the ice storage.

Generally, chillers should be controlled at full load during charging to prevent the reduction of system efficiency and incomplete charging of ice storage. The chiller leaving temperature setpoint should be set at or below the minimum required charging temperature so that the chiller is fully loaded throughout the charging cycle.

Charging and Discharging

For encapsulated ice storage systems, the charging temperature decreases during the charge cycle as the thickness of ice through which heat is transferred increases. Arnold (1991) reported that encapsulated containers are subject to supercooling, i.e., cooling of the liquid water inside the container below its freezing point prior to the ice formation. Supercooling occurs only in fully discharged containers and results in a reduced rate of heat transfer at the beginning of the charging process. Supercooling can be significantly reduced by the addition of nucleating agents.

For an entering chilled water temperature at the beginning of charging of 32°F (0°C) and a chilled water temperature at the end of charging of 20 to 26°F (− 6.7 to − 3.3°C), a typical range of charging temperatures is between 4 and 12°F (2.2 and 6.7°C) corresponding to a charging period of 8 to 16 h.

Encapsulated ice storage systems have a steadily falling discharge rate when the discharge temperature is kept constant, or a steadily rising discharge temperature when the discharge rate is constant. This is due to the decreasing area of ice in contact with the container as the ice melts. The encapsulated ice storage discharge temperature typically begins at 32°F (0°C) and ends at a discharge temperature of 38 to 45°F (3.3 to 7.2°C) with a discharge temperature range of 6 to 13°F (3.3 to 7.2°C).

System Characteristics

System characteristics of encapsulated ice storage systems are listed in Table 31.1.

31.5 ICE-HARVESTING ICE STORAGE SYSTEMS

System Description

In an ice-harvesting ice storage system, ice is produced in a harvester, which is separate from the storage tank where ice is stored. The evaporator of the chiller is a vertical plate heat exchanger mounted above a water/ice storage tank. Low-pressure liquid refrigerant is forced through the

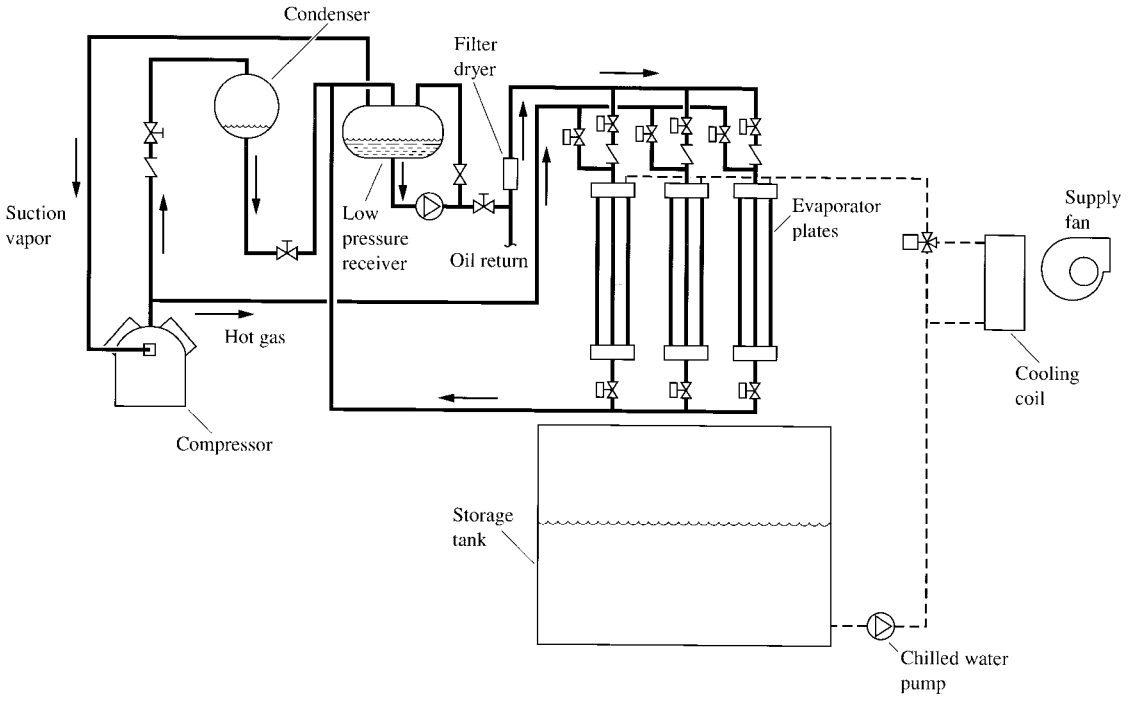


FIGURE 31.6 Schematic diagram of a typical ice-harvesting ice storage system.

hollow inner part of the plate heat exchanger, in which liquid refrigerant is vaporized, and produces a refrigeration effect.

The central plant (cooling) and brine incorporated ice storage system in an ice-harvesting ice storage system consists of the following equipment and main components: chillers, an ice harvester, storage tank, air system controls, piping, and accessories, as shown in Fig. 31.6.

Ice Making or Charging

During ice-making or -charging mode, a chilled aqueous ethylene glycol solution with a concentration of 25 to 30 percent is pumped from the storage tank and distributed over the outer surface of the evaporator plates at a temperature equal to or slightly above 32°F (0°C). It then flows downward along the outer surface of the plate in a thin film. Water is cooled and then frozen into ice sheets approximately 0.2 to 0.3 in. (5 to 7.5 mm) thick. Periodically, hot gas is introduced into one-fourth of the evaporator plates by reversing the refrigerant flow. Ice is harvested, or released from the outer surface of the plates, in the form of flakes or chunks and falls into the storage tank below. Ice is formed in 20 to 30 min and is harvested within 20 to 40 s. During harvesting, this section of plate evaporator acts as a condenser.

Ice accumulates in the storage tank to occupy slightly less than 60 percent of the volume of the tank. Because the ice flakes are usually smaller than 6 in. by 6 in. by 0.25 in. (1500 mm by 1500 mm by 63 mm), there is a large contact area between the return brine from the cooling coils and the ice. The time required to melt the ice in the storage tank is less than one-tenth of the time needed in ice making or charging.

For a reciprocating compressor using an evaporative condenser, the power consumption of the chiller during ice making is about 0.95 to 1.1 kW/ton (COP 3.7 to 3.22). Because the evaporator

plates must be located above the storage tank, ice-harvesting systems need more headroom than other ice storage systems.

Chiller Operation

During off-peak hours, an ice-harvesting system can also be used to lower the temperature of brine returning from the AHUs. Brine at a temperature of 34°F (1.1°C) is supplied to the air-handling units to cool the air to a supply temperature of 42 to 45°F (5.6 to 7.2°C) during direct cooling. It is then returned to the ice harvester at 50 to 60°F (10 to 15.6°C) and distributed over the evaporator plates directly. After falling from the evaporator plates, brine is again cooled to a temperature of 34°F (1.1°C) before it leaves the storage tank.

Because of the higher temperature of return brine distributed over the evaporator plates, the capacity of the ice harvester increases, and its power consumption decreases during chiller operation. These changes in capacity and power consumption depend mainly on the temperature of return brine distributed over the evaporator plates. During chiller operation, power consumption usually varies between 0.75 and 0.85 kW/ton (COP 4.69 and 4.14).

Ice-harvesting ice storage systems are effective in ice-making and ice-burning operations. The temperature of brine from the storage tank of the ice harvester can be lowered to 34°F (1.1°C), which is 2°F (1.1°C) lower than in the ice-on-coil, internal-melt ice storage system. Ice-harvesting systems have been successfully used in load shifting and load leveling to reduce electric demand and energy cost. However, melting of the ice during the harvesting process not only decreases the amount of ice harvested, but also adds an incremental refrigeration load to the system, which increases the power consumption during the ice-making process.

An ice-harvesting ice storage system with brine system is an open system. More water treatment is required than in an ice-on-coil, internal-melt ice storage system whose brine system is a closed system.

System Characteristics

System characteristics of an ice-harvesting ice storage systems are listed in Table 31.1.

31.6 COMPARISON OF VARIOUS ICE STORAGE SYSTEMS

According to Dorgan and Elleson (1993), the discharge temperature, the energy consumption of the chillers, the roughly estimated installed cost of storage tanks and accessories, and the chiller cost without installation are as shown below:

	IMISS	EMISS	EISS	IHISS
Discharge temperature, °F	34–38	34–36	34–38	34–36
Chiller energy use, kW/ton	0.85–1.2	0.85–1.4	0.85–1.2	0.95–1.3
Storage tank volume, ft^3/ton·h	2.4–2.8	2.8	2.4–2.8	3.0–3.3
Storage installed cost, $/ton·h	50–70	50–70	50–70	20–30
Chiller cost, $/ton	200–500	200–500	200–500	1000–1500

According to Potter et al. (1995), a survey of 196 thermal storage systems was undertaken. The users of the thermal storage systems were asked to assess the ice storage systems according to an index of 1 to 10. One indicated complete dissatisfaction, whereas 10 meant complete satisfaction. The ratings for the ice storage systems are as follows:

Ice-on-coil, internal-melt ice storage systems	7.82
Ice-on-coil, external-melt ice storage systems	6.78
Encapsulated ice storage systems	7.25
Ice-harvesting ice storage systems	5.02

In addition to these systems, there is another ice storage system called an ice slurry ice storage system in which suspended ice crystals are formed. Ice slurry systems have limited commercial application because of their higher costs.

31.7 STRATIFIED CHILLED WATER STORAGE SYSTEMS

System Description

A stratified chilled water storage system often uses a large storage tank to store chilled water at a temperature between 40 and 45°F (4.4 and 7.2°C). The stored chilled water offsets the building refrigeration load during on-peak hours to shift the load to the off-peak hours and reduces the energy cost. Chilled water in the storage tank is stratified into three regions because of its gravity: top warmer return water from the AHUs, middle region of steep temperature gradient, and bottom colder chilled water from the chillers. The central plant (cooling) and the chilled water incorporated storage system consist of chillers, a cylindrical storage tank, pumps, piping, air system controls, and accessories.

Basics for Chilled Water Storage

The stored cooling capacity of a chilled water storage system depends on the temperature difference between the warm water return from the AHUs and the chilled water stored in the tank, and the amount of water stored. The larger the storage tank, the lower the capital cost per unit stored volume. According to Dorgan and Elleson (1993), a chilled water storage system is economical when its storage capacity exceeds 2000 ton·h or 200,000 gal (7000 kWh or 760 m^3). Currently, chilled water systems achieve thermal separation between cold charged water and warm return water by stratification, multiple tanks, membrane, diaphragm, and baffles. The stratified tank is the simplest and most efficient method. Chilled water storage systems need a storage tank volume of 11 to 21 ft^3/ton·h (0.089 to 0.169 m^3/kWh) compared to 2.4 to 3.3 ft^3/ton·h (0.019 to 0.027 m^3/kWh) for ice storage systems.

Charging and Discharging. Charging is the process of filling the storage tank with chilled water from the chiller, usually at a temperature between 40 and 45°F (4.4 and 7.2°C). Meanwhile, the warmer return chilled water from the air-handling units or terminals, usually at a temperature between 55 and 60°F (11.1 and 15.6°C), is extracted from the storage tank and pumped to the chiller to be cooled.

Discharging is the process of discharging the chilled water, at a temperature between 41 and 45°F (5.0 and 7.2°C), from the storage tank to the air-handling units and terminals. At the same time, the warmer return chilled water from the coils fills the tank by means of storage water pumps.

Loss of Cooling Capacity during Storage. During the storage of chilled water, the following processes result in losses in cooling capacity:

- Stored chilled water is warmed by direct mixing of warmer return chilled water and stored colder chilled water.

- Heat from prviously stored warmer return chilled water is transferred from the warmer tank wall to the stored chilled water.
- Heat is transferred through the tank wall from the warmer ambient air.

Figure of Merit. A more easily measured, enthalpy-based *figure of merit* (FOM) is often used to indicate the loss of cooling capacity of the stored chilled water during the charging and discharging processes in a complete storage cycle. The FOM is defined as

$$\text{FOM} = \frac{Q_{\text{dis}}}{Q_{\text{ch}}} = \frac{\Sigma \dot{m}_w c_{pw}(T_{rc} - T_o)}{\dot{m}_w c_{pw}(T_{rc,m} - T_i)} \tag{31.1}$$

where Q_{dis} = cooling capacity available during discharge process, Btu/h (W)
Q_{ch} = theoretical cooling capacity available during charging process, Btu/h (W)
$\dot{m}_w$, $\Sigma \dot{m}_w$ = mass flow rate and summation of mass flow rate of water, lb/h (kg/s)
c_{pw} = specific heat of water, Btu/lb · °F (J/kg · °C)
T_{rc} = warmer return chilled water temperature filling storage tank during discharge process, °F (°C)
T_o = outlet temperature of stored chilled water, °F (°C)
$T_{rc,m}$ = mass-weighted average temperature of return chilled water at inlet during discharge process, °F (°C)
T_i = inlet temperature of stored chilled water during charging process, °F (°C)

The smaller the losses of cooling capacity during chilled water storage, the greater the value of FOM.

Storage Tanks

Chilled water storage tanks are usually flat-bottomed vertical cylinders. A cylindrical tank has a lower surface-to-volume ratio than a rectangular tank. Large cylindrical tanks typically have a height-to-diameter ratio of 0.25 to 0.35. Steel is the commonly used material for above-grade tanks, and concrete is widely used for underground tanks. In certain projects, precast, prestressed, cylindrical concrete tanks with enclaved watertight steel diaphragms are used for large chilled water storage facilities with a volume over 2.5 million gal (9463 m^3). All outdoor above-grade structures should have a 2-in.- (50-mm-) thick external insulation layer spray-on polyurethane foam, a vapor barrier, and a highly reflective top coating.

Stratified Tanks

Stratified tanks rely on the buoyancy of warmer return chilled water, which is lighter than colder chilled water, to separate these two chilled waters during charging and discharging. Diffusers are used to lower entering and leaving water velocity to prevent mixing. In a stratified tank, colder stored chilled water is always charged from the bottom diffusers arranged concentrically, as shown in Fig. 31.7*a*. It is also discharged from the same bottom diffusers.

The warmer return chilled water is introduced to and withdrawn from the tank through the top lateral diffusers.

According to field measurements, stratified tanks have a figure of merit between 0.85 and 0.92. Tran et al. (1989) showed that there is no significant difference in FOM between stratified tanks and membrane tanks or empty tanks. A membrane tank is a storage tank in which a membrane separates the colder stored chilled water and warmer return water. An empty tank is a storage tank in which walls are used to separate the colder and warmer chilled water.

Compared with membrane tanks and empty tanks, stratified tanks have the advantages of simpler construction and control, greater storage capacity, and lower cost. Stratified tanks are widely used in chilled water storage installations.

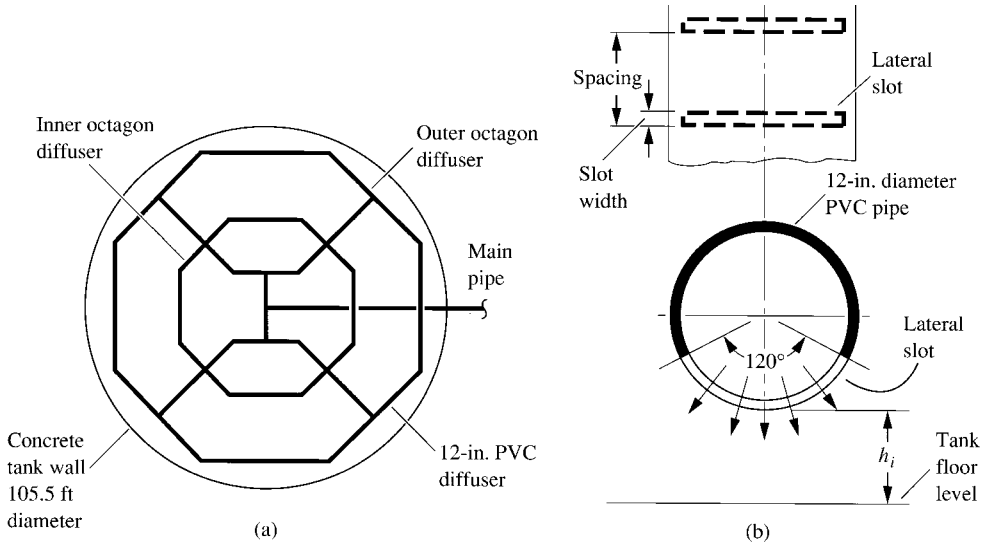


FIGURE 31.7 Double-octagon diffuser for a cylindrical stratified tank: (*a*) plan view of bottom diffusers; (*b*) 12-in. diameter PVC pipe diffuser.

Temperature Gradient and Thermocline

Vertical temperature profiles are formed during charging or discharging in stratified tanks at various time intervals. Temperature profiles may be illustrated on a height-temperature (*H-T*) diagram at the beginning, the middle, and near the end of the charging process, as shown in Fig. 31.8. In the middle of the charging process along the vertical height of the storage tank, chilled water is divided into three regions: bottom colder-and-heavier stored chilled water, thermocline, and top warmer-and-lighter return chilled water.

A thermocline is a stratified region in which there is a steep temperature gradient. The water temperature often varies from 42 to 60°F (5.6 to 15.6°C). The thermocline separates the colder stored chilled water from the warmer return chilled water. The thinner the thermocline, the smaller the mixing loss.

Diffusers

The layout and configuration of diffusers in a stratified tank have a significant effect upon the mixing of the colder and warmer chilled water, as well as on the formation of the thermocline. The purpose of the diffusers and their connecting piping is to distribute the incoming chilled water evenly, so that it flows through the inlet openings with sufficiently low velocity (usually lower than 0.9 ft/s, or 0.27 m/s) to minimize mixing of colder and warmer chilled water. Wildin (1990) recommended that diffuser design and installation take into account the following:

- Warmer return chilled water should be introduced as closely as possible to the top water level in the stratified tank. Colder stored chilled water should be introduced just above the bottom floor of the tank.
- The inlet temperature of chilled water should be controlled within a narrow band (± 2°F or ± 1.1°C) during charging to avoid additional mixing.

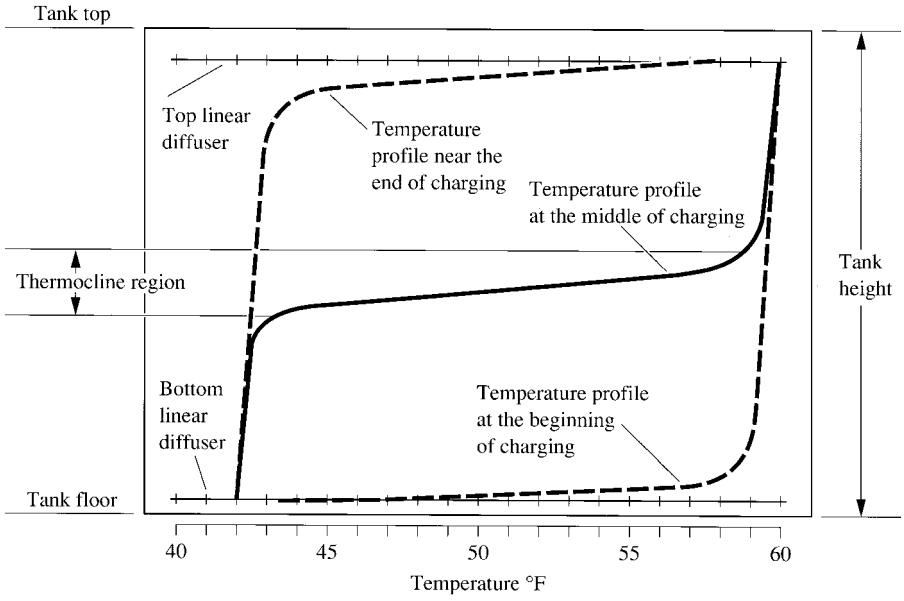


FIGURE 31.8 Temperature profile and thermocline in a stratified tank.

- Obstructions in the flow crossing the tank, other than diffusers and the connecting piping, should be minimized.
- The primary function of diffusers is to reduce mixing. Mixing can occur at two points: at the start of the charging and discharging processes during the formation and reformation of the thermocline, and at the inlet side of the thermocline after the thermocline has been formed.
- Mixing near the inlet diffuser can be minimized if the incoming chilled water initially forms a thin layer of gravity current that travels across the tank because of the gravity difference, rather than inertia. Gravity current slowly pushes the chilled water originally in the tank out of the way so that mixing only occurs at the front of the gravity current when it first crosses the tank.
- There are two kinds of diffusers: linear diffusers and radial disk diffusers. Large stratified tanks usually incorporate linear diffusers.
- Inlet flow from the top diffusers should be upward or horizontal. Bottom diffusers should flow downward and have slots spreading at 120°. The cross-sectional inlet area of the branch pipe leading to the diffuser should be at least equal the total area of the diffuser openings in that branch.
- Mixing on the inlet side of thermocline depends on the inlet Reynolds number Re_i and Froude number Fr_i. The inlet Reynolds number is closely related to the inlet velocity and is defined as

$$\mathrm{Re}_i = \frac{\dot{V}_w}{l_{\mathrm{dif}}\nu_w} = \frac{0.00223\dot{V}_{\mathrm{gal}}}{l_{\mathrm{dif}}\nu_w} \tag{31.2}$$

where $\dot{V}_w$ = volume flow rate of chilled water, ft^3/s

$\dot{V}_{gal}$ = volume flow rate of chilled water, gpm
l_{dif} = linear length of diffuser, ft
ν_w = kinematic viscosity of water, ft^2/s

According to Wildin (1990), when $Re_i < 850$, loss due to mixing and loss of cooling capacity during discharge can be significantly reduced. If Re_i is determined, the length of the linear diffuser can be calculated from Eq. (31.2). The inlet Froude number Fr_i is defined as

$$Fr_i = \frac{\dot{V}_w}{l_{dif}[gh_i^3(\rho_i - \rho_a)/\rho_i]^{0.5}} \tag{31.3}$$

where g = acceleration of gravity, ft/s^2 (m/s^2)
ρ_i = density of inlet water, lb/ft^3 (kg/m^3)
ρ_a = density of ambient water stored in tank at diffuser level, lb/ft^3 (kg/m^3)

In Eq. (31.3), h_i indicates the inlet opening height, ft (m). It is the vertical distance occupied by the incoming flow when it leaves the diffuser and forms the gravity current. For the bottom diffusers, inlet opening height h_i indicates the vertical distance between the tank floor and the top of the opening of the diffuser.

Self-balancing. Self-balancing means that the water flow introduced to or extracted from the tank should be self-balanced according to requirements, i.e., evenly distributed at all flow conditions. This includes the following requirements:

- The piping design should be symmetric.
- Branch pipes should be equal in length.
- Flow splitters should be added at the appropriate points.
- Pipe diameter reduction should be combined with the flow splitter.
- Long-radius elbows should be used.

Charging and Discharging Temperature versus Tank Volume

The FOM of a stratified tank is closely related to the temperature difference of the outlet temperature of stored chilled water during discharging T_o and the inlet temperature of stored chilled water during charging T_i. For a complete charging and discharging cycle, the average T_o during discharging is always greater than the average T_i because of the mixing loss and heat gains, provided that the water flow rate is constant. The smaller $T_o - T_i$, the higher the FOM.

Figure 31.9 shows curves of chilled water temperature versus tank volume during the charging and discharging processes of a complete chilled water storage cycle in a large stratified tank. Inlet and outlet temperatures are measured at the openings of the top and bottom diffusers. During the charging process, return chilled water is extracted from the top diffusers of the stratified tank, cooled in the chiller, and charged into the stratified tank again through the bottom diffusers. The inlet temperature of the stored chilled water T_i gradually decreases as the stored volume increases. This is due to a comparatively lower rate of heat transfer to the inlet water from the warmer ambient water, piping, and tank wall after the beginning of the charging process.

During the discharging process, stored chilled water is extracted from the stratified tank and supplied to the cooling coils in the air-handling units and terminals. The return chilled water is introduced to the stratified tank through the top diffusers. The outlet temperature of the stored chilled water T_o gradually increases as the stored volume decreases.

Both the outlet temperature of return chilled water during charging $T_{rc,o}$ and inlet temperature of return chilled water during discharging $T_{rc.i}$ should be controlled between 55 and 60°F (12.8 and 15.6°C) so that stratification can be maintained in the storage tank.

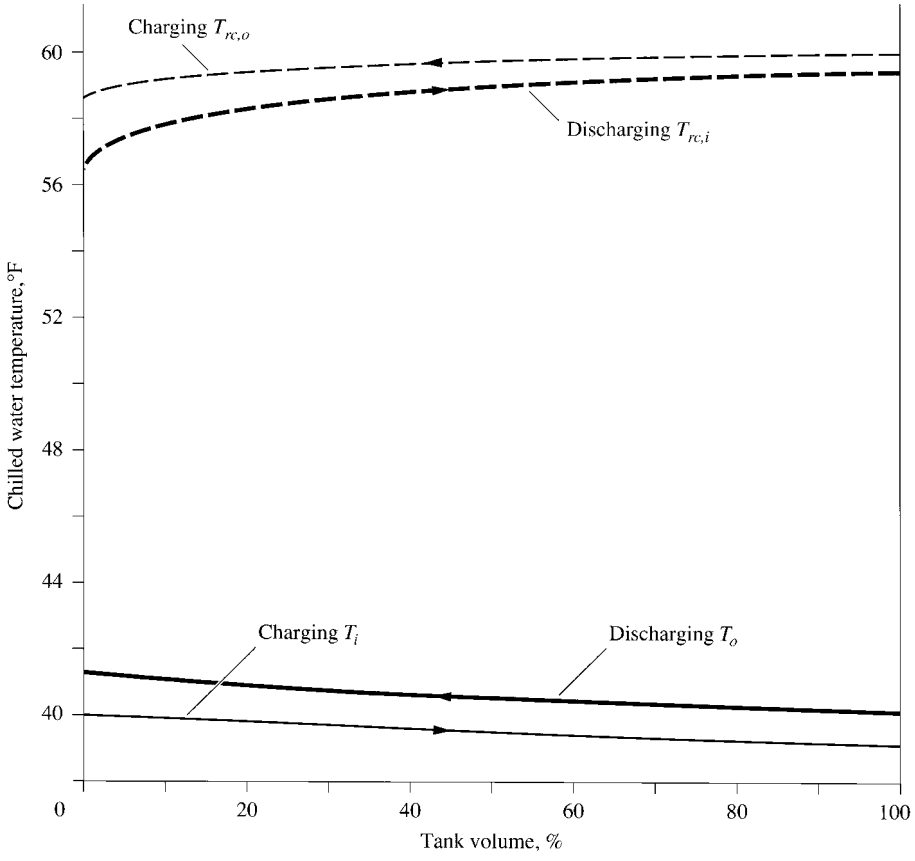


FIGURE 31.9 Chilled water temperature versus tank volume curves during charging and discharging processes.

System Characteristics

System characteristics of stratified chilled water storage systems are listed in Table 31.1.

31.8 CASE STUDY: A STRATIFIED CHILLED WATER STORAGE SYSTEM

A full-storage stratified chilled water storage system was completed in August 1990 to serve a 1.142 million ft^2 (106,134 m^2) electronics manufacturing facility in Dallas, Texas. The following are the details of this project as described by Fiorino (1991).

Chilled Water Storage System

The 2.68-million-gal (10,144-m^3) stratified tank is a precast, prestressed, cylindrical concrete water tank. The design parameters of this stratified chilled-water storage system are as follows:

Storage cooling capacity	24,500 ton· h (861,420 kWh)
Maximum refrigeration load	3200 tons (11,251 kW)
Charge process duration	16 h
Discharge process duration	8 h
Inlet temperature during charging T_i	40°F (4.4°C)
Limiting outlet temperature during discharging T_o	42°F (5.6°C)
Inlet temperature during discharging $T_{re,i}$	56°F (13.3°C)
Maximum volume flow rate	5120 gpm (323 L/s)
Tank diameter	105.5 ft (32.2 m)
Tank height	41 ft (12.5 m)
Tank volume	2.68 million gal (10,144 m^3)
Usable tank volume	90%

There are two 1200-ton (4220-kW) and two 900-ton (3164-kW) centrifugal chillers to serve zone 1 and zone 2, as shown in Fig. 31.10. There are also five chiller pumps, one of which is a standby. Two variable-speed building pumps are installed for each zone, one of which is a standby pump. Three storage pumps are used for chilled water storage, one of which is standby.

Concentric Double-Octagon Diffusers

Concentric double-octagon diffusers are used at both the top and bottom of the stratified tank, as shown in Fig. 31.7*a*. Each octagon introduces 50 percent of the total volume flow during the charging process. Over time, a double-octagon arrangement provides nearly twice the effective linear length for diffusers of a single-octagon arrangement.

The total effective length of the eight diffusers in the outer octagon is 559 ft (170 m). For a total flow of $0.5 \times 5120 = 2560$ gpm (162 L/s) through the outer octagon, the volume flow rate per linear ft (0.30 m) of diffuser is

$$\frac{\dot{V}_{\text{dif}}}{l_{\text{dif}}} = \frac{2560 \times 0.1337}{559 \times 60} = 0.0102 \text{ ft}^2/\text{s}$$

The kinematic viscosity of water at 42°F is 1.66×10^{-5} ft^2/s, so the inlet Reynolds number is calculated as

$$\text{Re}_i = \frac{\dot{V}_{\text{dif}}}{l_{\text{dif}} \nu_w} = \frac{0.0102}{1.66 \times 10^{-5}} = 615$$

Similarly, the inlet Reynolds number for inner octagon is 1068. Both are close to the upper limit of 805.

The acceleration of gravity $g = 32.2$ ft/s^2, and

$$\frac{\rho_i - \rho_a}{\rho_i} = \frac{62.4263 - 62.3864}{62.3864} = 0.00064$$

For a common inlet opening height $h_i = 5.64$ in, or 0.47 ft (0.14 m), the inlet Froude number for diffusers in the outer octagon is calculated as

$$\text{Fr}_i = \frac{0.0102}{[(32.2)(0.47)^3(0.00064)]^{0.5}} = 0.22$$

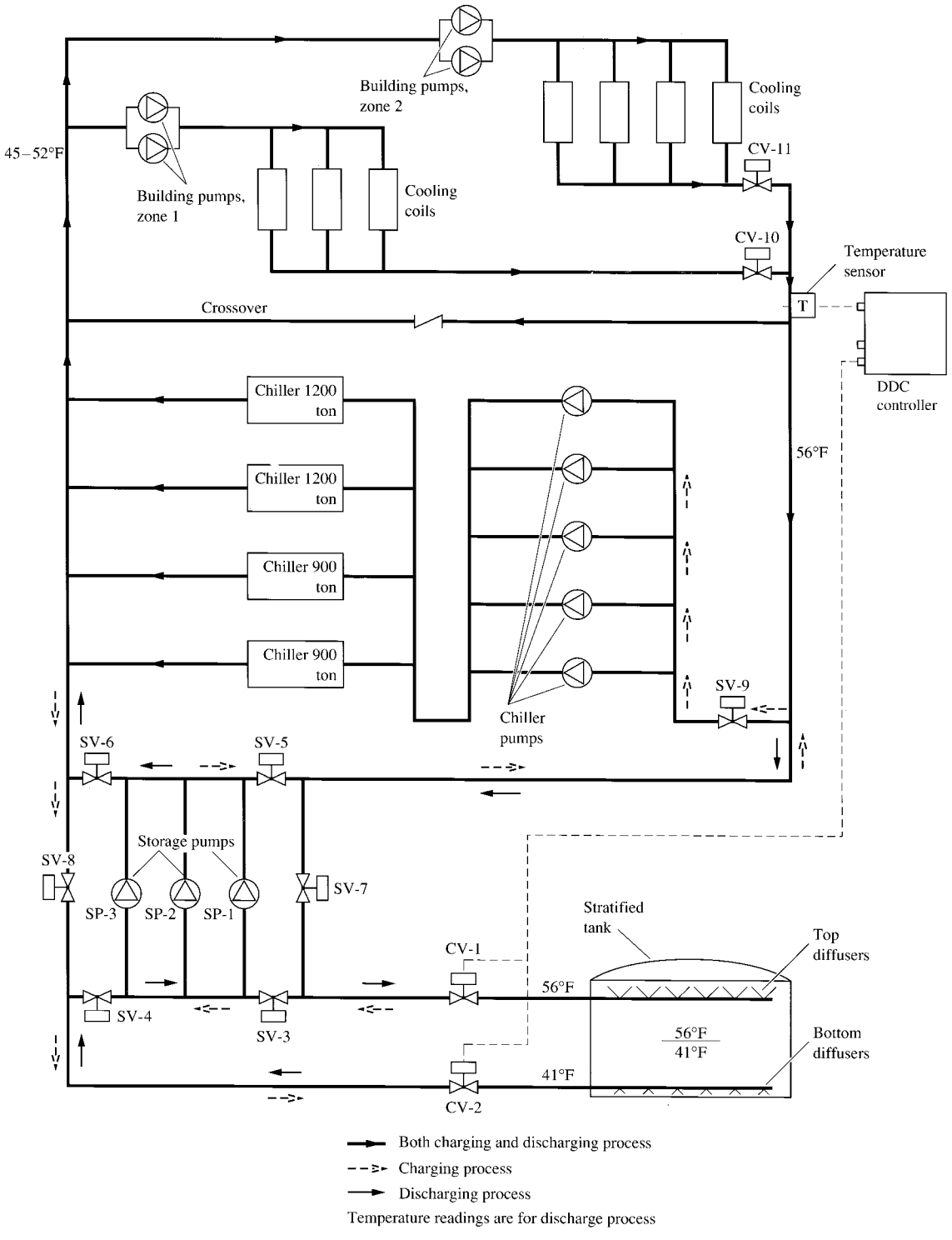


FIGURE 31.10 Schematic diagram of a stratified chilled water storage system for an electronic facility in Dallas, Texas.

Similarly, the inlet Froude number for diffusers in the inner octagon is 0.38.

At a lateral slot spacing of 0.5 ft (0.15 m) in the inner octagon diffusers and 0.87 ft (0.26 m) in the outer octagon diffusers, each linear diffuser has 32 lateral slots. If the maximum inlet velocity is 0.9 ft/s (0.27 m/s), the opening area for each lateral slot is

$$\frac{5120 \times 0.1337}{60(0.9)(32)(8 + 8)} = 0.025 \text{ ft}^2 \ (0.023\text{m}^2)$$

Because the lateral slot is spread at an angle of 120° downward, if the length of the lateral slot is about 1 ft (0.30 m), the width of the slot is 0.025/1 = 0.025 ft, or about 0.3 in. (7.6 mm). If the cross-sectional area of each linear diffuser is equal to the slot openings, the diameter of the diffuser is therefore

$$D = \left(\frac{4A}{\pi}\right)^{0.5} = \frac{4 \times 0.025 \times 32}{\pi}(0.5) = 1.01 \text{ ft (305 mm)}$$

To provide even distribution of incoming water flow, flow splitters are used to divide the water flow evenly to the split mains and branches.

Charging Process

During full-load operation in hot weather, charging is performed from 8 p.m. until noon the next day. All four chillers and all water pumps except the standby pumps are operated simultaneously to provide direct cooling during off-peak hours as well as the required stored cooling capacity during on-peak hours the next day.

The direct-cooling refrigeration load during off-peak hours varied from 1980 to 2600 tons (6962 to 9142 kW), with a total of 34,800 ton·h (122,357 kWh). The required stored cooling capacity to meet the refrigeration load during on-peak hours was 21,300 ton·h (74,891 kWh) on July 17, 1989. For a charging process of 16 h, the four chillers are operated at an average refrigeration load of about 3600 tons (12,658 kW). The set point of the chilled water temperature leaving the chiller is 39.5°F (4.2°C), and the outlet return chilled water temperature is around 56°F (13.3°C).

Before charging, the chillers and water pumps are started. During the charging process, the control valves CV-1 and CV-2 and solenoid valves SV-3, SV-5, SV-8, and SV-9 are opened, as shown in Fig. 31.10; and solenoid valves SV-4, SV-6, and SV-7 are closed. This provides both the charging of the storage tank and the supply of chilled water to building pumps during off-peak hours.

Return chilled water at around 56°F (13.3°C) is extracted from the stratified tank through the top diffusers and the control valve CV-1 and solenoid valve SV-3 by storage pumps SP-1 and SP-2. It flows through solenoid valves SV-5 and SV-9 and the chiller pumps, cools in the chiller, and leaves the chiller at 39.5°F (4.2°C). After that, the chilled water is divided into two streams. One of the streams flows through solenoid valve SV-8 and control valve CV-2 and is charged to the stratified tank through the bottom diffusers at 40°F (4.4°C). The other stream is extracted by the building pump and supplied to the air-handling units and terminals for direct cooling.

Discharging Process

Before the shutoff of the chillers and chiller pumps, the control valves and solenoid valves should be switched over to the followings: control valves CV-1 and CV-2 and solenoid valves SV-1, SV-2, SV-4, SV-6, and SV-7 should be open, and solenoid valves SV-3, SV-5, SV-8, and SV-9 should be closed.

Stored chilled water at 41°F(5°C) is extracted by storage pumps SP-2 and SP-3 via bottom diffusers and control valve CV-2. It then flows through SV-6, is extracted again by the building pumps, and is supplied to the cooling coils in the air-handling units and terminals. Return chilled water at a

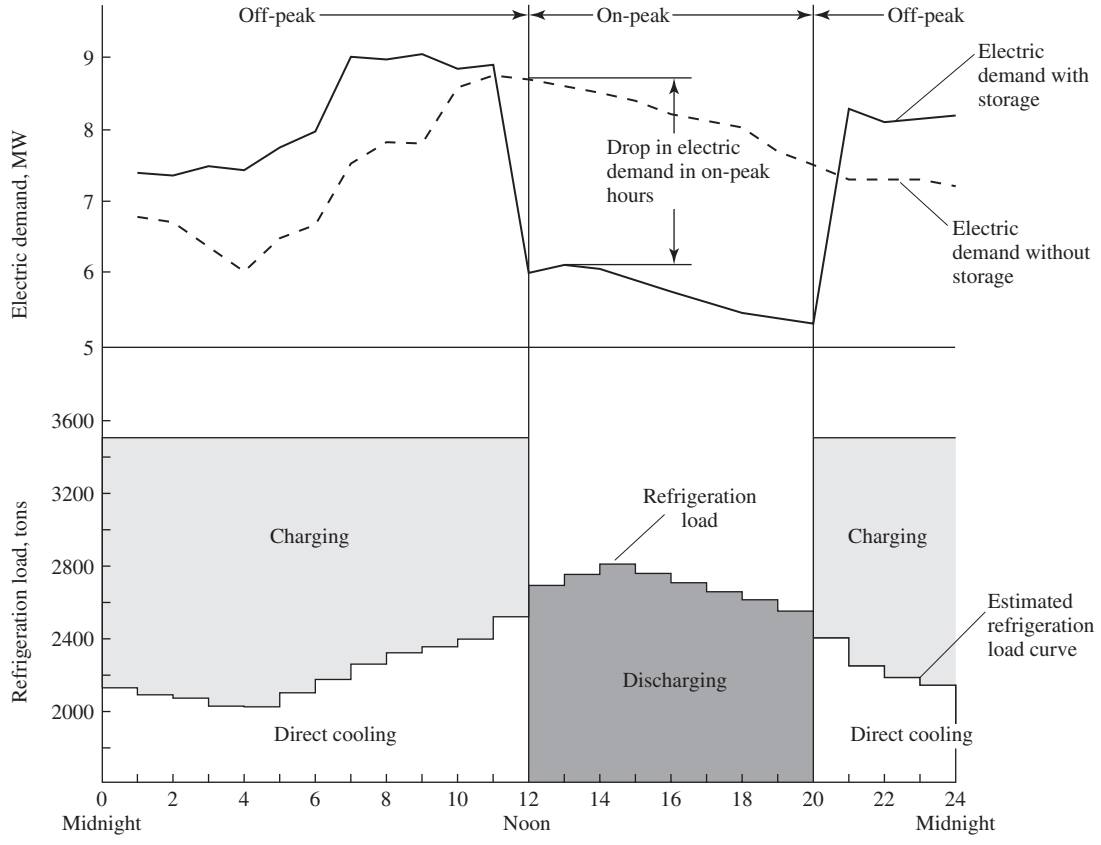


FIGURE 31.11 Electric demand curves and the storage cycle of the stratified chilled water storage system for an electronic manufacturing facility in Dallas, Texas.

temperature $T_{rc} = 56°F$ (13.3°C) is then forced through solenoid valve SV-7 and control valve CV-1 and introduced to the stratified tank through the top diffusers.

Temperature T_{rc} is maintained by changing the chilled water temperature supplied to the air-handling units from 45 to 52°F (7.2 to 11.1°C) instead of 41°F (5°C). As soon as temperature sensor T2 senses a return chilled water temperature T_{rc} drop below 56°F (13.3°C), the DDC controller closes the control valve CV-z slightly. Less stored chilled water is extracted by the building pumps. The required amount of return chilled water at 56°F (13.3°C) bypasses the crossover and mixes with the 41°F (5°C) stored chilled water, which results in a higher supply temperature to the cooling coils.

During on-peak hours, the discharging process requires a refrigeration load from 2500 to 2800 tons (8790 to 9845 kW), with a total of 21,250 ton·h (74,715 kWh) for the 8-h on-peak period.

Part-Load Operation

This electronic manufacturing facility includes clean rooms, computer rooms, compressed air aftercoolers, and manufacturing equipment, all of which need 24-h continuous cooling. Its daytime refrigeration load averages about 1240 tons (4360 kW).

When the refrigeration load is reduced or the entering temperature of condenser water drops because of a lower wet-bulb temperature, the following adjustments are made during part-load operation:

- The temperature of inlet water entering the stratified tank during the charging process T_i is raised from 39.5° to 42.5°F (4.2 to 5.8°C), which increases the chillers' capacity and lowers their power consumption. All four chillers and their auxiliary equipment do not have to operate simultaneously.
- Instead of 8 p.m. daily, the start of the charging process can be delayed until nearly all the stored chilled water in the stratified tank is discharged. The discharging process might last 10 to 14 h during part-load operation instead of 8 h in full-load operation.

System Performance

According to the operating cycle extended from August 24 to August 26, 1990, the electric demand dropped about 2.5 MW during on-peak hours as intended because four chillers and their corresponding auxiliary equipment were shut down during on-peak hours, as shown in Fig. 31.11. The maximum storage capacity of the stratified tanks was 27,643 ton·h (97,193 kWh). The difference between the average outlet temperature during discharging and the average inlet temperature during charging $T_o - T_i$ was 1.1°F (0.61°C), and the figure of merit was 92.2 percent.

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CHAPTER 32
COMMISSIONING AND MAINTENANCE

32.1 HVAC&R COMMISSIONING 32.1
- Commissioning and Design Intent 32.1
- Necessity of HVAC&R Commissioning 32.1
- Scope of HVAC&R Commissioning 32.2
- Testing, Adjusting, and Balancing (TAB) 32.3
- HVAC&R Commissioning Team 32.4
- When to Perform HVAC&R Commissioning 32.4
- Cost of HVAC&R Commissioning 32.5

32.2 HVAC&R MAINTENANCE 32.5
- Basics 32.5
- Maintenance Contractors and Maintenance Personnel 32.5
- Monitoring and Fault Detection and Diagnostics Assisting Predictive Maintenance 32.6

REFERENCES 32.6

32.1 HVAC&R COMMISSIONING

Commissioning and Design Intent

According to Wilkinson (1999a, 1999b), *commissioning* is defined as a systematic, documentated, and collaborative process that includes inspection, testing, and training conducted to confirm that a building and its associated serving systems are capable of being operated and maintained in conformance with the design intent. Design intent is the occupants' assumed intention of the design as well as their assumed operation of the building and the servicing systems. Design intent sets the requirements for occupant satisfaction and forms the foundation and basis of technical criteria and documents—design drawings and specifications. HVAC&R is one of the building servicing systems.

Wilkinson (1999a, 1999b) recommended that the design intent document include the following:

- General description of the building type and occupancy category
- Particular needs such as air cleanliness, and outdoor air volume flow rate, as well as processes that require special indoor environments
- Outdoor and indoor design conditions
- Space pressurization and relative pressurization of adjacent spaces
- Emergency operation during utility outages
- Applicable codes and fire/life safety requirements
- HVAC&R system selection if it is mandated by the owner
- Operations and maintenance manuals and staff training

Necessity of HVAC&R Commissioning

Air conditioning or HVAC&R systems in buildings are commissioned based on a sequence of planning, design, bidding, and construction. Many installed air conditioned systems do not work as

expected by the owners. Because of the higher demands of indoor environmental control, indoor air quality, and energy-efficient operation as well as the advances in technology in recent years, especially the microprocessor-based controls, an air conditioning system becomes more and more complicated and is required to provide considerably more functions than several decades ago.

Tseng (1998) reported on a survey of 60 commercial buildings; the results were presented in the 1994 National Conference on Building Commissioning sponsored by Portland Energy Conservation, Inc. This survey found the following:

- More than one-half of the buildings suffered control problems.
- Of the buildings 40 percent had HVAC&R equipment problems.
- Of the buildings 25 percent had energy management control systems, economizers, and variable-speed drives that did not run properly.

There are many of building owners and facility managers who claim that the functional performance of their HVAC&R systems does not meet their expectations.

ASHRAE/IESNA Standard 90.1-1999 mandates that HVAC control systems shall be tested to ensure that control elements are calibrated, adjusted, and in proper working condition. For projects larger than 50,000 ft^2 (4650 m^2) conditioned space area, except warehouses and semiheated spaces, detailed instructions for commissioning HVAC systems shall be provided by the designer in plans and specifications.

Portland Energy Conservation noted top deficiencies discovered by HVAC&R commissioning:

- Incorrect scheduling of HVAC&R
- Incorrect cooling and heating sequence of operation
- Incorrect calibration of sensors and instrumentation
- Lack of control strategies for optimum comfort and energy-efficient operations
- Improper air and water economizer operations
- Microprocessor-based DDC systems not fully utilized
- Short cycling of HVAC&R equipment
- Lack of design intent
- Lack of training of HVAC&R operators or service contractor for complex systems

Tseng (1998) cites the benefits of HVAC&R commissioning:

- HVAC&R commissioning is the quality control tool for building owners, and they know that the savings from fewer costly change orders and lower operating and maintenance costs will accrue.
- For those jobs with commissioning requirements, contractors and subcontractors know that it is not advisable for them to cut corners, and they benefit from lowering their cash set-asides for warranty reserves and callbacks.
- The architects can expect a building with far fewer postconstruction headaches to handle after commissioning.
- The engineers know that the HVAC&R systems are virtually assured of operating as intended, thus eliminating postoccupancy troubleshooting visits.
- The building occupants enjoy a higher-performing building with a smoothly functioning HVAC&R system which provides a healthy and comfortable indoor environment with the benefits of increased productivity.

Scope of HVAC&R Commissioning

Tseng (1998) and Ellis (1998) described the scope of HVAC&R commissioning as follows:

- Clarifly owner priorities and design intent. The commissioning outcomes include a healthy and comfortable indoor environment, acceptable indoor air quality, energy efficiency, optimized functional performance, maintainability, and constructibility. Constructibility is the ability of a project to be built as designed.
- Document and monitor all facets of construction. Documentation includes preparation of the commissioning specification, preparation of the verification test procedures, and review of key equipment and instrumentation submittals. Monitoring includes preparation of system readiness checklists such as electricity to the fans, pumps, and compressors; monitoring and verifying equipment/system start-up and operation as well as the performance of temperature and EMCS controls; and testing, adjusting, and balancing (TAB) work.
- Verify TAB work. In HVAC&R commissioning, TAB is a tool and is a part of the commissioning. It is often provided by a TAB subcontractor.
- Extensively test all subsystems and their components and controls (acceptance commissioning). Verify and document functional performance testing of all systems including control systems, so that all systems comply with the contract documents. Establish an as-delivered performance record, and verify the as-built record for all systems. The documentation should also include the corrective action reports for every deficiency found, the follow correction and retest.

  For acceptance, the final commissioning report should be completed and submitted, and thus, the recommendation of acceptance determined. All documents should be turned over to the owner.
- Provide detailed training of operating personnel for each major system and equipment. Training plans and attendance records should be supervised. It is important to conduct system training as well as its operation so that all the individual components are functioning together as a system. System operators cannot be expected to operate the system properly unless they know how it is intended to work and why. If possible, operation and maintenance staff are encouraged to participate in the verification tests as an excellent training.
- Continue post-acceptance commissioning and ongoing monitoring. Post-acceptance commissioning and ongoing monitoring are a continuation of adjustment, optimization, and improvement of HVAC&R systems to meet a specified goal by monitoring specific system operating parameters. Data trend and diagnostic capabilities of DDC control systems are valuable during HVAC&R commissioning and post-acceptance commissioning.

Testing, Adjusting, and Balancing (TAB)

Testing means to determine quantitative performance of equipment. Adjusting denotes that fluid flow rate and air flow pattern are adjusted as specified in design. Balancing indicates that flows are adjusted in proportion within the distribution system based on design requirements.

ASHRAE/IENSA Standard 90.1-1999 mandates that construction documents shall require that all HVAC systems be balanced in accordance with generally accepted engineering standards. Ducted air and water flow rates shall be measured and adjusted within 10 percent of design flow rates. Variable-speed, variable-volume flow distribution systems need not be balanced upstream of a pressure-independent control device. A written balance report shall be provided to the owner for HVAC&R systems serving zones with a total conditioned area exceeding 5000 ft^2 (465 m^2). Standard 90.1-1999 also mandates that air systems shall be balanced first to minimize throttling losses. Then, in systems with fans of power greater than 1 hp (0.75 kW), fan speed shall be adjusted to meet design flow conditions, except variable-flow distribution systems need not be balanced upstream of the controlling device (such as a calibrated VAV box).

Standard 90.1-1999 also mandates that hydronic (water) systems shall be proportionally balanced first to minimize throttling losses, and then the pump impeller shall be trimmed or pump speed shall be adjusted to meet design volume flow requirements. Each water system shall have facility either to measure pressure increase across the pump or have test ports on each side of each

pump. Exceptions include pumps with motors of 10 hp (7.5 kW) or less or when throttling losses are no greater than 5 percent of the power draw or 3 hp (2.3 kW), whichever is greater, above that required after the impeller is trimmed.

HVAC&R Commissioning Team

An HVAC&R commissioning team should be organized to perform commissioning. According to Tseng (1998), this team should include the following members and perform the duties listed below:

Owner. The owner should be actively involved in commissioning, such as by defining requirements and assigning commissioning responsibilities, defining the scope and tasks for the commissioning authority, and developing enforceable contractual provisions to ensure compliance by the contractor.

Commissioning Authority or Comission Consultant (CC). The commissioning authority or commission consultant should have specific duties including reviewing the plans and specifications that related to the commissioning processes, scheduling and conducting all pertinent commissioning activities, such as precommissioning activities, training, walk-through inspections, tests and balances, and documentation. The commission consultant plays a key role in an HVAC&R commissioning. For a large HVAC&R project, an outside expert can be hired as an independent CC. For a small HVAC&R project, the owner, the engineer, the general contractor, or the principal mechanical subcontractor may serve as the CC.

Architect. The architect ensures that the commissioning authority reviews the shop drawing submittal.

Engineer. The engineer must be actively included in the commissioning activities: design intent documentation, verification testing, and training.

General Contractor. The general contractor must include the cost for commissioning requirements in the bid price and ensure that such requirements in the mechanical and electrical subcontracts are complied with.

Mechanical/Electrical Subcontractor. The mechanical/electrical subcontractor must also include commissioning requirements in the contract price, ensure participation of subcontractors, coordinate all testing with pertinent specialty subcontractors, conduct walk-through inspections and hands-on trainings, provide certification of system performance and functional performance of major equipment to the CC, and turn over a complete set of as-built drawings to the design engineer.

Test, Adjusting, and Balancing Contractor. The TAB contractor conducts TAB work, demonstrating to the CC the tested performance and participating in training sessions.

Energy Management and Control System (EMCS) Contractor. A properly installed and debugged EMCS system is essential for HVAC&R commissioning to succeed.

When to Perform HVAC&R Commissioning

Ellis (1998) recommended three kinds of HVAC&R commissioning processes:

First, design-through-occupancy commissioning is ideal for the HVAC&R commissioning process. The CC serves as the owner's technical liaison throughout the design, construction, and start-up of the project. The CC reviews the design engineer's plans for compliance with the design

intent. The CC also makes sure that during verification testing, the temperature, pressure, fluid flow, control system signals, and actions are commissionable and included in the design.

Second, postconstruction commissioning is most prevalent recently. Postconstruction commissioning often requires at least 1 year after the completion of the construction so that both summer cooling and winter heating mode operations can be included. In this case, the commission consultant is most often an outsider who was not involved in the problems and politics of the design and construction. Therefore, conciliatory and professional interpersonal skills are extremely important for a CC. The CC also will learn about, evaluate, test, and document the installed HVAC&R system and become involved in the operators' training.

Third, existing building recommissioning helps to reduce HVAC&R energy consumption, improves IAQ, and improves indoor environmental control.

Cost of HVAC&R Commissioning

Because the HVAC&R commissioning process is still being developed and standardized, a rule of thumb for HVAC&R commissioning cost is between 2 and 5 percent of the construction cost of the systems to be commissioned. The cost of HVAC&R commissioning can be set aside as an independent budget item or added to the architectural/engineering or contractor fees. The commissioning cost will be recovered by reductions in change orders and claims.

32.2 HVAC&R MAINTENANCE

Basics

HVAC&R maintenance is the work required to maintain or restore HVAC&R systems, including equipment, instrumentation, components, and materials, to condition such that they can be effectively operated to meet specified requirements.

ASHRAE Handbook 1999, HVAC Applications, defines *repair* as to make good or restore to good or sound conditions, and defines *service* as what is necessary to effect a maintenance program short of repair. In the ASHRAE handbook, maintenance also is classified into the following categories:

- Run-to-failure is a kind of arrangement such that no money is spent on maintenance prior to equipment or system breakdown.
- In planned maintenance, all functions and resources in this category must be planned, budgeted, and scheduled. Planned maintenance can be subdivided into preventive and corrective maintenance. Preventive maintenance is a kind of scheduled maintenance for an HVAC&R system, equipment, or components, in order to maintain durability, reliability, efficiency, and safety. In corrective maintenance, corrective action is often the remedial action performed before failure occurs. Corrective action taken during a shutdown in response to a failure is called a repair.
- Predictive maintenance is based on equipment and system monitoring, the operating conditions, and performance to discover faults and degradations and thus the remedy performed.

Most manufacturers give detailed instructions in how to operate and maintain the specific equipment and instruments. These instructions must be followed.

Maintenance Contractors and Maintenance Personnel

In most small facilities with comparatively simple HVAC&R systems, outside maintenance contractors often provide maintenance service based on the specified maintenance program. The time inter-

val between two maintenance programs depends on the operating hours of the HVAC&R system; dirt accumulation in the filter, coil, and distribution devices; and recalibration requirements of the instruments. For small and complicated HVAC&R systems, whenever the operator cannot repair and service the HVAC&R system, the owner or the facility manager should ensure that qualified contractors are hired for maintenance.

For more complicated medium-size multizone HVAC&R systems, operating personnel are often responsible for HVAC&R system maintenance. The maintenance programs should be detailed in the operation and maintenance manual and tailored to each specific building. For highly technical equipment and systems, an outside maintanance contractor with the specific expertise required should be called for service.

For large HVAC&R systems with central plants, a management organization with one operations person and one maintenance person is required. Computerized maintenance programs should be used to provide detailed timing of system maintenance procedures. Logged information and proper data management associated with predictive maintenance will reduce failure response requirements. An outside maintenance service contractor with specific expertise may need to be called.

Monitoring and Fault Detection and Diagnostics Assisting Predictive Maintenance

As discussed in Sec. 5.15, in EMCS and fault detection and diagnostics, many operating parameters of the HVAC&R system are monitored and measured. Addition of necessary monitoring parameters for predictive maintenance may be considered.

Faults occur when the actually measured parameters deviate from the normal operating values. Performance degradation is an evolving fault accumulated during a certain period of time.

By using rule-based expert systems, autoregressive with exogenous inputs (ARX) models, and artificial neural network (ANN) models, performance degradation and faults can be determined, and the corrective maintenance action recommended.

HVAC&R maintenance is already an important part of optimizing an energy-efficient HVAC&R system operation, to improve IAQ, and to maintain a safe, healthy, and comfortable indoor environment. More and more computer software will be developed to combine optimizing, energy-efficient HVAC&R system operation and predictive maintenance in the future.

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APPENDIX A
NOMENCLATURE AND ABBREVIATIONS

A.1 NOMENCLATURE

A	area, ft^2
a	turbulence factor
A_a	face area, ft^2
A_c	core area of outlet, ft^2
A_D	Dubois surface area of naked body, ft^2
$A_{e,l}$	effective leakage area, ft^2
A_f	area of fins, ft^2
A_i	inner surface area, ft^2
A_k	net or unobstructed area of grille, ft^2
A_o	outer surface area, ft^2
A_p	primary surface area, ft^2
A_r	cross-sectional area of room perpendicular to airflow, ft^2
Ar	Archimedes' number
AS	annual savings, dollars
B	ratio of outer surface area to inner surface area A_o/A_i
Bi	Biot number
C	scale factor; concentration of air contaminant, mg/m^3; local loss coefficient; cost, dollars
C_{cc}	cloudy-cover factor
C_d	discharge coefficient, degradation coefficient
C_{di}	duct installation cost, dollars
C_e	energy cost, dollars
C_{cir}	circulating factor
C_{iu}	unit cost of duct installation, \$/ft^2
C_n	clearness number of sky
C_o	local loss coefficient
c_p	specific heat at constant pressure, Btu/lb · °F
c_{pa}	specific heat of moist air at constant pressure, Btu/lb·°F
c_{pd}	specific heat of dry air at constant pressure, Btu/lb · °F
c_{pr}	specific heat of liquid refrigerant at constant pressure, Btu/lb · °F
C_{pre}	pressure loss coefficient
c_{ps}	specific heat of water vapor at constant pressure, Btu/lb · °F
c_{pw}	specific heat of water at constant pressure, Btu/lb · °F
c_{sat}	saturation-specific heat per degree wet-bulb temperature, Btu/lb · °F
$C_{s,i}$	inlet system effect loss coefficient
$C_{s,o}$	outlet system effect loss coefficient
C_i	scale factor for temperature lines, °F/ft
C_{to}	total cost, dollars
C_v	flow coefficient, a flow rate of 1 gpm at a pressure drop of 1 psi
C_w	scale factor for humidity ratio lines, lb/lb · ft
$C + R$	convective and radiative heat loss, Btu/h · ft^2
CDD	cooling degree-day, degree-day
COP	coefficient of performance
COP_c	coefficient of performance of chiller
COP_{hp}	coefficient of performance of heat pump

COP_{hr}	coefficient of performance of heat recovery
COP_{ref}	coefficient of performance of refrigeration
CRF	capital recovery factor
D	diameter, in. or ft; depreciation, dollars per year
D_{aw}	mass diffusivity for water vapor through air, ft^2/s
D_e	equivalent diameter, in. or ft
D_h	hydraulic diameter, in. or ft
D_{lv}	mass diffusivity of liquid and vapor, ft^2/h
D_T	mass diffusivity due to temperature gradient, ft^2/h
E	evaporative heat loss, $Btu/h \cdot ft^2$; electric potential, Volt; efficiency
e	energy, Btu/b
E_{dif}	evaporative heat loss due to direct diffusion, $Btu/h \cdot ft^2$
E_{max}	maximum level of evaporative heat loss, $Btu/h \cdot ft^2$
E_r	unit energy cost, $/kWh
E_{rsw}	sweating due to thermoregulatory mechanism $Btu/h \cdot ft^2$
EER	energy efficiency ratio, $Btu/h \cdot W$
ET*	effective temperature, °F
F	factor, shape factor, solar heat gain factor
f	friction factor; frequency, Hz
$F_{a,t}$	air transport factor
F_{block}	blockage factor
F_{cl}	clothing efficiency
F_{cyc}	cycling loss factor
F_p	performance factor
F_{rej}, HRF	heat rejection factor
F_s	coil core surface area parameter
F_t	fin thickness, in.; fan total pressure, in. WC
Fo	Fourier number
Fr	Froude number
G	mass velocity, $lb/h \cdot ft^2$
g	gravitational acceleration, ft/s^2
g_c	dimentional conversion factor, $32\ lb_m \cdot ft/lb_f \cdot s^2$
Gr	Grashof number
H	head, ft of water column; height, ft; hour angle, deg
h	enthalpy, Btu/lb; heat-transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
h_b	boiling heat-transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
h_{con}	condensing heat transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
H_f	frictional head loss, ft of water
h_{ig}	latent heat of vaporization, Btu/lb
$h_{fg,r}$	latent heat of vaporization of refrigerant, Btu/lb
$h_{fg,32}$	latent heat of vaporization at 32°F, Btu/lb
h_i	inner surface heat-transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$
h_m	convective mass-transfer coefficient, ft/h
h_o	heat-transfer coefficient at outer surface, $Btu/h \cdot ft^2 \cdot °F$; outdoor air enthalpy, Btu/lb
h_r	enthalpy of space air
h_{ru}	enthalpy of recirculating air, Btu/lb
H_s	static head, ft of water
h_s	enthalpy of saturated air film, Btu/lb
h	enthalpy of supply air
$\Delta H_{s,r}$	head difference between supply and return mains, ft WC
$\Delta h_{s,r}$	enthalpy of saturated air film at evaporating temperature, Btu/lb
H_t	total head, ft of water
H_v	velocity head, ft of water
h_{wet}	heat-transfer coefficient of wetted surface, $Btu/h \cdot ft^2 \cdot °F$
HC	heating capacity, $Btu/ft^2 \cdot °F$
HDD	heating degree-day, degree-day
I	electric current, amp
i	interest rate, percent
I_D	direct radiation, $Btu/h \cdot ft^2$
I_d	diffuse radiation, $Btu/h \cdot ft^2$
I_G	global radiation on a horizontal plane, $Btu/h \cdot ft^2$

I_{DN}	solar radiation on a surface normal to sun rays, Btu/h · ft^2
i_m	moisture permeability of clothing
I_a	extraterrestrial solar intensity, Btu/h · ft^2
I_{rad}	effective radiant field, W/ft^2
I_{ref}	reflection of solar radiation, Btu/h · ft^2
I_t	total intensity of solar radiation, Btu/h · ft^2
I_{tur}	intensity of turbulence
I_{sc}	solar constant, 434.6 Btu/h · ft^2
IC	initial cost, dollars
IL	insertion loss, dB
J	Joule's equivalent, 778 ft · lb$_f$/Btu
j	cost escalation factor
K	constant, factor, coefficient, derivative gain
k	thermal conductivity, Btu · in./h · ft^2 · °F, Btu/h · ft · °F
K'	wet-bulb temperature constant
K_{cc}	cloudy reduction factor
K_i	integral gain
K_p	power constant
K_p	proportional gain, pressure constant
K_V	volume constant
L	distance, thickness, ft; sound level, dB; latitude angle, deg
L_{af}	airflow noise, dB
L_e	equivalent length, ft
L_w	sound power level, dB re 1 pW
L_p	sound pressure level, dB re 20 μPa
L_{pA}	sound pressure level in dBA
L_{pr}	room sound pressure level, dB
L_w	vertical distance between state points, ft
L_t	horizontal distance between state points, ft
$L_{w,b}$	branch power division, dB
L_{wr}	room sound power level, dB
Le	Lewis number
LHG	latent heat gain, Btu/h
LR	Lewis relation
M	molecular weight; metabolic rate, Btu/h · ft^2
m	mass, lb
m''	slope of air enthalpy saturation curve
$\dot{m}$	mass flow rate, lb/min, lb/h
$\dot{m}_{par}$	rate of air contaminants generated, mg/s
$\dot{m}_r$	mass flow rate of refrigerant, lb/h, lb/min
m_s	surface density, lb/ft^2
N	number
n	number of moles, mol; number of air changes, ach; circulation number; amortization period, year
n'	depreciation period, year
N_r	number of rows
NTU	number of transfer units
Nu	Nusselt number
OC	operating cost, dollars
P	power, hp, kW; perimeter, ft; penetration
p	pressure, psi, psia, psig
P_{air}	air power, hp
p_{at}	atmospheric pressure, psia
P_{cfm}	power per unit volume flow, W/cfm
P_{com}	compressor power, hp
p_{con}	condensing pressure, psig
p_{dis}	discharge pressure of compressor, psig
Δp_{dy}	dynamic loss, in WC
p_{ev}	evaporating pressure, psig
P_f	fan power, input, hp
Δp_f	pressure drop due to frictional and dynamic losses, in. WC
p_{fill}	fill pressure, abs psia
Δp_{fix}	fixed part of system pressure loss, in. WC
$\Delta p_{f,s}$	frictional loss per floor inside the pressurized stairwell, in. WC

$\Delta p_{f,u}$	duct frictional per unit length, in. WC
P_H	horizontal projection, ft
P_{in}	power input, hp
Δp_{od}	total pressure loss of damper when fully open, in. WC
P_p	pump power, hp
$\Delta p_{p\text{-}od}$	total pressure loss of airflow path excluding damper, in. WC
Δp_{res}	residual pressure, in. WC
p_s	static pressure, in WG, psig
$\Delta p_{s,i}$	inlet system effect pressure loss in. WC
$\Delta p_{s,o}$	outlet system effect pressure loss, in. WC
$\Delta p_{s,r}$	static regain, in. WC
p_{st}	pressure due to stack effect, $\text{lb}_\text{f}/\text{ft}^2$
p_{suc}	suction pressure of compressor, psig
Δp_{sy}	system pressure loss, in. WC
Δp_t	total pressure, in. WC
$\Delta p_{\text{t,ex}}$	external total pressure, in. WC
P_V	vertical projection, ft
p_v	velocity pressure, in. WC
Δp_{var}	variable part of system pressure loss, in. WC
p_{vo}	velocity pressure at outlet, in. WC
p_{vw}	wind velocity pressure, in. WC
p_w	water vapor pressure, psia
PMV	predicted mean vote
Pr	Prandtl number
Q	rate of heat transfer, Btu/h
q	rate of heat tansfer, Btu/h
Q_c	coil's load, Btu/h
Q_{cc}	cooling coil's load, Btu/h
$Q_{c,c}$	corrected cooling capacity, Btu/h
$Q_{c,r}$	catalog-listed cooling capacity, Btu/h
$q_{c,wet}$	heat and mass transfer, Btu/h
Q_{ev}, Q_{ref}, Q_{rl}	refrigeration load at evaporator, Btu/h
q_{lg}	heat input to the first-stage generator, Btu/h · ton
Q_{ch}	heating coil load, Btu/h
q_{int}	internal heat gain, Btu/h
Q_{rc}	space cooling load, Btu/h
$Q_{\text{rej,}}$ THR	total heat rejection, Btu/h
Q_{rh}	space heating load, Btu/h
$q_{\text{RC}i}$	inward heat flow from inner surface of sunlit window, Btu/h
Q_{rs}	space sensible cooling load, Btu/h
Q_{rsp}	space sensible cooling load at part load, Btu/h
$q_{rs,t}$	sensible cooling load at time t, Btu/h
q_{sen}	rate of sensible heat transfer, Btu/h
q_{tran}	tramission loss, Btu/h
q_{wi}	heat gain admitted into conditioned space, Btu/h
R	gas constant, $\text{ft} \cdot \text{lb}_f/\text{lb}_m \cdot °\text{R}$; electric resistance, Ω
R	R-value, $\text{h} \cdot \text{ft}^2 \cdot °\text{F/Btu}$
R^*	thermal resistance, $\text{h} \cdot °\text{F/Btu}$; flow resistance, in $\text{WC}/(\text{cfm})^2$; ratio
R_c	radius of curvature, in. or ft
R_{cl}	thermal resistance of clothing, $\text{h} \cdot \text{ft}^2 \cdot °\text{F/Btu}$
R_{com}	compression ratio
R_{en}	entrainment ratio
R_f	fouling factor, $\text{h} \cdot \text{ft}^2 \cdot °\text{F/Btu}$
R_{fa}	ratio of free area to gross area
R_g	gas-side thermal resistance, $\text{h} \cdot \text{ft}^2 \cdot °\text{F/Btu}$
R_{load}, LR	load ratio
R_o	universal gas constant $\text{ft} \cdot \text{lb}_f/\text{lb}_m \cdot °\text{R}$
$R_{T,l}$	ratio of temperature lift
Re	Reynolds number
ROR	rate of return
S	salvage value, dollars; heat storage, $\text{Btu/h} \cdot \text{ft}^2$
s	specific entropy, $\text{Btu/lb} \cdot °\text{R}$; dimensionless distance
S_f	fin spacing, fins/in.
S_H	shaded height, ft
S_W	shaded width, ft

SC	shadding coefficient
S_c	Schmidt number
SHG	sensible gain, Btu/h
SHGF	solar heat gain factor, Btu/h · ft^2
SHR	sensible heat ratio
SHR_c	sensible heat ratio of cooling and dehumidification process
SHR_s	sensible heat ratio of space conditioning line
SHR_{sp}	sensible heat ratio of space conditioning line at part load
SP	simple payback, years
St	Strouhal number
T	temperature, °F
T'	wet-bulb temperature, °F
T^*	thermodynamic wet-bulb temperature, °F
T_∞	bulky air temperature unaffected by surface, °F
t_{an}	annual operating hours, h
T_{co}	changeover temperature, °F
T_{dew}, T''	dew point temperature, °F
T_{dis}	discharge temperature, °F
ΔT_f	fan temperature rise, °F
T_{en}	mass temperature of building envelope, °F
T_m	temperature of mixture, °F
ΔT_m	log-mean temperature difference, °F
T_o	operative temperature, °F; outdoor temperature, °F
T'_{os}	summer mean coincident wet-bulb temp, °F
$T_{o,\,ws}$	statistically determined winter design outdoor temperature, °F
T_p	plenum air temperature, °F
T_R	absolute temperature, °R
T_r	space temperature, °F
T_{ra}, T_{rad}	mean radiant temperature, °F
T_{rm}	average space temperature, °F
T_{rp}	space temperature at part load, °F
T_{rp}	space temperature in perimeter zone, °F
T_{ru}	temperature of recirculating air, °F
T_s	supply temperature, °F
ΔT_{sa}	temperature difference between the surface and air, °F
$\Delta T_{t,r}$	throttling range, °F
T_{ws}	chilled water supply temperature, °F
TD	temperature differential, °F
TL	transmission loss, dB
TL_{in}	break-in transmission loss, dB
TL_{out}	breakout transmission loss, dB
U	overall heat transfer coefficient, Btu/h · ft^2 · °F
u	internal energy, Btu/lb; peripheral velocity, fpm
U_i	overall heat-transfer coefficient based on inner surface area, Btu/h · ft^2 · °F
U_u	overall heat-transfer coefficient based on outer surface area, Btu/h · ft^2 · °F
UAC	uniform annual cost, dollars
V	volume, ft^3
$\dot{V}$	volume flow rate, cfm
v	velocity, fpm of ft/s^v specific volume or moist volume, ft^3/lb
v_c	centerline velocity, fpm
v_{con}	air velocity in constricted part of damper or duct fittings, fpm
$\dot{V}_{conv}$	volume flow rate of upward convective flow, cfm
$\dot{V}_{ef}$	volume flow rate of exfiltrated air, cfm
v_{fc}	face velocity, fpm
$\dot{V}_{gal}$	volume flow rate of chilled water, gpm
$\dot{V}_{lk}$,$\dot{V}_L$	volume flow rate of air leakage, cfm
$\dot{V}_o$	volume flow rate of outdoor air, cfm
$\dot{V}_{o,dm}$	minimum outdoor air supply volume flow at design conditions, cfm
$\dot{V}_{oif}$	volume flow rate of outdoor and infiltrated air, cfm
$\dot{V}_{osn}$	zone outdoor air supply volume flow rate, cfm

$\dot{V}_p$	piston displacement, cfm
$\dot{V}_s$	supply volume flow rate, cfm
$\dot{V}_{sp}$	supply volume flow rate at part load, cfm
W	work, Btu/lb; mechanical work performed, Btu/h·ft^2; sound power, dB; width, ft
w	relative velocity, fpm; humidity ratio, lb/lb
W_{in}	work input, Btu/lb
W_{isen}	isentropic work, Btu/lb
W_{rsw}	wetted portion of human body due to sweating
w_s^*	saturated humidity ratio at thermodynamic wet-bulb temperature, lb/lb
X	moisture content, dimensionless or percent; mass fraction
x	mole fraction; quality or dryness fraction; coordinate dimension
x_{rl}	quality or refrigerant leaving overfeed cooler
y	vertical drop of air jet, ft
Z	compressibility factor
z	elevation, ft
z_{stat}	stationary level, ft

Subscripts

a	air, ambient, absorber
ab	absorber, air at dry-wet boundary
ae, aen	entering air
alv	leaving air
am	ambient
at	atmospheric
av	average
b	body bleed, branch, building material
bg	building
by	bypass
c	coil, cooling, cold, convective, condenser, compressor, common end, corrected
ca	cooling air, cooled air
cc	cooling coil
c,d	closed door
ce	entering condenser
hc	heating coil
cl	clothing, cooling load, cooling coil, leaving condenser
cn	common
co	changeover
com	compressor
con	condensing, condenser
corr	correction
cr	body core
cs	sensible cooling coil
d	duct, design
dam	damper
deh	dehumidifier
des	desiccant
dif	diffuser
dis	discharge
dl	process air leaving desiccant dehumidifier
dy	dynamic
e	entering
ee	entering evaporator
ef	exfiltrated
eff	effective
el	elevation, leaving evaporator, equivalent
en	entering
en,*c*	cooling water entering condenser
ev	evaporating, evaporator
ev,*c*	evaporative cooling
ex, exh	exhaust
exf, ef	exfiltration
f	fan
fc	fan coil
fix	fixed part
fl	floor
fu	furnace
g	moisture gain, gas, globe, ground, generator
1*g*	first-stage generator

2*g*	second-stage generator
go	saturated water vapor at 0°F
h	higher, heat exchanger, heating, hot
hg	heat gain
h,t	heat transformer
hu	humidifier
i	inlet, input, indoor, interior, intermediate, inner surface
in	input, indirect, interior
inf	infiltration
int	intermediate
k	conduction
l	latent, liquid, lower, lights, leaving
lc	liquid at condenser
le	water entering evaporator
liq	liquid
lk	leakage
lr	liquid refrigerant
lv	leaving
m	mixture, mean, maximum, motor
mat	material
max	maximum
min	minimum
mo, mot	motor
n	number
o	outdoor, output, outlet, outer surface, oversaturation
o,d	open door
oi	inward flow from outer surface
os	outer surface
o,s	summer outdoor
*o,*sys	system outdoor air
out	outdoor air
p	constant pressure, people, part-load, pump, process air, primary air
par	particulates in air
pd	dry air at constant pressure
pl	process air after sensible heat exchanger
ps	water vapor at constant pressure, primary air supply
pt	plant
r	space, room, return, refrigerant, radiative, regeneration air
rc	space cooling, refrigeration capacity
rd	return duct
rec	recirculating
ref	refrigeration
ref·f	free refrigeration
rel	release
res	residual, respiration
ret, rt	return
rf	return fan, relief fan, refrigeration effect
rg	regeneration air entering desiccant dehumidifier
rh	space heating
rl	space latent
ro	regeneration air leaving desiccant dehumidifier
rp	return plenum
rs	space sensible
r·s	return system
ru	recirculating air entering the AHU or PU
s	supply, steam, saturated state, surface, sunlit, straight-through end, summer
s	saturated at thermodynamic wet-bulb temperature
sa	solution at absorber
s·a	saturated air film
sat	saturation
sb	surface at dry-wet boundary
sc	subcooled
s·c	cold air supply
sd	supply duct
sen	sensible
sf	supply fan
sg	solution at generator
sh	shaded

s·h	warm air supply
si	supply air for interior zone
sil	silencer
sk	skin
sn	supply air for zone *n*
sn,d	zone supply air at design condition
sn,p	supply air for zone *n* at part load
sol	solar
s·s	supply system
s·t	supply temperature
suc	suction
sun	sunlit
sur	surroundings
sx	supply air for perimeter zone
T	total, overall, temperature
t	time
un	unconditioned
var	variable part
ve	saturated water vapor leaving evaporator
w	water, condensate, winter
wb	water at dry-wet boundary, boiler hot water
we	water entering
wet	wet air
wl	water leaving
w,o	winter outdoor
ws	water vapor at saturated state
xl	regeneration air leaving the sensible heat exchanger, exit of heat exchanger

Greek Letter Symbols

α	mean temperature coefficient; angle between the air conditioning process and horizontal line on psychrometric chart, deg; thermal diffusivity, ft^2/s; absorptance; spreading angle of air jet, deg; damper characteristic ratio
β	solar altitude angle, deg; blade angle, deg
γ	ratio of specific heat at constant pressure to constant volume, surface-solar azimuth angle, deg
Δ	difference
δ	solar declination angle, deg
ϵ	emissivity; absolute roughness, in.; effectiveness; effectiveness factor
ϵ_{ex}	air exchange efficiency
ϵ_{sat}	saturation effectiveness
ϵ_{wet}	wet coil effectiveness
η	efficiency
η_{cb}	combustion efficiency
η_{com}	compressor efficiency
η_{cp}	compression efficiency
η_{dr}	efficiency of driving mechanism
η_f	fan total efficiency; fin efficiency
η_{fu}	furnace efficiency
η_{isen}	isentropic efficiency
η_{mec}	mechanical efficiency
η_{mo}, η_{mot}	motor efficiency
η_{ov}	overall efficiency
η_p	pump efficiency
η_s	fan static efficiency, fin surface efficiency
η_{sat}	saturation effectiveness
$\eta_{sy,h}$	system efficiency for heating
η_t	fan total efficiency
η_v	volumetric efficiency
η_{ww}	wire-to-water efficiency
θ	angle of incidence, deg; effective draft temperature, °F
μ	absolute viscosity, $lb/ft \cdot s$; degree of saturation; mechanical efficiency
ν	kinematic viscosity, ft^2/s
ρ	density, lb/ft^3; reflectance
ρ_{suc}	density of suction vapor, lb/ft^3
Σ	angle between tilting and horizontal surface, deg
σ	Stefan-Boltzmann constant, 0.1714×10^{-8} $Btu/h \cdot ft^2 \cdot °R^4$; standard deviation
τ	transmittance

φ	relative humidity, percent; solar azimuth angle, deg
φ_f	fin resistance number
ψ	surface azimuth angle, deg
Ω	profile angle, deg

A.2 ABBREVIATIONS

ABMA	American Boiler Manufacturers Association
abs.	absolute
AC	air conditioning
ACEC	American Consulting Engineers Council
ACI	adjustable current inverter
ACP	alternate component package
ADC	Air Diffusion Council
ADPI	air diffusion performance index, percent
AFUE	annual fuel utilization efficiency
AHU	air-handling unit
AI	analog input
AMCA	Air Movement and Control Association
ANSI	American National Standards Institute
AO	analog output
ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
ASME	American Society of Mechanical Engineers
ASTM	American Society of Testing and Materials
AVI	adjustable voltage inverter
BAS	building automation system
BHP, bhp	brake horsepower
BI	binary or digital input, backward-inclined blade
BLAST	building loads analysis and systems thermodynamics
BMS	building management system
BO	binary or pulsed output
BOCA	Building Officials and Code Administrators
CABDS	computer-aided building design system
CADD	computer-aided design and drafting
CFC	chlorofluorocarbon
CLF	cooling load factor
CLTD	cooling load temperature difference
COP	coefficient of performance
CTD	condenser temperature difference
DA	direct-acting
dc	direct current
DDC	direct digital control
DECOS	design energy cost
DOE	Department of Energy
DOP	di-octyl phthalate
DSA	double-strength sheet glass
DWDI	double-width double-inlet
DX	direct expansion, dry expansion
ECB	energy cost budget
EEPROM	electrically erasable, programmable, read-only memory
EIA	Energy Information Administration of the Department of Energy
EMS	energy management system
EPA	Environmental Protection Agency
EPROM	erasable programmable read-only memory
ETD	equivalent temperature difference
FC	fan coil, forward-curved blade
FDA	Food and Drug Administration
FOM	figure of merit
GWHP	groundwater heat pump
GWP	global warming potential
HEPA	high-efficiency particulate air
HR	heart rate
HSPF	heating seasonal performance factor

HVAC&R	heating, ventilating, air conditioning, and refrigeration
IAQ	indoor air quality
ILD	internal load density
I/O	input/output
I-P	inch-pound
IPLV	integrated part-load value
IRS	Internal Revenue Service
LiBr	lithium bromide
LiCl	lithium chloride
LPG	liquefied petroleum gas
MAU	makeup air unit
MCPC	microcomputer constructed psychrometric chart
MPS	manual position switch
NBC	National Broadcasting Corporation
NC	noise criteria, normally closed
NCDC	National Climatic Data Center
NFPA	National Fire Protection Association
NIOSH	National Institute of Occupational Safety and Health
NO	normally open
NPL	neutral pressure level
NPSH	net positive suction head
NWWA	National Water Well Association
ODP	ozone depletion potential
PC	personal computer
PI	proportional plus integral
PID	proportional-integral-derivative
PU	packaged unit
PTAC	packaged terminal air conditioner
PURPA	Public Utility Regulatory Policies Act
PVC	polyvinyl chloride
PWM	pulse-width modulated inverter
RA	reverse-acting
RAM	random access memory
RAU	recirculating air unit
RC	room criteria
RH	relative humidity
ROM	read-only memory
RTD	resistance temperature detector
RTS	room temperature sensor
SBS	sick building syndrome
SEER	seasonal energy efficiency ratio
SEUF	seasonal energy utilization factor
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
SI	International System of units
SPF	seasonal performance factor
SSE	steady-state efficiency
SSU	Saybolt-seconds univeral viscosity
SWSI	single-width single-inlet
TA	time-averaging
TARP	thermal analysis research program
TETD	total equivalent temperature differential
TFM	transfer function method
TIMA	Thermal Insulation Manufacturers Association
TRAV	terminal regulated air volume
UL	Underwriters' Laboratories
ULPA	ultra low-penetration air filters
VAV	variable air volume
VDC	volts of direct current
VLSI	very large-scale integrated
VVVT	variable-volume variable-temperature
WHO	World Health Organization
WSHP	water-source heat pump
WWR	window-to-wall ratio

APPENDIX B

PSYCHROMETRIC CHART, TABLES OF PROPERTIES, AND I-P UNITS TO SI UNITS CONVERSION

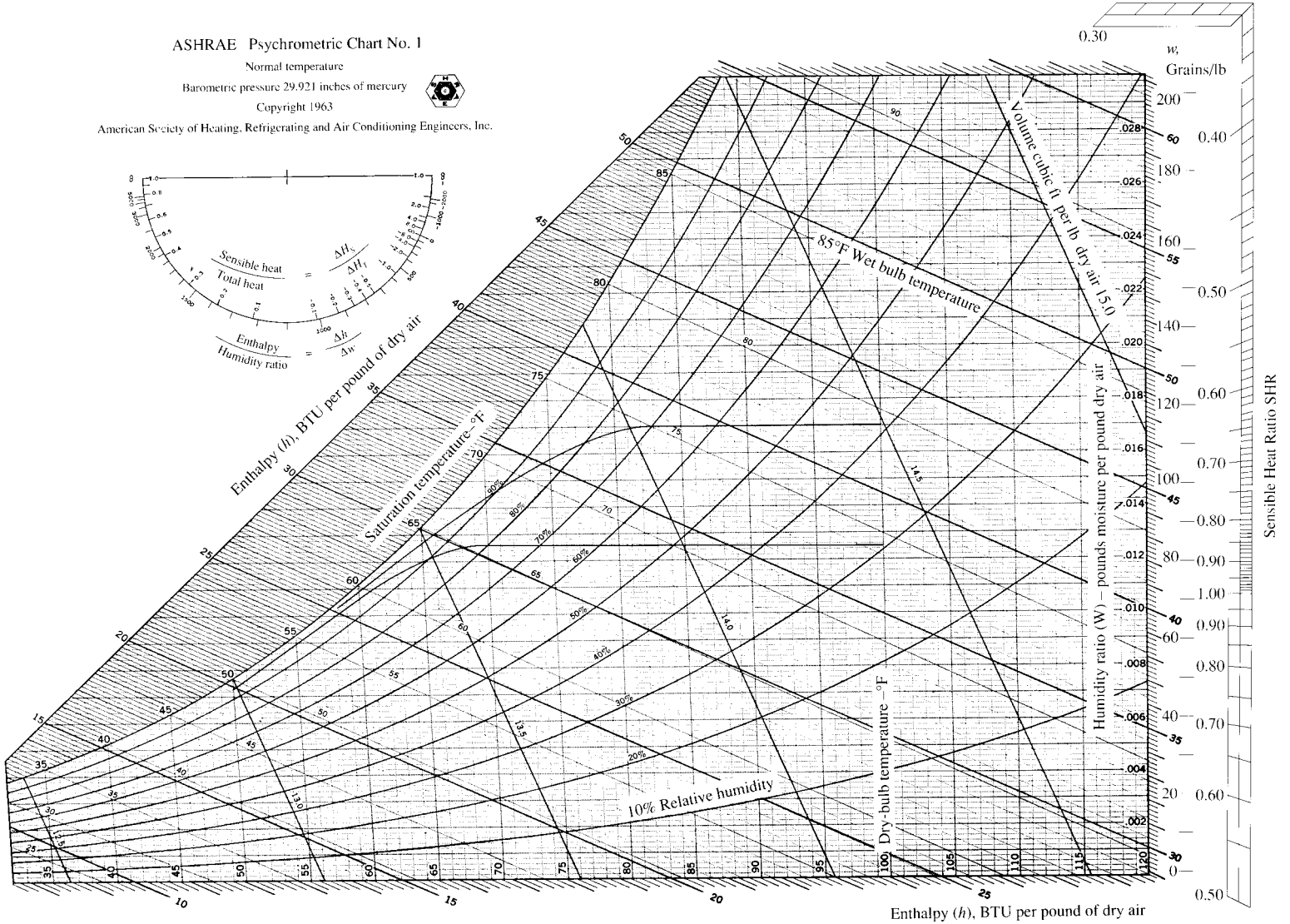
ASHRAE Psychrometric Chart No. 1
Normal temperature
Barometric pressure 29.921 inches of mercury
Copyright 1963
American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.
Sensible heat / Total heat = ΔH_s / ΔH_t
Enthalpy / Humidity ratio = Δh / Δw
Enthalpy (h), BTU per pound of dry air
Saturation temperature - °F
85°F Wet bulb temperature
Volume cubic ft per lb dry air 15.0
10% Relative humidity
Dry-bulb temperature - °F
Humidity ratio (W) – pounds moisture per pound dry air
w, Grains/lb
Sensible Heat Ratio SHR
Enthalpy (h), BTU per pound of dry air

FIGURE B.1 Psychrometric chart. Based on ASHRAE Psychrometric Chart No. 1. Reprinted with permission from ASHRAE Inc. Sensible heat ratio (SHR), humidity ratio scale in grains/lb, and two cooling and dehumidifying curves were added by author.

TABLE B.1 Thermodynamic Properties of Moist Air (at Atmospheric Pressure 14.696 psia) and Water

Temp T, °F	Humidity ratio w_s, lb_w/lb_{da}	Volume, ft^3/lb dry air			Enthalpy, Btu/lb dry air			Saturated water vapor Absolute pressure p		Enthalpy, Btu/lb		
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	psi	in. Hg.	Sat. water liq. h_f	Evap. h_{ig}/h_{fg}	Sat. water vapor h_g
32	0.003790	12.389	0.075	12.464	7.687	4.073	11.760	0.08865	0.18049	− 0.02	1075.15	1075.14
33	0.003947	12.414	0.079	12.492	7.927	4.243	12.170	0.09229	0.18791	0.99	1074.59	1075.58
34	0.004109	12.439	0.082	12.521	8.167	4.420	12.587	0.09607	0.19559	2.00	1074.02	1076.01
35	0.004277	12.464	0.085	12.550	8.408	4.603	13.010	0.09998	0.20355	3.00	1073.45	1076.45
36	0.004452	12.490	0.089	12.579	8.648	4.793	13.441	0.10403	0.21180	4.01	1072.88	1076.89
37	0.004633	12.515	0.093	12.608	8.888	4.990	13.878	0.10822	0.22035	5.02	1072.32	1077.33
38	0.004820	12.540	0.097	12.637	9.128	5.194	14.322	0.11257	0.22919	6.02	1071.75	1077.77
39	0.005014	12.566	0.101	12.667	9.369	5.405	14.773	0.11707	0.23835	7.03	1071.18	1078.21
40	0.005216	12.591	0.105	12.696	9.609	5.624	15.233	0.12172	0.24783	8.03	1070.62	1078.65
41	0.005424	12.616	0.110	12.726	9.849	5.851	15.700	0.12654	0.25765	9.04	1070.05	1079.09
42	0.005640	12.641	0.114	12.756	10.089	6.086	16.175	0.13153	0.26780	10.04	1069.48	1079.52
43	0.005863	12.667	0.119	12.786	10.330	6.330	16.660	0.13669	0.27831	11.04	1068.92	1079.96
44	0.006094	12.692	0.124	12.816	10.570	6.582	17.152	0.14203	0.28918	12.05	1068.35	1080.40
45	0.006334	12.717	0.129	12.8946	10.810	6.843	17.653	0.14755	0.30042	13.05	1067.79	1080.84
46	0.006581	12.743	0.134	12.877	11.050	7.114	18.164	0.15326	0.31205	14.05	1067.22	1081.28
47	0.006838	12.768	0.140	12.908	11.291	7.394	18.685	0.15917	0.32407	15.06	1066.66	1081.71
48	0.007103	12.793	0.146	12.939	11.531	7.684	19.215	0.16527	0.33650	16.06	1066.09	1082.15
49	0.007378	12.818	0.152	12.970	11.771	7.984	19.756	0.17158	0.34935	17.06	1065.53	1082.59
50	0.007661	12.844	0.158	13.001	12.012	8.295	20.306	0.17811	0.36263	18.06	1064.96	1083.03
51	0.007955	12.869	0.164	13.033	12.252	8.616	20.868	0.18484	0.37635	19.06	1064.40	1083.46
52	0.008259	12.894	0.171	13.065	12.492	8.949	21.441	0.19181	0.39054	20.07	1063.83	1083.90
53	0.008573	12.920	0.178	13.097	12.732	9.293	22.025	0.19900	0.40518	21.07	1063.27	1084.34
54	0.008897	12.945	0.185	13.129	12.973	9.648	22.621	0.20643	0.42031	22.07	1062.71	1084.77
55	0.009233	12.970	0.192	13.162	13.213	10.016	23.229	0.21410	0.43592	23.07	1062.14	1085.21
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.850	0.22202	0.45204	24.07	1061.58	1085.65
57	0.009938	13.021	0.207	13.228	13.694	10.790	24.484	0.23020	0.46869	25.07	1061.01	1086.08
58	0.010309	13.046	0.216	13.262	13.934	11.197	25.131	0.23864	0.48588	26.07	1060.45	1086.52
59	0.010692	13.071	0.224	13.295	14.174	11.618	25.792	0.24735	0.50362	27.07	1059.89	1086.96
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467	0.25635	0.52192	28.07	1059.32	1087.39
61	0.011496	13.122	0.242	13.364	14.655	12.502	27.157	0.26562	0.54081	29.07	1058.76	1087.83
62	0.011919	13.147	0.251	13.398	14.895	12.966	27.862	0.27519	0.56029	30.07	1058.19	1088.27
63	0.012355	13.172	0.261	13.433	15.135	13.446	28.582	0.28506	0.58039	31.07	1057.63	1088.70

TABLE B.1 (*Continued*)

Temp T, °F	Humidity ratio w_s, lb_w/lb_{da}	Volume, ft^3/lb dry air			Enthalpy, Btu/lb dry air			Saturated water vapor Absolute pressure p		Enthalpy, Btu/lb		
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	psi	in. Hg.	Sat. water liq. h_f	Evap. h_{ig}/h_{fg}	Sat. water vapor h_g
64	0.012805	13.198	0.271	13.468	15.376	13.942	29.318	0.29524	0.60112	32.07	1057.07	1089.14
65	0.013270	13.223	0.281	13.504	15.616	14.454	30.071	0.30574	0.62249	33.07	1056.50	1089.57
66	0.013750	13.248	0.292	13.540	15.856	14.983	30.840	0.31656	0.64452	34.07	1055.94	1090.01
67	0.014246	13.273	0.303	13.577	16.097	15.530	31.626	0.32772	0.66724	35.07	1055.37	1090.44
68	0.014758	13.299	0.315	13.613	16.337	16.094	32.431	0.33921	0.69065	36.07	1054.81	1090.88
69	0.015286	13.324	0.326	13.650	16.577	16.677	33.254	0.35107	0.71478	37.07	1054.24	1091.31
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097	0.36328	0.73964	38.07	1053.68	1091.75
71	0.016395	13.375	0.351	13.726	17.058	17.901	34.959	0.37586	0.76526	39.07	1053.11	1092.18
72	0.16976	13.400	0.365	13.764	17.299	18.543	35.841	0.38882	0.79164	40.07	1052.55	1092.61
73	0.017575	13.425	0.378	13.803	17.539	19.204	36.743	0.40217	0.81883	41.07	1051.98	1093.05
74	0.018194	13.450	0.392	13.843	17.779	19.889	37.668	0.41592	0.84682	42.06	1051.42	1093.48
75	0.018833	13.476	0.407	13.882	18.020	20.595	38.615	0.43008	0.87564	43.06	1050.85	1093.92
76	0.019491	13.501	0.422	13.923	18.260	21.323	39.583	0.44465	0.90532	44.06	1050.29	1094.35
77	0.020170	13.526	0.437	13.963	18.500	22.075	40.576	0.45966	0.93587	45.06	1049.72	1094.78
78	0.020871	13.551	0.453	14.005	18.741	22.851	41.592	0.47510	0.96732	46.06	1049.16	1095.22
79	0.021594	13.577	0.470	14.046	18.981	23.652	42.633	0.49100	0.99968	47.06	1048.59	1095.65
80	0.022340	13.602	0.487	14.089	19.222	24.479	43.701	0.50736	1.03298	48.06	1048.03	1096.08
81	0.023109	13.627	0.505	14.132	19.462	25.332	44.794	0.52419	1.06725	49.06	1047.46	1096.51
82	0.023902	13.653	0.523	14.175	19.702	26.211	45.913	0.54150	1.10250	50.05	1046.89	1096.95
83	0.024720	13.678	0.542	14.220	19.943	27.120	47.062	0.55931	1.13877	51.05	1046.33	1097.38
84	0.025563	13.703	0.561	14.264	20.183	28.055	48.238	0.57763	1.17606	52.05	1045.76	1097.81
85	0.026433	13.728	0.581	14.310	20.424	29.021	49.445	0.59647	1.21442	53.05	1045.19	1098.24
86	0.027329	13.754	0.602	14.356	20.664	30.017	50.681	0.61584	1.25385	54.05	1044.63	1098.67
87	0.028254	13.779	0.624	14.403	20.905	31.045	51.949	0.63575	1.29440	55.05	1055.06	1099.11
88	0.029208	13.804	0.646	14.450	21.145	32.105	53.250	0.65622	1.33608	56.05	1043.49	1099.54
89	0.030189	13.829	0.669	14.498	21.385	33.197	54.582	0.67726	1.37892	57.04	1042.92	1099.97
90	0.031203	13.855	0.692	14.547	21.626	34.325	55.951	0.69889	1.42295	58.04	1042.36	1100.40
91	0.032247	13.880	0.717	14.597	21.866	35.489	57.355	0.72111	1.46820	59.04	1041.79	1100.83
92	0.033323	13.905	0.742	14.647	22.107	36.687	58.794	0.74394	1.51468	60.04	1041.22	1101.26
93	0.034433	13.930	0.768	14.699	22.347	37.924	60.271	0.76740	1.56244	61.04	1040.65	1101.69
94	0.035577	13.956	0.795	14.751	22.588	39.199	61.787	0.79150	1.61151	62.04	1040.08	1102.12
95	0.036757	13.981	0.823	14.804	22.828	40.515	63.343	0.81625	1.66189	63.03	1039.51	1102.55

Abridged from *ASHRAE Handbook 1997, Fundamentals.* Reprinted with permission.

TABLE B.2 Physical Properties of Air (at Atmospheric Pressure 14.696 psia)

Temp. T, °F	ρ, lb_m/ft^3	c_p, $Btu/lb_m \cdot °F$	$\mu \times 10^5$, $lb_m/ft \cdot s$	$\nu \times 10^3$, ft^2/s	k, $Btu/h \cdot ft \cdot °F$	α, ft^2/h	Pr	$\beta \times 10^3$, $1/°F$	$g\beta\rho^2/\mu^2$, $1/°F^3 \cdot ft^3$
0	0.0862	0.240	1.09	0.126	0.0132	0.639	0.721	2.18	4.39×10^6
30	0.0810	0.240	1.15	0.142	0.0139	0.714	0.716	2.04	3.28
60	0.0764	0.240	1.21	0.159	0.0146	0.798	0.711	1.92	2.48
80	0.0735	0.240	1.24	0.169	0.0152	0.855	0.708	1.85	2.09
100	0.0710	0.240	1.28	0.181	0.0156	0.919	0.703	1.79	1.76
150	0.0651	0.241	1.36	0.209	0.0167	1.06	0.698	1.64	1.22
200	0.0602	0.241	1.45	0.241	0.0179	1.24	0.694	1.52	0.840
250	0.0559	0.242	1.53	0.274	0.0191	1.42	0.690	1.41	0.607
300	0.0523	0.243	1.60	0.306	0.0203	1.60	0.686	1.32	0.454
400	0.0462	0.245	1.74	0.377	0.0225	2.00	0.681	1.16	0.264
500	0.0413	0.247	1.87	0.453	0.0246	2.41	0.680	1.04	0.163
600	0.0374	0.251	2.00	0.535	0.0270	2.88	0.680	0.944	79.4×10^3
800	0.0315	0.257	2.24	0.711	0.0303	3.75	0.684	0.794	50.6
1000	0.0272	0.263	2.46	0.906	0.0337	4.72	0.689	0.685	27.0
1500	0.0203	0.277	2.92	1.44	0.0408	7.27	0.705	0.510	7.96

Source: *Fundamentals of Momentum Heat and Mass Transfer,* Welty et al., 1976. John Wiley & Sons. Reprinted with permission.

TABLE B.3 Physical Properties of Water (at Atmospheric Pressure 14.696 psia)

T, °F	ρ, lb_m/ft^3	c_p, $Btu/lb_m \cdot °F$	$\mu \times 10^3$ $lb_m/ft \cdot s$	$\nu \times 10^5$ ft^2/s	k, $Btu/ft \cdot °F$	α, ft^2/h	Pr	$\beta \times 10^3$, $1/°F$	$g\beta\rho^2/\mu^2$ $1/°F \cdot ft^3$
32	62.4	1.01	1.20	1.93	0.319	5.06	13.7	− 0.350	
60	62.3	1.00	0.760	1.22	0.340	5.45	8.07	0.800	17.2
80	62.2	0.999	0.578	0.929	0.353	5.67	5.89	1.30	48.3
100	62.1	0.999	0.458	0.736	0.364	5.87	4.51	1.80	107
150	61.3	1.00	0.290	0.474	0.383	6.26	2.72	2.80	403
200	60.1	1.01	0.206	0.342	0.392	6.46	1.91	3.70	1010
250	58.9	1.02	0.160	0.272	0.395	6.60	1.49	4.70	2045
300	57.3	1.03	0.130	0.227	0.395	6.70	1.22	5.60	3510
400	53.6	1.08	0.0930	0.174	0.382	6.58	0.950	7.80	8350
500	49.0	1.19	0.0700	0.143	0.349	5.98	0.859	11.0	17350
600	42.4	1.51	0.0579	0.137	0.293	4.58	1.07	17.5	30300

Source: *Fundamental of Momentum Heat and Mass Transfer,* Welty et al., 1976 John Wiley & Sons. Reprinted with permission.

TABLE B.4 Conversion of Inch-Pound (I-P) Units to International System of Units (SI)

Unit	Equivalents	Unit	Equivalents
atm	= 14.696 lb_f/in^2	$Btu/h \cdot cfm$	= 06209 $W \cdot s/L$
	= 33.91 ft of water	$Btu/h \cdot ft$	= 0.961 W/m
	= 29.92 in. Hg.	$Btu/h \cdot ft^2$	= 3.155 W/m^2
	= 1.013 bars	$Btu/h \cdot ft^2 \cdot °F$	= 5.678 $W/m^2 \cdot °C$
	= 101,325 Pa	$Btu \cdot in./h \cdot ft^2 \cdot °F$	= 0.1442 $W/m \cdot °C$
Btu (British thermal unit)	= 778 $ft \cdot lb_f$	Btu/lb	= 2.326 kJ/kg
	= 1055 J	$Btu/lb \cdot °F$	= 4.187 $kJ/kg \cdot °C$
	= 252 cal	$Btu/lb \cdot ft$	= 7.63 $kJ/kg \cdot m$
$Btu \cdot ft/h \cdot ft^2 \cdot °F$	= 1.731 $W/m \cdot °C$	$Btu/yr \cdot ft^2$	= 0.000293 $kWh/yr \cdot ft^2$
Btu/h	= 0.293 W	clo (clothing insulation)	= 0.155 $m^2 \cdot °C/W$

TABLE B.4 Conversion of Inch-Pound (I-P) Units to International System of Units (SI) (*Continued*)

Unit	Equivalents	Unit	Equivalents
clo	= 0.88 h·ft²·°F/Btu	kW/ton	= [3.516/(kW/ton)] COP_{ref}
ft³/lb	= 0.0624 m³/kg	L	= 0.001 m³
cfm (cubic foot per minute)	= 7.481 gpm		= 0.0353 ft³
	= 0.4719 L/s	lb_f	= 4.45 N
	= 0.02832 m³/min	lb/Btu	= 0.4786 kg/kJ
cfm/ft	= 1.548 L/s·m	lb·°F/Btu	= 0.2659 kg·°C/kJ
cfm/ft²	= 5.078 L/s·m²	lb_f/ft²	= 0.0069 lb_f/in²
	= 18.2 m³/h·m²		= 4.88 kg/m²
cfm/ton_{ref}	= 0.1342 L/s·k W_{ref}	lb/ft³	= 16.0 kg/m³
$/ft²	= 10.76 $/m²	lb/ft·h	= 0.413 mPa·s
°F	= (°F − 32)/(1.8)°C	lb_f/ft·s	= 1490 mPa·s
fc	= 10.76 lx	lb/h	= 0.126 g/s
fpm	= 0.01136 mi/h	lb/h·ft²	= 4.88 kg/h·m²
	= 0.00508 m/s	lb/lb	= 1.0 kg/kg
ft	= 0.3048 m	lb (mass)	= 7000 gr
	= 304.8 mm		= 16 oz
ft²	= 144 in.²		= 0.4536 kg
	= 0.0929 m²	lb of water	= 0.01602 ft³
ft³	= 0.748 gal		= 0.12 gal
ft·lb_f	= 1.356 J	L/s	= 2.119 cfm
ft·lb_f/min	= 0.0226 W		= 15.85 gpm
ft/s, fps	= 0.3048 m/s	m	= 1.094 yard
ft²/s(kinematic viscosity)	= 92,900 mm²/s		= 3.281 ft
ft WC	= 0.4334 lb_f/in²		= 39.37 in.
	= 2.99 kPa	met	= 58.2 W/m²
gal	= 0.1337 ft³		= 18.46 Btu/h·ft²
	= 8.35 lb of water	mg	= 0.01543 gr
	= 3.785 L	mil	= 0.001 in.
gpm (U.S)	= 0.0631 L/s		= 25.4 μm
gpm/ton_{ref}	= 0.0179 L/s·kW	mi	= 5280 ft
h·°F/Btu	= 1.911 °C/W		= 1.61 km
h·ft²·°F/Btu	= 0.176 m²·°C/W	mi/h	= 88 fpm
hp	= 33,000 ft·lb_f/min		= 0.44 m/s
	= 550 ft·lb_f/s	mm	= 0.03937 in.
	= 0.746 kW	mm Hg	= 133.3 Pa
hp (boiler)	= 33,476 Btu/h	μg	= 1×10^{-6} g
	= 9808 W	μm	= 1×10^{-6} m
in. (inch)	= 25.4 mm		= 3.94×10^{-5}in.
in. Hg (mercury)	= 0.4912 lb_f/in.²	oz	= 0.0625 lb
	= 3.3 kPa		= 28.35 g
in. WC (water column)	= 0.0361 lb_f/in.²	Pa (pascal)	= 1 N/m²
	= 5.20 lb_f/ft²	pint	= 28.37 in.³
	= 248.6 Pa		= 0.4732 L
in. WC/(cfm)²	= 5.27×10^5Pa·s/m⁶	ppm (mass)	= 1 mg/kg
in. WG (water gauge)	= 248.6 Pa + 1 atm	psia (absolute)	= 2.307 ft water abs.
J (joule)	= 9.48×10^{-4} Btu		= 703.1 kg/m²abs.
kBtu/ft²·yr	= 3.153 kWh/m²·yr		= 6.895 kPa abs.
kg	= 2.2046 lb	psig (gauge)	= 1 lb_f/in.² + 1 atm
km	= 3281 ft	quad (quadrillion)	= 1×10^{15} Btu
	= 0.6214 mi		= 1.055 EJ
kW	= 3413 Btu/h	qt (quart)	= 57.75 in.³
	= 1.341 hp		= 0.9461 L
kWh	= 3413 Btu	rad (radian)	= 57.3°

TABLE B.4 Conversion of Inch-Pound (I-P) Units to International System of Units (SI) (*Continued*)

Unit	Equivalents	Unit	Equivalents
rpm	= 1 r/min	ton (metric)	= 1000 kg
therm	= 100,000 Btu	ton_{ref} (refrigeration)	= 12,000 Btu/h
	= 105.5 MJ		= 3.516 kW
ton·h	= 12,000 Btu	ton (short)	= 2000 lb
	= 3.516 kWh	torr	= 1 mm Hg
ton (long)	= 2240 lb	W	= 3.413 Btu/h
	= 1016 kg		

Most of the conversion equivalents are based on values in *ASHRAE Handbook 1997, Fundamentals.*